TRANSACTIONS

OF THE

AMERICAN SOCIETY

OF

MECHANICAL ENGINEERS

1931

VOLUME 53

AER-WDI

PUBLISHED BY
THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
29 WEST THIRTY-NINTH STREET, NEW YORK, N. Y.

COPYRIGHT, 1932, BY
THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Printed in the United States of America by
The Mack Printing Co., Easton, Pa.

Chem. Eng. Lib.
Chem.-Engin.
Direct
6-2-30

Foreword

VOLUME 53 of the TRANSACTIONS OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS contains 27 individual sections published during 1931 under the sponsorship of the Society's 15 professional divisions, as well as its annual Record and Index. The technical papers and reports that make up this volume represent the Society's annual contribution to the permanent record of mechanical-engineering achievement. Most of these papers and reports were presented at meetings of the Society and of its professional divisions and local sections. Other papers published by the Society but not included in the Transactions may be located by means of the Record and Index which forms a part of this volume.

The papers published in this volume were issued originally in pamphlet form. These pamphlets appear in sections, each section being sponsored by one of the Society's professional divisions. A member of the Society may register in any three of the fifteen professional divisions, and receives in pamphlet form as they are issued the sections of Transactions sponsored by those divisions.

All sections of the Transactions are bound together at the end of the year for the convenience of libraries and of engineers who wish all of the papers in permanent form. Copies of the bound Transactions will be found in depositories located in selected engineering, university, and public libraries throughout the world. A complete list of these depositories will be found on page 97 of the Record and Index. Copies of the bound Transactions have also been set aside for sale.

At the end of Volume 53 will be found the Record and Index of the A.S.M.E. for 1931. This is the permanent record of the Society's activities for the year, and includes lists of committees, reports of meetings and committee activities, memorial notices of deceased members, as well as indexes of special A.S.M.E. publications and of *Mechanical Engineering*.

In this volume the papers of each section are grouped together and are numbered serially. The different sections follow each other alphabetically. Each paper is designated by a symbol composed of key letters followed by the volume number and paper number. The letters refer to the section of the Transactions to which the paper is assigned. Thus AER refers to the Aeronautic section, IS to the Iron and Steel section, etc. The arrangement of the index and designations for locating material are explained at the beginning of the index.

THE PUBLICATIONS COMMITTEE.

AER-53-1

Progress in Aeronautics

Contributed by the Aeronautic Division

Executive Committee: Charles H. Colvin, *Chairman*, Alexander Klemin, *Secretary*, Wm. F. Durand, Thurman H. Bane, Porter Adams, Wm. B. Mayo, and Edwin E. Aldrin

Aerodynamics[1]

WHILE the year 1930 has not been noteworthy for radical developments in applied aerodynamics, it may nevertheless be said that marked progress has been made in facilities for testing, in methods of obtaining and applying laboratory and flight test data, and in the further development of aerodynamic theory.

The construction of the world's first full-scale wind tunnel was started early this year at Langley Field by the National Advisory Committee for Aeronautics. This tunnel, with its 30-ft. × 60-ft. test section, will, when completed, easily accommodate airplanes of normal size.

In addition to its full-scale wind tunnel the N.A.C.A. has completed a new vertical type (with open throat), equipped with a continuous-rotation dynamometer for autorotation tests of airfoils and model airplanes.

Several new wind tunnels, among the most noteworthy being the large single-return type of California Institute of Technology, have been recently put into operation and are now busily engaged upon problems of fundamental importance. Furthermore two commercial firms have built complete aerodynamical laboratories, namely, the Sikorsky Aviation Corporation, with its vertical-jet wind tunnel, and the B/J Aircraft Corporation, of Baltimore, Md., with its open-jet double-return type.

A great deal of information for the future design of wind tunnels has been obtained from model tunnel experiments recently conducted by the N.A.C.A.

The research of Dryden and Kuethe, of the Bureau of Standards, has given us a clearer insight into the mechanism of turbulence, and has further emphasized the necessity of using it as a basis of standardization of wind tunnels.

The past successes of Karman, Polhausen, Prandtl, Bairstow, and others in solving certain fundamental problems in connection with the flow of viscous fluids, has more recently focused the attention of advanced students and physicists on many problems more directly related to aeronautics, such as the calculation of the drag of airship hulls, struts, airfoils, etc. In addition, however, many refinements have been made in the application of classical hydrodynamics to the solution of numerous practical problems involved in the design of airplanes, airships, propellers, helicopters, autogiros, etc.

The now classical theories of Prandtl and Munk have been applied with success to the determination of the rolling moment and hinge moment of conventional ailerons, the calculation of span and chord loadings of wings, the determination of stability and balance of airplanes, the calculation of the magnitude and distribution of load over tail surfaces during certain types of maneuvers, etc.

While the slotted wing has undergone more development in the laboratory than in theory, it may be stated that the calculation of the lift and moment of a multi-slot wing of elementary form by Tchapliguine, indicates that in certain cases we may expect increased inherent stability with a wing of this type. The recent development in England by Handley Page of the slot-and-aileron type of lateral control with an "interceptor" for increasing the effectiveness of the "up" aileron by spoiling the action of the slot on the wing to be depressed, has greatly minimized the dangers of the so-called "flat spin."

Paralleling the work of Handley Page is the further development of the floating aileron by the Curtiss Aeroplane and Motor Company. Also, this type of lateral control has been recently subjected to model tests by the N.A.C.A. in its vertical wind tunnel, and preliminary experiments have indicated in all cases either a marked reduction or absence of autorotative couple. Moreover, in addition to the laboratory studies of the floating aileron, investigators in England and in this country have subjected it to a thorough theoretical treatment in order to place its design on a rational basis.

During the past year or more, great progress has been made in coordinating the aerodynamical and structural phases of aircraft design. The Experimental Engineering Section (Matériel Division) of the Air Corps has carried out a large amount of research in connection with the vibration of wings and tail surfaces. In fact, the mathematical and experimental researches of Green at Wright Field, and of Frazer and Duncan at the National Physical Laboratory in England, have gone far toward placing the design of cantilever wings and tail surfaces on a sound basis.

In the realm of full-scale aerodynamic research—and apart from that already mentioned in connection with lateral control—we find the N.A.C.A. undertaking the most complete and systematic program of spinning research ever carried out up to the present time. It has been necessary to develop a number of special instruments for these and other tests, by means of which the committee is now able to determine among other things the forces and moments about the three axes of the airplane and the path of the center of gravity during a spin. By varying the mass distribution and aerodynamic arrangement of the airplane it will now be possible with these new facilities to ultimately obtain data whereby the designer will be able to predict the spinning characteristics of a proposed machine.

Commercial-aircraft concerns throughout the country have carried out an intensive development of the cowling of air-cooled engines, and this past year has witnessed the placing of venturi-type cowling and Townend ring design on a fairly sound engineering basis. The introduction of these improvements in the streamlining of aircraft with air-cooled engines has had a marked effect upon the design of commercial aircraft in this country.

Improvements in the autogiro have been noteworthy during 1930. The latest machines of the Pitcairn Autogiro Company are being equipped with a self-starter and a brake for the rotor. The former eliminates the necessity of ground taxiing in order to start the blades before take-off, and the latter enables the pilot to stop the rotor immediately upon landing. The latest autogiros are capable of a most satisfactory performance, and it is gratifying to note that Mr. Cierva, the inventor of this amazing craft, has developed the technique of design to such a high degree.

[1] Prepared by Wm. H. Miller, Assistant Chief Engineer, B/J Aircraft Corp., Baltimore, Md. Mem. A.S.M.E.

Propeller research during the past year is worthy of mention. There has been considerable progress in both theory and experiment. Notable contributions to design theory have been made by Roy in France and Pistolesi in Italy. The importance of the effect of peripheral speed on profile drag of the blades has been determined in a quantitative way by full-flight tests in England and by the staff of the Propeller Research Tunnel of the N.A.C.A in this country. It is of greatest importance to know that very serious losses in efficiency obtain at tip speeds in excess of 1000 ft. per sec., and valuable data are now available with which to estimate such losses.

The designer now has sufficient information to determine in any given case the advisability of using a geared propeller.

Progress in Airplane Design and Construction[1]

In spite of the depressed market for aircraft in 1930, the year seems to have been a most encouraging one when the progress made in airplane design and performance is considered.

Perhaps the increase of courage in design, and consequently of many new types—especially when we look back at the commercial plane of 1927 and 1928—might be said to be due to the increased competition brought about by poor business conditions. On the other hand, airplane designers have now a much wider choice with regard to selection of engines, and as a result, designs are becoming more varied all the time.

Similar in structural design to the Travel Airplane, developed in 1929, were the majority of the planes in the All-American Derby (for planes using Cirrus engines). These planes practically all have a low monoplane wing braced with wires to the landing gear, and to the upper portion of the fuselage.

However, the winner of the free-for-all speed contest at the 1930 National Air Racesin Chicago was a biplane—a Laird single seater with a fully cowled radial engine that governed the gap between the wings, these being tangent to the round fuselage, top and bottom.

Speed seems to have been the general aim this past year, and many designers have been willing to sacrifice a certain amount of pilot's vision in order to attain as much speed as possible.

Streamlining and close study of the aerodynamics of existing production types is evidenced by the new lines, fillets, and fairings on many types of the previous year—notable among these was the three-engine Ford. A number of excellent streamlined all-metal planes were developed, notably the low-wing Boeing Monomail, the Buhl mailplane, the metal Lockheed Vega, the Lockheed Sirius, and the further development of the all-metal Fleetster. The Monomail was an exceptionally modern design with cantilever wing and round fuselage, with ring-cowled engine. The landing gear retracted into the low wing.

Monoplanes seemed to be still increasing over the biplanes, with the number of externally and internally braced monoplanes almost equal. In England, where the biplane predominated for so many years, 1930 has found an increasing number of monoplanes from the De Haviland Moth series to the large Fairly and Short machines. The Bellanca sesquiplane type has continued in its development, and showed promise with the notable performance of the Bellanca Airbus.

A number of quite new experimental designs have been developed during the year such as the Northrup flying wing, the Emsco all-wing, the Waterman variable-incidence and variable-dihedral monoplane, and the McDonald Doodlebug, developed in 1929 for the Guggenheim Safe Aircraft Competition. Development of other new designs has continued, with the Curtiss Company experimentally applying a number of features of its Tanager to some of its other types, and with continuations of Furnic's and Locke-Wolf's experiments with the tail-first or Ente type. The Autogiro has been considerably improved by the Pitcairn Autogiro Company, which has developed a clutch-type starter as well as increased the performance of this promising plane.

Sliding and soaring planes have increased in remarkable numbers, and naturally their experience has been reflected in the development of many low-powered planes. These planes, which are produced very cheaply, are showing much promise for the private market.

During the year, the writer returned to America, and made a two months' tour of the centers of the aircraft industry. After having been abroad for some time, he was astounded at the progress made in metal development. Metal structures of all types had been brought to a high degree of perfection, with heat-treated chrome-molybdenum steel parts and sheet dural structures seeming to be the most in evidence. Welded and heat-treated steel spars, built-up sheet dural and alclad wings, and monocoque and semi-monocoque dural fuselages seemed to have appeared almost overnight. Manufacturers seemed to show confidence and experience in carrying out these designs. In England, the sheet-steel-strip type of spar construction was the most common, while most varied types of construction were to be found in Germany. Most firms were continuing with their work in dural and at the same time making tests and experimental airplanes from the magnesium alloy elektron. At the Ernst Heinkel Flugzeugwerke, in addition to the experimental work with dural and elektron structures, a series of all-steel spars (sheet-strip type) were developed to gain actual experience and knowledge of all of the types of construction and materials. The elektron structures seemed to be quite promising for inland usage where the air is dry, but at present the methods of preservation are not satisfactory enough to allow the material to be used in moist, salt air.

Hyblum, a comparatively new alloy claimed to have high resistance to corrosion, has come into use during the year, and seems to show much promise.

Stainless-steel sheet and strip for wing ribs and for use in floats subjected to severe corrosion conditions have been made during the year, and tried in service. The U. S. Navy and the British Air Ministry have sponsored a number of projects in stainless steel for use in sea water.

In detail design, a great deal of progress has been made in the standardization of parts and units, which will enable these to be marketed and produced much more efficiently and cheaply. Aluminum-alloy castings, especially in control systems and for complicated structural parts, have increased in usage. Metal ribs have come to be quite common, with most commercial ribs having specific strength-weight ratios of about 1400 to 1500. The Hall Aluminum Aircraft Co. claims to have developed ribs with ratios as high as 1800 to 3000.

In wing construction, especially in cantilever design, many interesting experimental structures have been made from the multi-spar to the monocoque type. The Fokker and Towle concerns developed interesting multi-spar dural wings having no ribs but depending on the skin and the number of longitudinal stringers to give the proper airfoil section. The Army Air Corps at Wright Field carried out an interesting series of experiments on monocoque-type wings.

Composite constructions (wood bracing and dural skin and gussets) for experimental or cheap construction have been developed to an increasing amount by Ireland and Aircraft Products, while Loening and Sikorsky, the best-known users of this type of structure, have shifted over to all-dural and they have standardized on their boat lines.

[1] Prepared by Richard M. Mock, an American aeronautical engineer at the time of writing. Now Consulting Engineer, Ernst Heinkel Flugzeugwerke, Warnemunde, Germany. Mem. A.S.M.E.

Balloon and semi-balloon tires have increased in use, and become standard equipment on many planes. However, on only the lightest planes or on racing types have they been used without shock absorbers. Hydraulic shock absorbers, both oil and spring and oil and air, have practically replaced the earlier rubber types.

Wheel brakes have been common on almost all commercial planes, and practically all of the Servo type. Sikorsky has developed a brake especially suited for amphibian use. It operates on the principle of a friction disk actuated by a hydraulic mechanism.

Floats and seaplane development have seemed to reflect Navy experience in the lines of seaplane floats and flying-boat hulls. By giving contracts to Martin, Douglas, Hall, and Keystone for building flying boats of the P. N. Type, the U. S. Navy has considerably aided flying-boat development in the United States. Sikorsky has developed his amphibian along larger lines, and is now working on a 32-passenger, four-engine plane of that type.

Ireland produced a light wire-braced monoplane amphibian flying boat, while Aeromarine Klemm and Edo have experimented with twin-float amphibians with wheels housed in the floats.

As in England, there seems to be some controversy as to the relative advantages of the twin-float seaplane versus the flying boat. The Ford plane and the English three-engined Short seaplane are examples of large twin-float designs.

In England, it is interesting to watch the development of seaplanes and flying boats. After considerable deliberation, service tests are being made on the twin-float seaplane, the flying boat with wing floats for lateral balance, and also the float boat with sponsons similar to those of Dornier except that they have a step across the bottom. Water rudders seem to be common on all flying boats and some seaplanes, especially the Italian.

Air Transport and Other Civil Flying[3]

The twelve-month period following that covered in the progress report for 1929 has been one of more or less continued development in air transport despite the effects of overexpansion in the manufacturing side and the general depression in business, both of which became very acute at this time. While flying schools felt this depression severely and while private flying did not continue the expansion of the previous period, the mail operators and the regularly operated passenger routes showed some substantial increases in traffic as well as some further extension of air lines.

TABLE 1 MILEAGE AND TRAFFIC STATISTICS FOR PRIVATELY OPERATED AIR-MAIL ROUTES WITHIN THE U. S. A. FOR YEAR ENDING JUNE 30, 1930[1]

Month	Number of routes	Length of routes, miles	Miles of service actually flown	Total weight of mails dispatched, pounds	Amount paid to contractors
July 1929....	25	13,928	1,260,980	645,653	$1,247,401
Aug. 1929....	25	13,928	1,308,186	701,538	1,361,619
Sept. 1929....	26	14,377	1,243,819	658,984	1,269,510
Oct. 1929.....	25	14,369	1,303,938	705,772	1,341,982
Nov. 1929....	25	14,369	1,222,201	624,750	1,213,433
Dec. 1929....	25	14,369	1,056,728	618,991	1,245,663
Jan. 1930.....	26	14,386	1,103,357	518,398	1,011,841
Feb. 1930....	25	14,387	1,096,591	556,913	1,095,105
Mar. 1930....	25	14,548	1,250,968	625,174	1,224,906
Apr. 1930....	25	14,565	1,309,240	676,255	1,296,945
May 1930....	25	14,659	1,392,439	702,499	1,188,961
June 1930....	25	14,907	1,389,821	680,814	1,134,419
Totals.....	..	....	14,938,268	7,715,741	$14,631,785

[1] Compiled from Post Office Dept. statistics by Black & Bigelow, Inc., Air-Transport Engineers.

Manufacturing and Operating. The latter part of 1929 and the first several months of 1930 covered a period of drastic deflation in the aeronautical industry, particularly affecting operators of schools and manufacturers of aircraft and engines. The primary cause of this deflation was overproduction and overexpansion in the several months preceding. During that period, most manufacturers turned out more airplanes and engines than the market was able to absorb, while schools were increased in number and capacity considerably faster than applicants became available. Many companies directed their energies more toward obtaining additional financing than to the expanding of their markets. The effect of this overexpansion combined with the general business depression resulted in a serious drop in sales and school enrolment. As a consequence, many of the smaller airplane and engine builders have been eliminated and school operations have had to be curtailed.

Air Mail. Air-mail volume continued to grow more or less regularly throughout the earlier part of this period, but a peak was reached in October of 1929 and an appreciable drop followed, the bottom being reached about January, 1930, after which the volume began to climb back toward that of the previous peak. A large part of the 1929-30 winter slump may be charged against weather conditions, which always interfere with air-mail operations in the first few months of the year, and which early in 1930 were unusually bad for the maintenance of regular flying operations. It is evident, however, that the failure of the air mail to continue its growth is not fully explained by weather, and it seems probable that the general business depression was one of the underlying causes. In addition to this, some uncertainty existed on the part of the carriers (with some consequent slacking of their efforts) due to doubt on their part regarding the effect of contracts to be made under the Watres Bill. This bill was passed by Congress early this year and authorizes the Postmaster General to contract for the carriage of air mail by existing passenger air lines on the basis of a fixed rate per mile. In return for this compensation, carriers will be required to make available to the Post Office a certain definite volume and a certain cubic-footage capacity, which is to be paid for whether fully used or not. At the time of writing, bids have been recently opened and the first contract under this law has been awarded. Others are awaiting decision by the Postmaster General. It is encouraging to note that from February of 1930 to date, air-mail volume has been generally increasing, and with these new mail contracts approaching settlement, further growth may be looked for in the closing months of the year.

Fare Reductions. Early in 1930 several of the larger operators put into effect drastic reductions in passenger fares, bringing the cost of travel by air very close to that of travel by rail, and, in at least one case, the air-line fare was made slightly less than the rail fare for the same trip. Public response to these reduced rates was almost instantaneous, traffic on many lines doubling almost overnight, while the increase in some cases reached as high as 400 per cent in a comparatively short time. After operating for several months at the reduced rates, some of the lines put into effect new tariffs carrying a slight increase over the rates in effect for a few months previous. These increases, being comparatively small, appear to have had little or no effect upon the growth of passenger traffic, and while very few passenger lines are paying at this time, the volume of traffic has shown most encouraging growth, and the operating losses of many operators have been substantially decreased. Parallel with these changes in rates, operating economies were effected in many cases by the reduction of personnel and more intensive use of the flying equipment, both of which combined to appreciably reduce operating costs. Indeed, some operators have substantially reduced their operating forces while increasing their traffic volume.

Consolidations. While the consolidations referred to in the

[3] Prepared by Archibald Black, President and General Manager, Black & Bigelow, Inc., New York, N. Y. Mem. A.S.M.E.

TABLE 2 AIR-TRANSPORT MILEAGE OF THE PRINCIPAL COUNTRIES—OPERATIONS OF SCHEDULED SERVICES IN AIRPLANE-MILES[1]

	1919	1920	1921	1922	1923	1924	1925	1926	1927	1928	1929
Canada	.....	.....	.....	.....	.....	.....	.....	.....	.....	2,297,000	4,200,000
France	.....	506,000	1,460,000	2,015,000	2,115,000	2,249,000	2,946,000	3,244,000	3,755,000	4,534,000	5,863,000
Germany	.....	.....	.....	843,000	886,000	1,860,000	3,070,000	3,816,000	5,922,000	7,114,000	6,571,000
Great Britain	168,000	644,000	225,000	717,000	943,000	936,000	862,000	840,000	769,000	916,000	1,189,000
Holland	.....	51,000	217,000	246,000	336,000	483,000	680,000	598,000	814,000	760,000	1,193,000
Italy	.....	.....	.....	.....	.....	.....	.....	.....	824,000	1,238,000	1,851,000
United States	393,000	880,000	1,828,000	2,329,000	1,743,000	2,221,000	2,911,000	4,407,000	6,009,000	10,673,000	20,243,000

[1] The figures in this table should be considered as being only approximate, as accurate records are not available in most cases. Compiled by Black & Bigelow, Inc., Air-Transport Engineers.

previous progress report were not continued through 1930, one combination of outstanding importance took place during this period. This was the consolidation of Pan-American Airways and the New York, Rio, and Buenos Aires Line, the former company having purchased the latter. Some realignments took place in other quarters, notably the absorption of National Air Transport by United Aircraft & Transport Corporation.

Accidents. The total number of accidents has been steadily increasing, but, on the other hand, the ratio between the total number of accidents and the total miles flown has been showing a steady improvement, the accident ratio decreasing more or less regularly in the past two years. A considerable amount of stunt flying is still being carried on, and this and reckless flying represent two important contributing causes of accidents at present. The Department of Commerce in 1930 began to apply its regulations with greater rigidity, and this effort to reduce accidents has been reflected in a substantial increase in violations of air-commerce regulations during the second and third quarters of 1930. Violations of these regulations dealing with acrobatic or dangerous flying headed the list of specific offenses. Other violations included flying without license, flying without navigation lights, or flying without identification numbering. The total number of violations charged was 278 in the third and 177 in the second, as compared with the total of 122 for the first quarter of the year. Accident statistics, for the preceding two years compare as follows:

	Jan.–June 1928	July–Dec. 1928	Jan.–June 1929	July–Dec. 1929
Total accidents, all services	430	692	774	924
Miles flown per accident, all services	38,337	78,308	72,612	85,433

Airports. The total number of airports and landing fields in use in this country during the period covered by this report increased from 1224 as of January 1, 1929, to 1527 at November 4, 1929, and 1655 at July 1, 1930. At the time of writing these airports are divided as follows: 500 municipal, 558 commercial privately owned, 79 army and navy, and 518 Department of Commerce and other intermediate or emergency fields and other Government fields. Considerable extension was made in line airways in this period, continuing the development program started a few years ago by the Department of Commerce Aeronautics Branch, and 583 airports and fields are now lighted. While considerable study has been given in various quarters to the design of airports, at present very little effort appears to have been made to apply this information in actual practice, due possibly to the tendency of many municipalities to place the entire design of their airports in the hands of the existing city engineer's staff without calling for aid from the aviation industry.

Progress of Naval Aviation[4]

Satisfactory progress is being made toward the completion of the five-year building program for naval aviation as authorized by Act of Congress, approved June 24, 1926. This act authorized an increase in useful airplanes to 1000 in five yearly increments. The last appropriating year contemplated by the act is the one ending June 30, 1931. It is expected that the aircraft to be procured under the 1930 and 1931 authorizations will raise the number of naval airplanes on hand to 1000 during the fiscal year 1932.

In the interest of improvement in safety in flying, the Bureau of Aeronautics maintains a detailed record of all failures of material and of all accidents. Each such occurrence is analyzed on the spot by a board of three officers, who submit findings of fact, express their opinions as to the cause, and make pertinent recommendations. These reports and records enable the Bureau to take such steps toward the elimination of material faults and make such recommendations for the control of improper flying and personnel derelictions as will best contribute to safety in aviation, and yet not restrict the employment of aircraft in the proper development of this new art of naval warfare. During the past year a total of 266,984 hours were flown by naval airplanes with but 14 fatal accidents and 18 fatalities. For every fatality there were thus flown a new record total of 14,832 hours, and for each fatal accident, 19,070 hours. These figures compare with the previous best records in 1929 of 6773 and 11,289 hours, respectively. This new record may also be expressed by stating that in a total of 252,095 flights during the past year only 14 resulted in fatal injuries to occupants of the aircraft, or in other words, flights were 99.9945 per cent safe from fatalities.

The past year has witnessed progress in the elimination of wood from aircraft structures. No new construction for several years has used wooden fuselages, and the passing of the fiscal year 1930 carried with it the last of the wooden-fuselage airplanes of the Navy. Wooden wing ribs and wooden tail structures have been entirely replaced by metal construction in new designs, while almost all new experimental types and new-production aircraft have metal wing beams as well.

One of the recent interesting material investigations has been in the field of corrosion-resisting steels. Streamline tie rods made of various steel compositions are being service tested; seaplane floats of stainless steel are under construction. Wing ribs constructed of spot-welded strip stainless steel are being incorporated in certain wings for service tests. Sources of supply for this material in sheet, bar, and seamless tubing, and methods of handling and working it, are under development.

A step in the solution of our difficulties in the use of aluminum alloys in the construction of hulls and floats has been the discovery that incorrect heat treatment has been responsible for much of the rivet corrosion. Tests indicate that improved methods, which will insure all rivets of obtaining correct and uniform heat treatment, will largely eliminate this trouble. After experiment and service tests the anodic treatment, whereby aluminum and aluminum alloys are given electrolytically a coating of aluminum oxide, has been adopted as the standard method of preparing these materials to receive paint finishes. This treatment not only provides a surface to which finishes adhere readily, but one which has also considerable corrosion resistance in itself.

Because of their advantages in weight, reliability, cost, durability, ease of maintenance, and performance at altitude, the policy has been continued of using radial air-cooled engines

[4] Prepared by Rear-Admiral William A. Moffett, U.S.N., Chief of the Bureau of Aeronautics, Washington, D. C.

in all service airplanes. The Bureau has kept touch with all lines of engine development by awarding experimental contracts for liquid-cooled, air-cooled in line, Diesel (radial), and several types of foreign engines, but at present the American radial air-cooled seems the best for naval uses. Definite progress has been made in increasing the power of these engines by better cylinder and crankshaft design, and by the use of higher compression ratios and intake pressures.

The chief requirements of an airship engine are durability and fuel economy. The Maybach engines, which are to be used by the Navy in the *ZRS-4*, possess these requirements to a marked degree. The Bureau looks to the compression-ignition heavy-oil engine, however, as the airship engine of the future, and has been concentrating its efforts and experimental fund on its development. Two such experimental engines are now under contract, and both are scheduled for delivery within the present fiscal year.

Good progress has been made by the Goodyear-Zeppelin Corporation in the construction of the airship *ZRS-4* (U.S.S. *Akron*). The contractor's shed was ready for occupancy in October, 1929, and shortly thereafter, with appropriate ceremonies, the keel of the airship was laid. The contractor is working to a schedule which calls for completion of the airship in June, 1931, and it is believed this date can be met. A number of developments have been incorporated in the design as the work proceeded; however, there have been only a few minor changes involving modification of the contract requirements.

Rigid-airship operations throughout the year have shown definite progress toward the solution of the problem of handling large airships. The mobile mooring mast has become a regular part of the handling equipment. Other mechanical appliances which play necessary roles in a complete mechanical-handling system, have been developed and are now in habitual use. Improvements are necessary, but an adequate system for handling airships mechanically is definitely visualized, and requires only time and effort for its realization.

Operations in the Fleet have shown the necessity for the modification to a certain extent of previous ideas as to the proper types and models of airplanes suitable for use in naval aviation. Patrol flying boats used in the Fleet and at air stations were originally developed by the United States Navy. Smaller aircraft assigned to cruising vessels were originally adapted from models similar to those used in military operations on shore. It is interesting to note that practically all changes that have been made in naval aircraft in the last few years, except such as have been due to the development of the art, have been in the line of introducing features that fit them to operate from ships. Ship-based landplanes have emergency flotation gear for keeping them afloat in a forced landing at sea, and have carrier arresting gear and low stalling speeds to permit satisfactory operation from the decks of carriers. Most naval planes have radio in order to maintain touch with their moving bases. Observation planes have sturdy float-type landing gear to permit landing in rough water, and the use of the carriers has emphasized the need for a satisfactory but light amphibian landing gear. The size of fleet airplanes must be kept at a minimum in order that the limited space on board ship may be utilized to the best advantage. The differences in coordinate types of aircraft as used by the Army and Navy have arisen not because the airplanes are produced by different organizations, but because the peculiar conditions under which they operate must be taken into consideration from the inception of their design. Although it is considered sound policy to build as few different kinds of naval airplanes as possible for meeting the needs of the service, recent actual operation of naval aircraft in the Fleet has indicated that greater differentiation than now exists may be desirable if naval aircraft are to perform satisfactorily the service required. The Bureau constantly endeavors to keep in the closest touch with Fleet operations, and with the latest ideas that have been evolved in the Fleet, in order that airplane types entirely suitable for the purpose may be produced without delay.

Naval operations of the fiscal year have served further to emphasize the great value of aircraft carriers to the Fleet, and to indicate the urgent necessity of continuing our progress along these lines by building up to the carrier tonnage allowed under the London Limitation of Armament Treaty. This is considered the most vital need of naval aviation at the present time.

Development of Military Aeronautical Equipment in 1930[5]

Development of military aeronautical equipment during 1930 has been carried on under the direction of the Chief of the Air Corps by the Matériel Division, located at Wright Field, Dayton, Ohio. The five-year program, calling as it does for a greater number of up-to-date airplanes in the hands of service squadrons, has stimulated interest in new equipment. It is not the desire of the Air Corps to darken the skies with military aircraft, nor to spend vast sums of money on purely military flying, but it is the duty of the Air Corps to the nation to thoroughly and painstakingly investigate and test a large amount of promising experimental equipment. The value of governmental researches and development to the commercial-aircraft industry, as well as providing satisfactory equipment for national-defense purposes, has justified the soundness of continuing an experimental program. Closer cooperation with aircraft manufacturers and an increasing willingness to try out less conventional equipment is resulting in machines of increased performance and greater effectiveness. The lessons learned from joint Army and Navy maneuvers, the extensive Air Corps maneuvers held last spring in the vicinity of Sacramento, California, and the trend of high-performance commercial aircraft as well as the 36,000,000 airplane-miles actually flown last year by the Air Corps, are being applied directly to the new designs now in embryonic stages.

During the year two new designations of airplanes have been added—"Photographic" and "Basic Training," bringing the number of types to eight, i.e., Attack (symbol A), Bombardment (B), Transport (C), Observation (O), Pursuit (P), Photographic (F), Primary Training (PT), and Basic Training (BT). The symbol prefixed by "X" denotes that the airplane is "Experimental" and when prefixed by "Y" denotes a "Service Test" (in the hands of operating squadrons) status. Increased performance in all types except training has been attained by reduction of resistance through refinement of detail, installation of engines of increased horsepower, new methods of construction, and the use of accessories such as ring cowls for air-cooled engines, wheel streamlines, etc.

Engine development, upon which so much of the effectiveness of aircraft as a military weapon depends, has been carried on unceasingly along two general lines: direct air cooling and liquid cooling. While air-cooled engines dominated the field in commercial and naval operations and took a most important part in Air Corps activities, yet water cooling was still much in evidence. With the increasing of the power plants of observation airplanes from 425 hp. to 600 hp. the geared water-cooled Curtiss Conqueror was generally employed. The substitution of ethylene-glycol for water as a cooling liquid so successfully employed during 1929 in 12-cylinder 425-hp. engines, met certain problems when applied to engines of 500 hp. or greater. Recent design changes in larger engines appear to have overcome the diffi-

[5] Prepared by Lieut. S. P. Mills, Matériel Division, Air Corps, U. S. A., Wright Field, Dayton, Ohio.

culties encountered and promise to provide a high-powered liquid-cooled power plant whose total installed resistance will be a joy to the airplane designer. It should be remembered that the cooling liquid in such installations is delivered to the radiator at 300 deg. fahr. instead of 180 deg. fahr., giving twice the temperature difference attained with water under normal flying conditions. A radiator of one-fourth to one-third the size formerly used with the attendant saving in amount of liquid used effects a substantial saving in frontal area and total weight. Greater fuel economy with only about 2 per cent decrease in maximum engine power is also a highly desirable result.



CURTISS B-2 BOMBER POWERED WITH TWO 600-HP. WATER-COOLED CURTISS ENGINES

(Used in limited numbers by the Air Corps. Has on several occasions carried a 4000-lb. bomb.)



KEYSTONE BOMBER POWERED WITH TWO 525-HP. HORNET AIR-COOLED ENGINES

(Standard equipment for Air Corps. Carries crew of four and is capable of carrying 2000-lb. load.)

Development of air-cooled radial engines has been carried on vigorously, centering about the Pratt-Whitney Hornet and Wright Cyclone, each rated at 525 hp., which have been used extensively in bombardment airplanes. Models of these engines with cylinder displacements of 1850 cu. in. will develop 575 hp. The coming year will undoubtedly witness the more extensive use of other higher-powered types, some of which will be geared. The Wright V-1460, an inverted air-cooled 12-cylinder V-type of engine rated at 600 hp., has been tested in flight as well as on the dynamometer stand, showing good progress in this development.





TWO VERY INTERESTING SINGLE-ENGINE TRANSPORT MONOPLANES, THE FOKKER C-14 AND THE BOEING ALL-METAL PLANE

(The high-wing plane shown above is a Fokker C-14 with plywood wing and steel-tube fuselage, powered with a Pratt & Whitney Hornet engine. Below is shown the Boeing all-metal monomail low-wing transport plane. Each plane has high speeds above 140 m.p.h.)

The testing of a 1400-hp. (at 2000 r.p.m.) air-cooled 24-cylinder "X" type of engine for securing engineering data has proved interesting. This engine has four rows of six cylinders, each with a $5^3/_4$-in. bore and $7^1/_4$-in. stroke, set at 90-deg. intervals about a 6-cylinder-type crankshaft. When tests are completed upon this 4520-cu. in. engine, excellent data on the behavior of air-cooled cylinders in large sizes should be available.

With the advent of fuels of improved anti-detonating properties it has been possible to increase the compression, securing as a result an even better power-weight ratio and a lower fuel consumption. The 425-hp. Pratt-Whitney Wasp air-cooled engine as used in the late Boeing pursuit airplanes is now equipped with supercharger blowers geared 10 to 1, increasing the performance at altitude to a marked degree.

Three Diesel engines purchased from the Packard Company are now being tested in the dynamometer laboratory and in flight. This interesting engine of 240 hp., America's first Diesel-type aircraft engine, uses as fuel furnace oil instead of gasoline and eliminates the electrical ignition system. When supplied with fuel for several hours' operation, the overall power-weight ratio is very favorable. An apparent advantage arising from the use of this compression-ignition type of engine is the greatly reduced fire hazard due to the less inflammable fuel carried. Economical operation costs plus freedom from radio interference experienced in the presence of conventional ignition systems constitute additional advantages.

Less spectacular, but of far-reaching importance, is the development work on fuels making possible the use of higher compression and greater supercharging, with the attendant fuel economy. Studies in the torsional vibration of engine crankshafts by means of a newly designed torsiometer have resulted in correct counterbalancing and the determination of certain speeds which should be avoided in regular operation. A graphic record showing distortion or twist of crankshaft is easily obtainable at any desired speed.

Two other developments include the automatic supercharger regulator whose operation is governed by the change of barometric pressures due to altitude, and the solid-fuel-injection system using a fuel pump and spray jets for improving the fuel distribution.

During the coming year radial air-cooled and V-type liquid-cooled engines of over 500 hp. will be used in numbers, with the air-cooled "V" and "in-line" struggling for a foothold. The tendency of the times seems to indicate a material increase in m.e.p. due to increased supercharging coupled with the use of new and improved fuels. The use of gearing will probably increase, and controllable-pitch propellers will undoubtedly make their appearance on commercial as well as military aircraft.

Progress in the development of the airplane proper has continued along certain very definite lines. A marked tendency toward the use of monoplanes is noted in the experimental programs, although the biplane is very much in the majority in the hands of service organizations. Monoplanes have been or will be tried out in each one of the eight different types of aircraft used by the Army. In construction the tendency appears to be toward the all-metal machines incorporating the smooth-

surface monocoque type of fuselage and wings whose structure and covering are metal. The use of duralumin construction is undoubtedly growing, although again the majority of machines in actual use today are of the steel-tube, welded-joint type. Many studies have been carried on with a view to reducing the resistance and the weight of structures of a given strength in order to increase the performance. A short discussion of the airplanes in present use, and a word about the tendency of design in each of the basic types, may prove of interest here.

The present attack type of airplane, designed for exceptionally low-altitude flying and surprise attacks with machine guns and small bombs, is a biplane built by Curtiss and known as the A-3B. This two-place airplane mounts M-2 machine guns in the wings just outside of the propeller disk and is equipped with a Browning M-2, .30 caliber, flexible machine gun in the rear seat. The power plant consists of a Curtiss D-12 engine of 450 hp. giving a high speed of over 140 m.p.h. Low-wing monoplanes have been suggested, but as yet none have been tested. Such planes equipped with powerful water-cooled engines should add materially to the effectiveness of this type.

In the bombardment class the Keystone B-3A biplane powered by two 525-hp. Pratt & Whitney Hornet engines is now standard equipment. A new contract for 60 of these airplanes has been placed, deliveries of which have already commenced. This type of airplane may also come equipped with 525-hp. Wright Cyclone engines. Geared air-cooled engines have not as yet been used extensively in bombing airplanes, but may make their appearance at some future time. The gear ratios generally employed are 2:1 or 3:2.

The Curtiss B-2 biplane with two 600-hp. geared Curtiss Conqueror water-cooled engines is used in smaller quantities. Gearing may be expected in several of the new bombers as it affords a substantial increase in speed and climb. Late improvements in the Keystone bombers include greatly improved instrument boards, engine shutter controls for altitude and winter flying, and suitable tail wheels as a replacement for skids.

During the past year many commercial companies have submitted transport airplanes for Air Corps inspection. While the Ford and Fokker tri-engined transports have been continued as standard, yet there is a tendency toward single-engined lighter machines. A new type which will be used by the Service this coming year in limited numbers is the Fokker high-wing C-14. With its plywood wing raised 18 in. above the fuselage and the pilot mounted in an open cockpit aft, it should prove useful for transporting supplies, spare engines, etc. For many military



Douglas O-25 Observation Airplane Powered With Curtiss Conqueror 600-Hp. Prestone-Cooled Engine (Standard Equipment)



Thomas Morse O-19-B Observation Airplane Powered With Pratt & Whitney 425-Hp. Wasp Engine
(Standard equipment for Air Corps. Note corrugated duralumin body.)



Experimental Fokker 3-Place Observation Airplane Powered With Two 600-Hp. Curtiss Conqueror Engines
(Landing chassis is retractable into the engine nacelle.)



Experimental Two-Place Pursuit Biplane Built by Berliner-Joyce Co. Powered With 600-Hp. Curtiss Conqueror Prestone-Cooled Engine
(Note gull-wing effect for increase of visibility in a dive.)



Experimental Curtiss Pursuit Plane Powered With 600-Hp. Prestone-Cooled Curtiss Engine
(Note gull-wing construction and presence of wing-skin radiators.)



Boeing All-Metal Experimental Pursuit Plane XP-9 Powered With 600-Hp. Prestone-Cooled Curtiss Engine
(Note smooth-surface duralumin monocoque fuselage.)

requirements the lighter single-engined transport appears more practicable.

Marked progress in observation aircraft performance has been effected by increasing the power from 425 (Standard Liberty or Curtiss D-12) to 600 hp., which brings this class of machine definitely in the 150-m.p.h. class. Fifty Douglas O-25 A planes each powered with a water-cooled geared Curtiss Conqueror engine and mounting the new Browning .30 caliber machine guns, have been placed in the hands of the Service. Rate of climb and ceiling with a very quick take-off feature the model. The steel-tube fuselage and wood wing structure follow standard practice. A new rear gun mount eliminating the familiar ring mount and improved brakes add much to the safety of this machine. The Curtiss O-1E with a 425-hp. Curtiss D-12E engine is standard equipment. The Thomas Morse O-19B powered with a Pratt & Whitney Wasp air-cooled engine is also being used quite extensively. A new series of Douglas observation airplanes (O-38) powered with a Pratt & Whitney Hornet or Wright Cyclone engine of 525 hp. promises to contribute this year. In spite of the fact that all standard planes in this class are now of biplane construction, new types of construction in high-performance, cantilever, monoplane wings may bring about radical changes.



"The Longest Shot"—Aerial View of Mt. Rainier From a Distance of 266 Miles
(Photograph made possible by use of infra-red filters. Mt. Rainier is invisible to the eye at a much shorter distance than 266 miles.)

In the experimental field in this class the Fokker O-27 provides a most interesting exhibit, combining a cantilever wood wing in which two Prestone-cooled Curtiss Conqueror engines are streamlined, a small-section steel-tube fuselage, and a retractable landing chassis. The pilot and two observers sit in open tandem cockpits. Performance trials are now under way to determine its suitability for Service use. To the casual observer on the ground watching it pass, its two 600-hp. engines wide open and its landing chassis drawn up into the engine nacelles, it seems like a step in advance. Speeds in the 1929 pursuit class are hoped for but only time will tell.

That the pursuit plane must look to its laurels if it is to maintain a liberal margin of high speed over certain larger and heavier types, seems to have been demonstrated in British maneuvers in the defense of London. Light bombardment planes of the two-place class powered with water-cooled engines made a splendid showing, and did not appear to be particularly worried by the pursuit machines.

In our Service the Curtiss Hawk P-1C and the Boeing P-12C are the two standard types of pursuit. The P-12C incorporates certain improvements, among which is a ring cowl about the supercharged Wasp engine. With the cowl only about 12 in. wide, the visibility is not materially reduced. In order to gain increased performance at higher altitudes, the supercharger blower is geared 10:1. By careful handling of the throttle and not overloading the engine at ground levels, reasonably satisfactory maintenance may be expected. Full-capacity operation should take place only at altitudes above 8000 ft. With its short span and compact weight arrangement, it is a very maneuverable machine. The fuselage construction is of the square-tube, rounded-corner duralumin longeron type, riveted and bolted.

The Curtiss P-1C has been improved from time to time, but is still powered with a water-cooled Curtiss D-12E engine of 425 hp. It is probable that water-cooled pursuit airplanes will be generally of a higher horsepower than the P-1C.

Among the experimental airplanes being performance-tested at this time which may affect pursuit design, should be mentioned the two-place Berliner-Joyce, the Curtiss XP-10, the Boeing XP-9, and the Curtiss XP-17. The two-place Berliner-Joyce airplane is powered with a 600-hp. Prestone-cooled Curtiss Conqueror engine. This plane with the gull-wing type of construction in the upper wing has every appearance of a pursuit airplane, but the presence of a gunner immediately back of the pilot will afford a great protection from the pursuit pilot's weakness—vulnerability in case of an attack from the rear and above. The pilot's visibility in this plane during a dive should be excellent. Speeds comparable with last year's single-place pursuit airplane are hoped for in the tryouts to follow. A steel-tube fuselage and fabric-covered metal wing are used.

The Curtiss XP-10 pursuit airplane is built around the Prestone-cooled Curtiss Conqueror engine and incorporates the gull-wing construction in the upper wing. In order to attain the maximum speed, an improved type of wing-surface radiator has been applied to the upper wing extending about 5 ft. on each side. As in the Berliner-Joyce, a direct-drive engine is used. The airplane has been designed with a view toward securing a satisfactory high-speed performance, particularly at altitude, and a rapid climb. Performance tests on this machine will be completed by the end of the year.

The latest pursuit airplane produced by Boeing, known as the

XP-9, is of the externally braced monoplane type powered with a Prestone-cooled Curtiss Conqueror 600-hp. engine. The XP-9 is of unusual interest in that its fuselage is of the all-duralumin smooth-surface monocoque type and its wing construction entirely of metal. An axle type of landing gear, flat-nose-type radiator, and synchronized Browning M-2 machine gun are used.

The Curtiss XP-17 pursuit airplane powered with an inverted 12-cylinder air-cooled Wright engine of 575 hp. is now being experimented with. In the early stages of development this airplane is interesting due to the exceptionally good visibility afforded the pilot. Due to the inverted position of the engine a considerable amount of noise, dirt, and vibration appear less noticeable to the pilot. A considerable number of experiments have been conducted with modified forms of equipment. One such experiment is the use of a Heinrich low-resistance flat-tube radiator used in connection with a Prestone-cooled Curtiss D-12 engine whose compression ratio is above 7:1. A marked increase in high speed coupled with increased rapidity of climb resulted from this installation.

The Curtiss P-6 pursuit airplane powered with a 600-hp. Prestone-cooled Curtiss engine will soon be equipped with the new improved Conqueror engine designed especially for Prestone cooling and will then be given additional service tests.

The present tendency in pursuit development is to increase the high-speed performance, particularly at altitude, by the use of engines of greatly increased horsepower which are more highly supercharged. In all probability, future development in this type of machine will swing toward the smooth-surface all-metal fuselage and metal-covered wings. Duralumin construction will undoubtedly play a most important part in this development. The surprising increase in speeds of larger and heavier machines is forcing the pursuit to reduce resistance to the minimum and to use overpowered engines. The use of retractable landing gears, and even an inclosure for the pilot, have been suggested in the past and may be given more serious attention than heretofore.

The progress in aerial photography and mapping has been steady, although not particularly sensational. Many excellent photographs have been taken from points 30,000 to 40,000 ft. above the earth. One photograph taken with the aid of infrared filters shows the summit of Mt. Rainier at a distance of over 266 miles. The type of airplane now in the hands of the Service for photographic use is the Fairchild YF-1, which is undergoing extensive service tests. This externally braced monoplane powered with an air-cooled Wasp engine provides a roomy cabin for operation of the cameras and developing or printing, if desired. The high-wing type of monoplane affords the photographer good visibility for the taking of oblique photographs through the door, as well as for the usual verticals taken through the floor. Among the characteristics necessary in a photographic plane should be mentioned stable flying qualities and the ability to reach 15,000 ft. with the required load.

The past year has witnessed an investigation as to the primary training plane best suited to Air Corps needs. During the past 20 years we have watched the Army pilots being trained on the 30-hp. Wright engines, then the 75-hp. Curtiss, then the famous 90-hp. Curtiss OX-5 of the "Jenny era." Following the war period the power of training airplanes was increased to 150 and then to 180 hp. by adopting the Wright Hispano engines. With the coming of successful air-cooled engines, the power was again raised, until at present the power of training airplanes in service is 225, using the Wright J-5B Whirlwind engine. In view of improvements in airplane construction it is now believed unnecessary to use so great an amount of power. Exhaustive service tests are now under way to determine if the small airplane, such as the "Fleet" PT-6, powered with a 100-hp. Kinner engine or higher-powered planes such as Stearman, Verville, and others, will more satisfactorily meet A.C. requirements. There is naturally a considerable difference of opinion regarding this point. Some pilots believe that a medium-heavy plane using 150-hp. would be satisfactory. The tendency, however, seems to be toward a reduction in weight and horsepower and the use of a machine whose flying qualities are comparable with those of the more powerful service aircraft.

As a transition airplane to be used by newly trained pilots between their primary training and training on service-type airplanes, a new type known as "basic training" has been adopted by the Air Corps. At present this type is being supplied by the Douglas Company in a model known as the BT-2 series. The construction of this plane is similar to that of the Douglas O-2H



DOUGLAS BT-2B BASIC TRAINING PLANE POWERED WITH WASP 425-HP. AIR-COOLED ENGINE
(Note ring cowl, which increases speed about 5 m.p.h.)



A TYPICAL LIGHT TYPE OF TRAINING PLANE—THE "FLEET" XPT-6, POWERED WITH 100-HP. KINNER ENGINE—NOW ON SERVICE TEST

and O-2K observation, but its power plant is the 425-hp. Wasp air-cooled engine instead of the Liberty engine. The use of airplanes of this type is a very important step in the training of new pilots, and will undoubtedly fully justify its cost in the conservation of fully equipped and more expensive service airplanes.

Although little of a revolutionary nature has been advanced in the progress of military aeronautical equipment during the past year, yet the steady advance in airplane construction marked by the wider use of metal and cleaner designs together with corresponding advancement in power plant, higher compression, greater supercharging, and perfection of liquid cooling, has paved the way for a material advancement in performance.

AIRSHIPS[6]

The year 1930 was marked by one spectacular airship disaster, and by comparatively little progress of the sort that fills the public eye. Nevertheless very real progress was made in improving methods of airship mooring and ground handling, which have long been among the most important and intractable of airship problems. An intensive practical and theoretical investigation

[6] Prepared by C. P. Burgess, Bureau of Aeronautics, Navy Department, Washington, D. C.

of mooring and ground-handling methods has been carried out by the U. S. Navy, and the results have been tested and demonstrated in the handling of the U.S.S. *Los Angeles*.

The low, or stub, mooring mast developed at Lakehurst has now been in successful use for several years. With this mast the airship rides close to the ground, its stern resting upon a car running on a circular track, concentric with the mast. The airship is free to turn like a weather cock, always pointing into the wind; but all vertical motion is eliminated and the airship is conveniently accessible for servicing from the ground. An objection to the low mast was that, until recently, it was necessary to land the airship upon the ground by man power, in the old-fashioned way, and walk her to the mast. In the past year, the technique of mooring has been so improved that direct-flying moors to the low mast have become standard practice.

A very important development in ground handling has been the use of the mobile stub mooring mast at Lakehurst. This mast is mounted at three points on Athey truss wheels, which are a modification of the caterpillar-tread principle. The mast, with the bow of the *Los Angeles* attached to it, is towed by a tractor into and out of the hangar. It was also used for handling the *Graf Zeppelin* when she visited Lakehurst in June. A new system of side handling lines, attached to the ship and docking trolleys, has been developed to hold the ship parallel to the hangar when being taken in or out by means of the mobile mast. These handling lines are arranged to distribute the side force evenly over a large part of the ship's structure.

The *Graf Zeppelin* has continued in successful service throughout the year. Her numerous voyages in and across Europe, extending from Spain to Asia Minor, and from Africa to the North Cape, have become so common as to attract almost no notice in the daily press. She also made one notable voyage to North and South America, including a visit to Lakehurst, in June. This airship has now crossed the Atlantic Ocean seven times, and the Pacific once. Her South American voyage demonstrated the ability of a well-found and handled airship to encounter successfully the torrential rains of the equatorial region.

The British airship *R-100* made a round-trip voyage from England to Canada and back in July and August. She remained in Canada about two weeks; and during that time was based on the mooring mast at St. Hubert, Montreal, without the protection of a hangar. While coming up the St. Lawrence valley, the *R-100* was struck by a severe squall which partially ripped the fabric from both her horizontal fins. The accident was strikingly similar to that which happened to the *Graf Zeppelin* on her first voyage to America in October, 1928. The lessons of the two accidents were the same: a modern rigid airship can continue her voyage, even with badly damaged fins; and the fabric covering of the fins should be supported at closer intervals than had become standard practice in the slower airships of an earlier day. In neither the *Graf Zeppelin* nor the *R-100* did any weakness develop in the primary structure of the fins or hull. Repairing the damaged fins while the airship was riding to the high mooring mast at Montreal was a praiseworthy accomplishment.

This report is written immediately after the crash of the *R-101*. The thorough investigation into this disaster, which will undoubtedly be made by the British Government, may reveal some causes unsuspected at present; but it appears to be fairly obvious that the crash was the result of flying too low in bad weather. The low altitude was probably forced upon the commanding officer by the ship's being nearly 100 per cent full of gas, so that her pressure height was very low; and because of insufficiency of ballast, the altitude could not be increased without dropping fuel and valving gas. Owing to the small fuel capacity of the *R-101*, such procedure might have necessitated abandoning the attempt to reach India, and returning to Cardington. As many another commander would do, he chose to take a risk, rather than to play safe and admit failure of the task assigned to him. If the ship had been inflated with helium there would have been no fire, and probably no loss of life.

The moral of the catastrophe is that airships should be inflated with helium, should not be so filled with gas that their pressure height is less than a thousand feet above the terrain which they are crossing, and should have plenty of disposable ballast. All of these operating features are standard practice in the operation of the U.S.S. *Los Angeles*, which is the only rigid airship now in commission in America. The disaster points also to the great gain in safety resulting from the use of apparatus to recover water ballast from the engine exhaust. This is a feature which has been used only in the helium-filled airships of the United States.

Rapid progress has been made during the year in erecting the Navy's new rigid airship, the U.S.S. *Akron*, of 6,500,000 cu. ft. gas capacity. Already (October, 1930) over 500 ft. of the framing has been erected, and the vast structure is most imposing in appearance. In June, a successful test was made in inflating a gas cell in the midship main bay of the ship. This one cell has a volume of nearly a million cubic feet.

The Navy Department's Board of Inspection and Survey conducted a thorough material inspection of the U.S.S. *Los Angeles* in the spring of this year. The *Los Angeles* is now over six years old; she has been under way over 2500 hours, and has flown in that time over 110,000 air-miles. She has also operated about 1200 hours at mooring masts. No other rigid airship has had such a long and continuous useful life. The Board found the present condition of the *Los Angeles* to be good; and found no reason why she should not be continued in service from two to four years longer, provided she is operated in the same careful manner as in the past, with frequent inspections, and renewals of worn or deteriorated parts when necessary.

The Board recommended that when the *Los Angeles* is no longer fit for service she should be subjected to destruction tests in ground handling in heavy weather. Such tests would be of great service to the arts of airship design and operation.

American Airways Practice in 1930[7]

For the purpose of surrounding air lines engaged in the scheduled transportation of passengers in interstate commerce with all possible safeguards and with a view of providing air transportation with virtually the same uniformity of operation as is now obtained by other major transportation systems, the Department of Commerce has revised its Air Commerce Regulations to include a certificate of authority to operate such air transportation service, these regulations constituting a code of minimum requirements to govern the operation of the scheduled interstate air passenger routes. The issuance of a certificate of authority will assure the public that aircraft are reliable and suitable when provided with instruments and equipment properly adaptable to the nature of the service and conditions involved, together with an adequate number of qualified airmen fully competent in all phases of the particular operation. It furthermore assures the proper maintenance of aircraft, including engines and equipment, to the highest degree of operating efficiency, with inspection, repair, and overhaul at proper intervals. Under the certificate of authority, the air-transport line must operate over airways or routes provided with proper air-navigation facilities in the interest of safe and reliable operation with a proper

[7] Prepared by F. C. Hingsburg, Chief Engineer, Airways Division, U. S. Department of Commerce, Washington, D. C.

ground organization to administer efficiently all phases of ground operation. Each scheduled flight shall be authorized, delayed, suspended, or canceled by competent officials designated by the holder of the certificate of authority. Uniform instructions pertaining to the operations of the line will be issued, and these instructions will be made conveniently available for the information of all passengers by the holder of the certificate.

The principal airports along the routes are provided and operated by municipalities or commercial concerns. Under the Air Commerce Act, the airports are rated on general facilities, size of effective landing area, and night-flying equipment. An A-1-A rating indicates an adequate airport equipped with full facilities of the highest standard. Airports rated D-4-D or better will meet the minimum requirements set up for intermediate or emergency landing fields. The Airport Construction Bulletin and Airport Rating Regulations of the Department of Commerce cover the minimum requirements of airports. To meet the demand for a more powerful and distinctive beacon for airports, the Aeronautics Branch has designed a new type of 36-in. beacon. This design is based on optical principles which have been found satisfactory for aeronautical beacons and lighthouses, and shows alternate green and white flashes at 5-second intervals, using 1000-watt aviation lamps. The candlepower of the new unit is about two and one-half times that of the present 24-in. revolving beacon using the same size and type of lamps. The beacon is double ended, using but one standard 1000-watt airways lamp with lamp exchanger. The doublets are 15 in. in diameter and the optical lenses 36 in. in diameter, made in 13 pieces. The center bull's-eye of the lens is 20 in. in diameter. The beams of light are elevated by lowering the lamp, this principle having been proved practical and efficient in the present course lights and acetylene range-light units. The simplicity of design and workmanship is the outstanding feature of this unit. The 36-in. drum is mounted on a standard base of the type used with the 24-in. revolving beacon. The vertical shaft is mounted in ball bearings with a gear wheel and clutch mounted on the shaft. The beacon is driven by a fractional-horsepower motor with a case-hardened and polished steel worm engaging the gear wheel.

Air-navigation facilities on civil airways, according to revised regulations, require landing fields approximately 50 miles apart along the route to provide suitable landing places for aircraft under conditions of stress of weather, with additional intervening fields where hazardous and mountainous terrain is encountered. Where airports are not available to serve the purpose, the Federal Government establishes and maintains intermediate landing fields. The intermediate landing fields are located as close to the air line as possible, preferably on a main highway or at least accessible to a good road to provide for an emergency transportation and for maintenance of equipment. The location should be accessible to electric power and telephone lines wherever practicable. The ideal shape for the intermediate landing field is square as it permits landing in any direction against the wind. Such fields are not available in most locations, and therefore intermediate landing fields having at least two runways forming "L," "T," or "X"-shaped fields are selected. The runways vary between 2000 to 3000 ft. in length at sea level, increasing at higher altitudes, and are not less than 500 ft. wide. The intermediate landing fields should preferably have clear approaches on all sides. It is considered that an obstruction to the approaches cuts down the effective length of the runway approximately 7 ft. for each foot of height of the obstruction. The obstructions should be adequately marked and lighted unless arrangements can be made for their removal.

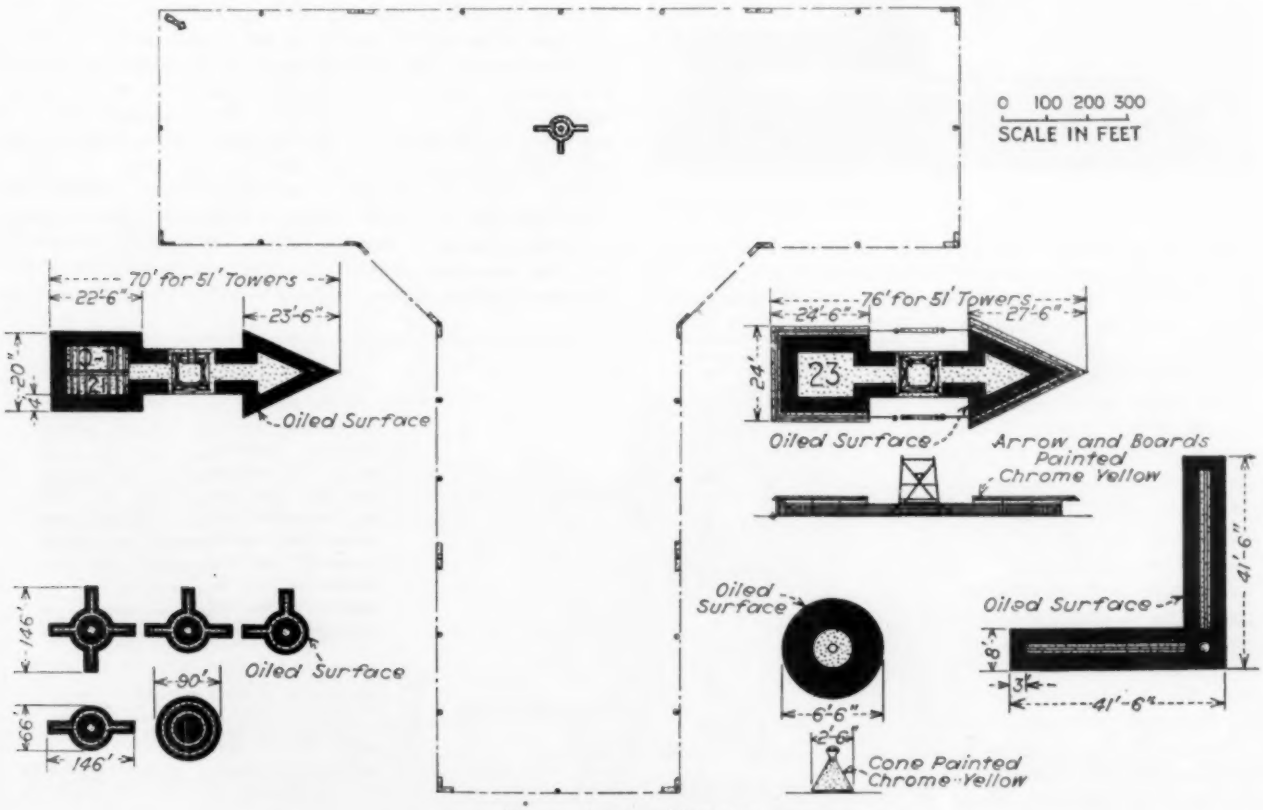


FIG. 1 AIRWAY MARKERS



FIG. 2 AIRWAY BEACON MOUNTED ON A 51-FT. SKELETON TOWER

Intermediate landing fields should be as nearly level as practicable, and of uniform surface without abrupt changes. The field should be free from ditches, rocks, or any irregularities which would be hazardous to aircraft in landing. Natural drainage is of vital importance to prevent softening of the surface in wet weather. To a limited extent, tile or artificial drainage is provided on the intermediate fields. The fields are conditioned and planted with crops of a variety suitable to the locality and growing not over 12 in. in height, the interlaced roots preventing erosion of the top soil. The crops are cut from time to time to keep the field in proper condition.

For day flying, intermediate landing fields are marked with boundary markers and a ground marker placed at the intersection of the runways as shown in Fig. 1. The boundary markers are placed on the corners and along the sides so as to outline the intermediate landing fields from the air and show the landing area. The corner and side markers consist of wooden planks attached to triangular skeleton steel frames supported by vertical posts about 30 in. high, the wooden planks being sloped so as to shed snow. They are high enough to prevent being obscured by vegetation, and are painted chrome yellow to provide maximum visibility. The ground under the marker is oiled with road oil showing a strip 8 ft. in width in contrast to the yellow color of the marker, the oil also preventing the growth of vegetation. A circle with runway indicators is placed at the intersection of the center lines of the runways, indicating the general shape of the landing strips. Where runway indicators are used in conjunction with the circle, the overall dimension of the marker is 130 ft. and the circle 50 ft. in diameter, while for square fields the circles are 90 ft. in diameter. In addition, a directional arrow is placed at the base of the airways beacon tower and points the direction of the route from south to north and west to east. Beacons are assigned mileage numbers by dropping the unit digit. The pilot can ascertain his location along the route by adding a digit to the beacon number.

A wind cone showing the direction of the ground wind is usually mounted on a bracket on the side of the beacon tower. At night, the wind cone is lighted internally, the light showing through the fabric. Intermediate landing fields are marked for use at night with a standard type of airways beacon usually mounted on a 51-ft. skeleton tower as shown in Fig. 2. The boundary of the field is marked by white lights on low boundary standards spaced 200 or 300 ft. apart and so located as to show the pattern of the field from the air at night. Green lights are mounted to show the best approaches to the landing field, and red lights in the boundary lighting circuit indicate that there is no clear approach from that side of the boundary. Red lights burning fixed are shown on all obstructions. Fifteen-watt lamps are used for white boundary lights, and 25-watt lamps for colored lights. Refractive globes are used in connection with the boundary lights to increase the effectiveness of the boundary lighting system. Metal boundary cones are used in

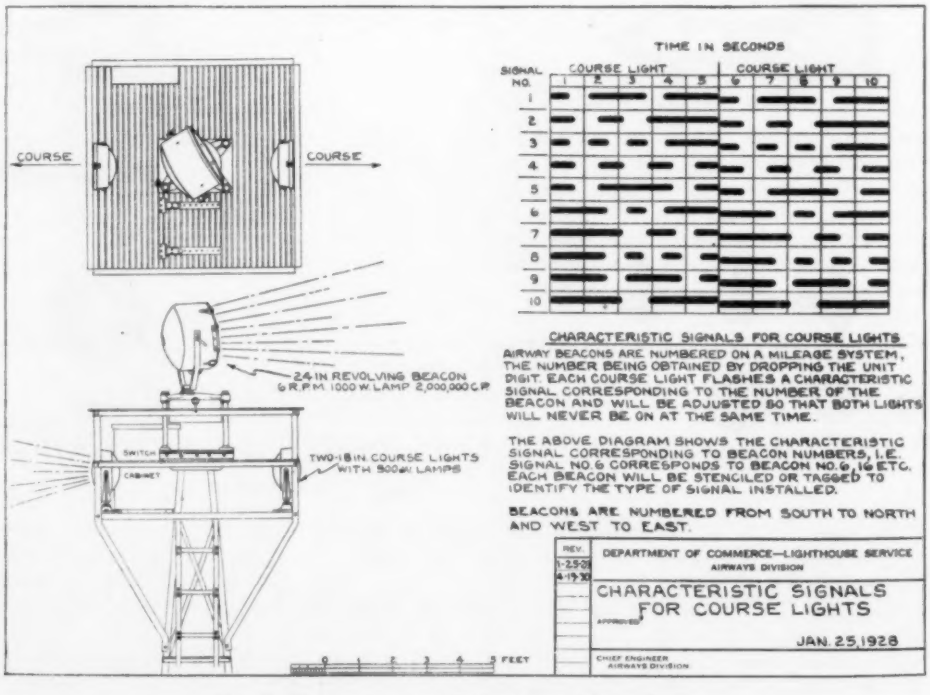


FIG. 3 CHARACTERISTIC SIGNALS FOR COURSE LIGHTS

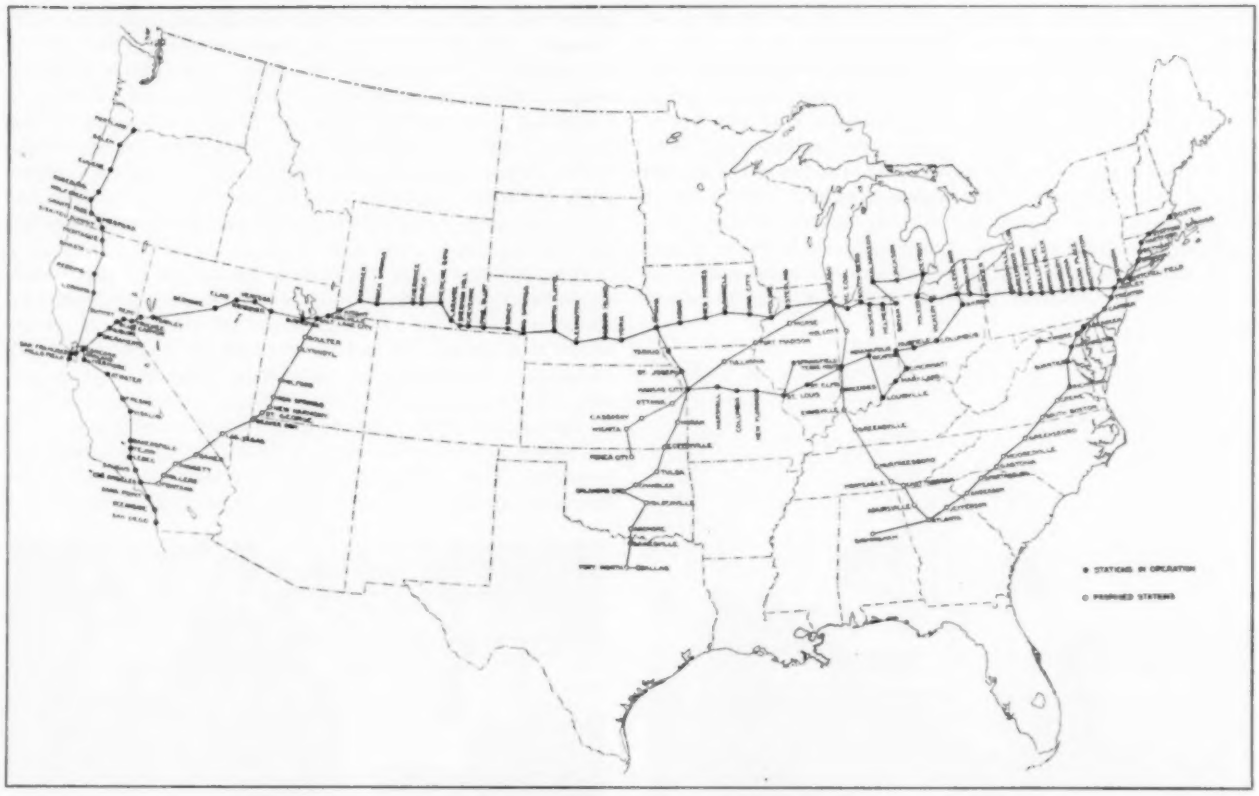


FIG. 4 AERONAUTICAL TELETYPE STATIONS, U. S. DEPARTMENT OF COMMERCE

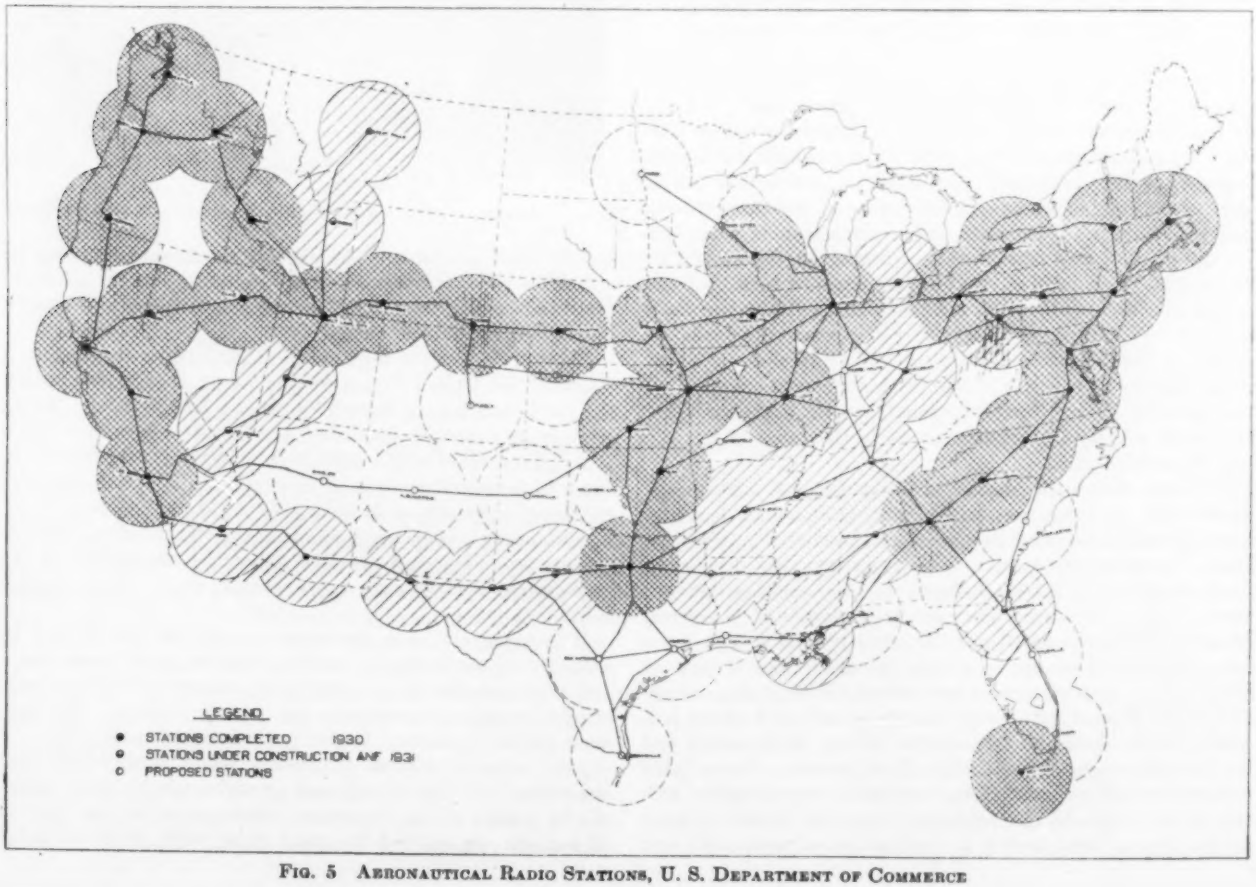


FIG. 5 AERONAUTICAL RADIO STATIONS, U. S. DEPARTMENT OF COMMERCE

conjunction with the boundary standards so as to minimize chances of collision and serve as markers for day use.

The Department of Commerce has established approximately 327 intermediate landing fields, the total area employed for landing-field purposes totaling approximately 27,000 acres. This represents an average of 70 acres per landing field. The rental paid for landing fields is over $115,000, the average price per acre being about $5. The total number of boundary lights maintained at the intermediate landing fields is about 7500, which is an average of about 24 lights for every field. There are on the average 5 red lights marking obstructions on or in the vicinity of intermediate fields, making a total of about 1500 obstruction lights.



FIG. 6 STANDARD AIRWAYS RADIO STATION OF AIRWAYS

Intermediate landing fields are available for the free use of the aeronautical public on condition that all aircraft pilots and the aeronautical public comply with the Air Commerce Regulations and the Intermediate Landing Field Rules promulgated by the Department of Commerce. Exclusive or preferential privileges are not granted, and commercialization of any intermediate landing field is not permitted. An airplane making a landing has the right of way over airplanes moving on the ground or taking off. Upon landing, the airplane must be taxied off the landing area to a location that will not interfere with or become a hazard to an airplane that may find it necessary to make an emergency landing without warning. The caretaker employed at the intermediate landing field renders reports covering all landings, and visiting ships are required to report to the caretaker to register and furnish information as to the cause of landing and the use of the field.

Airways beacons are established along the civil airways approximately 10 miles apart. The airways beacon is a 24-in. searchlight with a parabolic mirror and a 110-volt, 1000-watt Mazda lamp equipped with lamp exchanger and developing approximately 1,000,000 candlepower. The airways beacons rotate 6 r.p.m. showing a flash every 10 seconds of $^1/_{10}$-second duration, visible 20 to 40 miles in clear weather. The beam from the airways beacon is a high-intensity pencil of light of about 6 deg. beam divergence and clearly visible at close range. The cover glass is fitted with prisms to refract a small percentage of the light upward to within 55 deg. of the zenith and thus provide close-range visibility of the beacon. Course lights flashing definite mileage characteristics in synchronism with the airways beacons are projected along the course, showing to the pilot a light having a combination of short and long flashes with a luminous period of about 35 per cent and a color contrast. The beacon number characteristics of the course-light signals are shown in Fig. 3. Red course lights are used on beacons located where landings cannot be made, and green course lights are used at the intermediate landing fields. This system of lighting combines in the airways beacon all the desirable characteristics of high-intensity quick flashes for off-airways indication, and also long flash periods developing the full luminous value of the light source for on-course indication together with color contrast of light.

About 60 per cent of the airways beacons are operated automatically on commercial power, this source of supply being used wherever obtainable. The lights are turned on by a motor-wound astronomic clock, and burn from sunset to sunrise. Lamp exchangers are provided for all airways beacons so as to substitute a spare lamp in the focus of the reflector should the lamp in service burn out. Caretakers are hired at all beacon sites and paid a nominal consideration to look after the equipment and property. The principal work of servicing beacons is performed by an airways mechanician, who visits each site approxi-



FIG. 7 INTERIOR OF AIRWAYS RADIO STATION, CLEVELAND, OHIO

mately twice a month. The airways mechanicians use motor trucks for transportation which are equipped with all necessary tools and spare parts to make emergency repairs and replace burned-out and broken parts.

There are about 156 private aeronautical lights established in cities in the United States serving as landmarks from which the pilots can take a bearing to reach a landing field. Where a revolving beacon is established as a private aeronautical light, it is supplemented with a fixed 24-in. directional projector showing a high-intensity pencil of light pointed in the direction of the nearest airport or landing field to show the pilot where a safe landing may be made should he fly over the city. Private aeronautical lights are operated under the certification of the Department of Commerce as provided for in the Air Commerce Act.

It is important that obstructions along the routes such as radio towers, smokestacks, and those such as power-transmission and telephone lines in the vicinity of airports and landing fields should be marked to minimize the hazards to flying. For daytime visibility, skeleton towers are painted alternate bands of chrome yellow and black, the chrome yellow bands being twice the width of the black bands and approximately $^1/_7$ of the height of the height of the structure. Obstructions in the vicinity of airports are marked for night flying with fixed red lights

mounted at the height of the obstruction, using 100-watt lamps in high-transmission red waterproof globes. Radio towers are marked in the same manner, but in addition have fixed red lights consisting of 50-watt lamps in waterproof globes mounted on the diagonal corners at their $^1/_3$ and $^2/_3$ height so as to be visible from any angle of approach.

The supporting towers of transmission lines crossing navigable waters are painted as hereinbefore mentioned for daytime visibility, and are marked with red lights consisting of two 200-watt lamps mounted in 300-MM fresnel lanterns with red color screens visible in all directions. In addition red lights consisting of 100-watt lamps are mounted in waterproof globes so as to envelop an outline of the area over which flying is a hazard. The projected beams of lights are elevated so as to intersect at the height of the obstruction to be cleared. One or more red electric-code beacons showing flashing lights are installed on the corners of the hazardous area so as to give the pilot the proper warning at long range.

Weather is one of the uncontrollable factors that affect the reliability and safety of air transportation, and therefore an adequate weather-service and communication system in addition to other aids to navigation along the airways is essential for maintaining dependable schedules. Current weather information is collected on an hourly basis from the principal

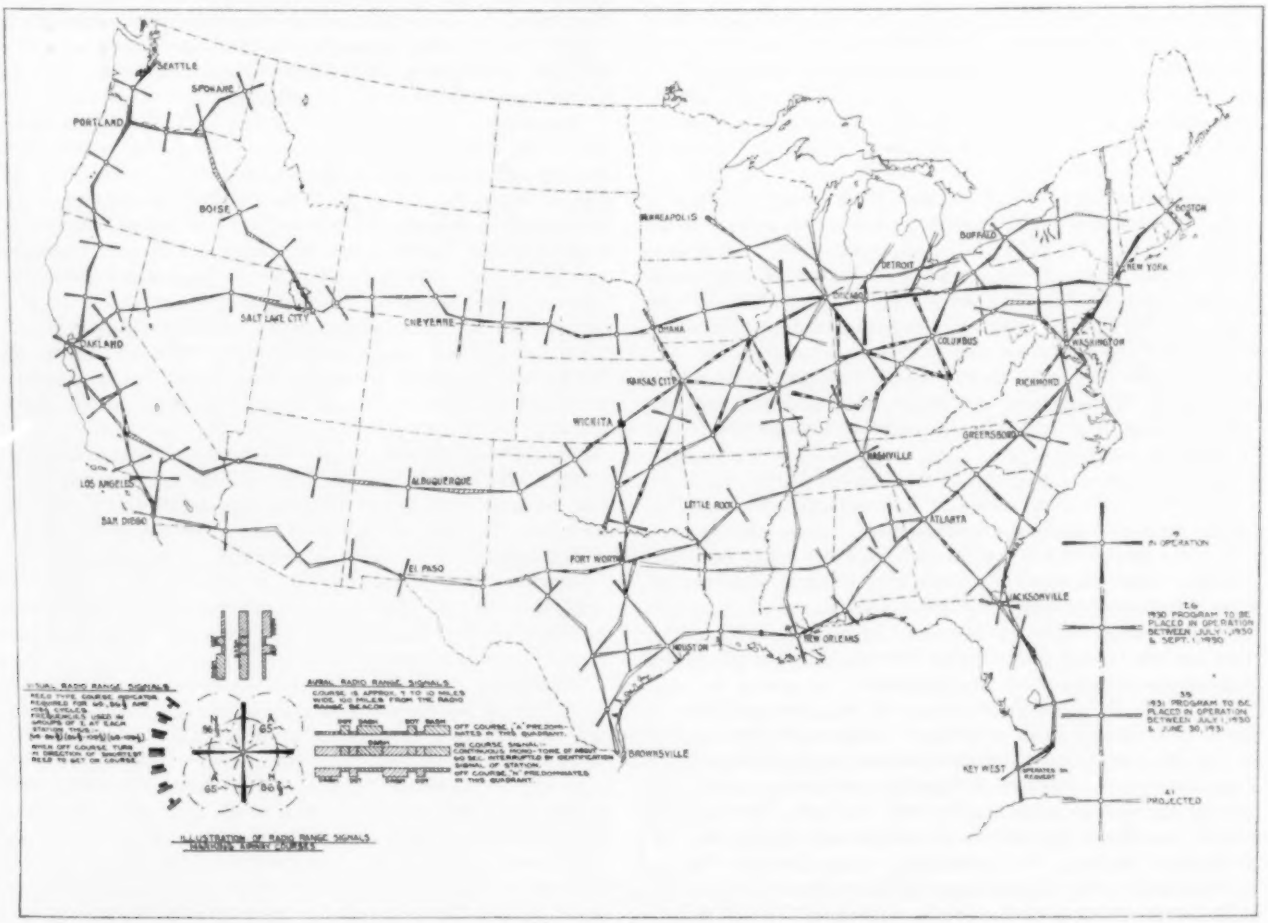


FIG. 8 AERONAUTICAL RADIO RANGE BEACON SYSTEM

at the $^1/_3$ and $^2/_3$ heights of the tower. The catenary of transmission lines crossing navigable waters is illuminated from the towers by 24-in. parabolic directional projectors showing a light of sufficient intensity to mark the crossing by the indirect illumination of the beam. Bridges are marked with 300 MM code lanterns having two 200-watt lamps mounted on the tops of the towers. For marking areas which may present a hazard to flying along a civil airway, a 24-in. standard revolving beacon equipped with a 1000-watt lamp and lamp exchanger and with red cover glass is used at a prominent point to indicate the existence of the hazardous area. Pilots are warned to give red revolving beacons a wide berth or fly over the area marked by such beacon at an altitude not less than 1000 ft. An alternative method of marking the hazardous flying area is the use of a high-intensity fixed projector pointed along the boundary of the area

intermediate landing fields and critical points along the route. Teletype communications are used in collecting weather information and for making aircraft-position reports and keeping track of airplanes flying over the route. The teletype system now in operation and the weather-reporting points are shown on the map in Fig. 4. Weather information is collected in sequence from each reporting station at the scheduled period set for each circuit, the collection taking from four to five minutes. At terminal points where a number of airways converge, the collections from the respective circuits are made just prior to the quarter-hour periods. There are at present 5650 miles of teletype in operation along the civil airways, and this mileage will be increased to 8600 miles during the next year. The teletype service utilizes a leased telephone circuit over which signals are transmitted by an automatic typewriter, leaving a type-

written record on all the teletype machines in the circuit. The transmission is at a rate of 40 words per minute. The weather information collected along the route is communicated by teletype to the Weather Bureau for interpretation and used in connection with weather maps. Weather information is disseminated by radio to airplanes in flight and to airports equipped with receiving apparatus. There are at present 35 airways radio stations in operation as shown on the map in Fig. 5. When the airways radio stations are completed in accordance with the program, there will be no area in any part the United States where flying takes place that will not be equipped with proper weather service, which may be received by the pilot in the air by tuning in.

In addition the Weather Bureau collects weather information from the primary net twice daily and from the secondary net every three hours, from which data weather maps are prepared for the interpretation of weather and the planning of schedules. A special broadcast is made every three hours for the use of airports covering the secondary net area. Upper air conditions are broadcast each 6 hours.

The standard airways radio station of the Airways Division is shown in Figs. 6 and 7. The equipment of the airways radio station consists of a 2-kw. intermediate-frequency transmitter for broadcasting weather information and messages to airplanes for the safety of flight on frequencies between 237 and 350 kilocycles (kc.) This band is divided into channels with 6 kc. separation, and the same frequencies are shared by stations approximately 700 miles apart. In addition to the broadcast transmitter, the station is equipped with a 500-watt high-frequency transmitter for point-to-point communications. This transmitter can be operated from a gasoline-driven generator in the event the station is cut off from power due to the failure of electrical transmission lines. The airways radio stations are operated in conjunction with the radio range-beacon service now being established along the airways sharing the same frequencies. The radio range beacon is stopped on the quarter-hour periods and identified by station announcement followed by correct time and voice broadcast of weather information. This concise statement of weather information along any sector of the route occupies about two minutes.

The system of radio range beacons to mark the civil airways in the United States is shown in Fig. 8. Nine radio range beacons of the aural type have been in operation during the past two years, and the 26 additional beacons now under construction will be placed in operation early next summer. The program for the fiscal year 1931 covers the installation of 33 additional radio range beacons. Consideration is being given to the use of visual-type radio range beacons for the 1931 program. Seven radio range beacons of the visual type have been manufactured by the Airways Division at the Detroit shops and are now being flight tested. The transmitter for the radio range beacon is a 2-kw. one and transmits signals on two-directional loops. The interlocking along the equisignal zone marks the airways course and coincides with the lighted route. The most important function of the radio range beacon is its homing feature, and therefore the radio range beacons are installed in the vicinity of landing fields so as to direct the pilot to a safe landing field under conditions of poor visibility.

Marker beacons are located at the intersection of two radio range courses and transmit identification signals on dual frequencies of the adjacent radio beacons. The marker beacons serve as mile posts along the route. The marker-beacon transmitters are equipped for transmitting voice in addition to the characteristic signals, and are used in acknowledging messages from airplanes and reporting their flight over the intermediate fields. The position reports from airplanes passing over the principal intermediate fields are transmitted by the airways keepers on the teletype to the terminals.

The airways keepers stationed at the principal intermediate fields from which weather reports and "plane over" reports are received will be equipped with radio receivers to stand a loud-speaker watch on the national calling and distress frequency for emergency messages from aircraft. The national frequency adopted in the United States for calling and distress messages from aircraft is 3106 kc. The airways keepers are instructed to keep a sharp lookout for airplanes flying over the route and transmit on the teletype the "plane over" movements of aircraft that are not equipped with radio. Airways keepers set flares for the landing of aircraft under conditions of stress of weather when instructed by a proper responsible official of the operating company.

In the United States a national frequency of 278 kc. has been set up for the use of airports to communicate to aircraft in flight. The power is limited to 10 watts and transmits to the airplane by voice. The airports are requested to be equipped with receiving sets having a loud speaker tuned to the national calling frequency of 3106 kc. for receiving messages from aircraft, thus permitting the airport manager to communicate with airplanes for such landing instructions or emergency communications as may be found necessary.

The Federal Radio Commission has adopted an aviation plan for the use of the air-transport companies so as to provide two-way radio communications. A number of frequencies in the 1500-6000-kc. band have been reserved for aviation purposes to be used by aeronautical or aircraft stations on a chain basis for communication purposes. Under this plan the air-transport operators will establish ground stations along the routes on which scheduled operations take place and keep a constant contact with aircraft while in flight. This system permits the air-transport operators to provide for themselves a two-way system of radio communication for safeguarding the lives of passengers and maintaining proper schedules.

Trend in Aircraft-Engine Development

By HAROLD CAMINEZ,[1] INDIANAPOLIS, IND.

It is difficult to ascertain the extent to which various improvements in aircraft engines will be adopted, and it is unwise to predict what the eventual types of aircraft power plants will be. However, a discussion of present-day aircraft engines with their possibilities and shortcomings is helpful in establishing some of the trends in engine development.

The author discusses general features of aircraft engines, such as in-line versus radial types and water-cooled versus air-cooled types, and shows their limitations and advantages.

THE TREND of aircraft-engine development is a subject over which there is always considerable controversy. Each engine manufacturer likes to believe that his developments are along present tendencies and that his latest product demonstrates the latest trend.

All developments in engines are for the purpose of eventually meeting the requirements of the aircraft builders. These requirements continually grow more exacting, and often necessitate changes in basic engine designs. In studying the trend in engine construction, improvements must be reviewed that will be made in aircraft and which will influence engine requirements.

To obtain any great increase in the market for commercial aircraft, airplanes must be made safer, more comfortable, more useful, and more economical. Improved safety requires not only increased engine reliability, but more especially superior airplane performance. As demonstrated in the recent Guggenheim safe-aircraft competition, airplanes will have lower minimum flying speeds, higher maximum speeds, and much steeper angles of climb. Such performance characteristics make flying safer and more simple, and will encourage more widespread use of airplanes. However, they introduce additional problems for the engine designers, since with such airplanes there will be steep climbing angles at full engine throttle with low air-speed velocity, and it will be necessary to cool the engine satisfactorily under such conditions. High propeller efficiency, not only at maximum speed, but also during climb, will be important in these airplanes, so that the need for low propeller r.p.m. will be more pronounced. These more difficult cooling conditions will have to be provided for, and lower propeller speeds will have to be obtained with no material increase in present total power-plant weight. These requirements will tax the ingenuity of the designer.

Increased comfort during flight is essential before air travel becomes truly popular with the general public. This necessitates the reduction of the noise and the vibration that are so prevalent in present-day airplanes. The worse offenses as regards noise are due to the propeller and to the engine exhaust. The requirements of comfort seem to suggest the necessity of propeller reduction gea ng to obtain quieter slow-turning propellers, together with the f efficient exhaust silencers to reduce exhaust noises. Whe these noises are diminished, it may also be necessary to investigate means of reducing some of the other engine noises, especially those due to high piston side clearances and to too large variations in valve-tappet clearances. Together with the reduction of noise, a serious investigation will be necessary toward the complete elimination of vibration, for not only is vibration disquieting to the nerves, but it is also a contributing cause to numerous airplane and engine failures. Elimination of vibration will mean not only complete inertia balance of the engine, but also low engine-torque variation throughout the cycle, and the complete elimination of all critical speeds of crankshafts and other fast-moving parts. It will also be necessary to get this quietness of operation and freedom of vibration without any increase in engine weight. There must be considerable divergence of opinion as to the best method to obtain these ends.

Greater Pay Loads Needed

The commercial usefulness of aviation will be increased when higher flying speeds are obtained along with increased flying radius and greater operating economy. Economy will require carrying greater pay loads without increasing power-plant sizes, as well as a reduction in engine cost and in fuel consumption. Reduced fuel consumption will also result in increased flying radius. Therefore higher flying speeds with increased radius and economy will be obtained not merely by increasing the power in a given airplane, but by refinements in design and aerodynamic qualities. This will result in more careful consideration of the parasite drag or head resistance of the power plant and cooling system, as well as in improvements of propulsive efficiency. The air-transport operator will also seriously consider the economy in operation that will be effected by longer engine life and lower maintenance cost. It is possible that this increased engine life and lower maintenance cost may be obtained by improving airplane performance to such an extent that reasonably large pay loads can be carried with satisfactory performance when operating the engine at only part throttle for most all flying conditions. However, it is more reasonable to expect that the engine builder will so improve his product that adequate life and reliability will be obtained at full-throttle as well as at part-throttle engine operation. In small airplanes operated by private fliers it is also essential that adequate reliability be obtained under all conditions of operation. It is difficult to visualize any real widespread popularity of private flying as long as there are any "don'ts" in connection with engine operation. Signs on airplane instrument boards saying "do not climb at air speeds less than—m.p.h." and instructions in engine books saying "do not run engine unnecessarily at full throttle" must be eliminated. One of the joys of flying is the absence of traffic and the needlessness of speed laws. There should be no engine requirements that will diminish the full appreciation of these joys.

---

[1] Engineer in Charge, Aircraft Engine Department, Allison Engineering Company. Mr. Caminez was educated at Cornell University, where he received the degree of M.E. in 1920. He was then employed for five years in the Engine Design Department of the Engineering Division of the U. S. Army Air Corps. In 1925 he became Chief Engineer of the Fairchild Caminez Engine Company, which company specialized in the development of aircraft engines. He resigned in 1928, and was then engaged in making an investigation of the aircraft-engine industry both in this country and abroad. In 1929 he was employed as Design Engineer by the General Motors Corporation, and later in his present position.

Presented at the Fourth National Aeronautic Meeting, Dayton, Ohio, May 19 to 22, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

The requirements of the commercial aircraft engine have often been discussed. However, it is now seen that in addition to the well-recognized necessity of low engine weight per horsepower, together with adequate engine reliability and reduced engine cost, careful consideration must be given to improving present commercial aircraft engines in the following respects: (1) Ability to cool adequately at full throttle under the low airspeed conditions that are obtained in climb with airplanes that have high-speed ranges, together with provisions for the regulation of engine cooling under the more extreme flight conditions that will be obtained with these airplanes; (2) lower head resistance of power plant and cooling system at maximum flying speeds; (3) more efficient and quieter propeller speeds, and therefore the more widespread adoption of propeller reduction gearing; (4) more general use of exhaust silencers as well as quieter running engines; (5) complete balancing of all engine inertia forces, low cyclic variation in engine torque, and elimination of critical shaft speeds and other vibrating tendencies; and (6) better full-throttle engine operation in actual aircraft installations particularly in regard to increased engine life, improved engine reliability, lower maintenance costs, and reduced fuel consumption.

It would be difficult to ascertain all the various experiments to improve present aircraft engines that are going on all over the world, as well as it would be unwise definitely to predict the eventual types of aircraft power plants. However, a discussion of present-day aircraft engines, together with the possibilities and shortcomings of the various types as now revealed in service, would be helpful in establishing some of the trends in engine development.

Air-Cooled Versus Water-Cooled Engines

In this country the majority of the engine manufacturers are building and further developing radial air-cooled engines. A few companies have recently started in the production and development of in-line air-cooled engines. One manufacturer is in regular production on water-cooled aviation engines. In England, where the modern air-cooled radial engine was first developed, there are two large companies regularly producing this type, while two other equally large aircraft-engine manufacturers are still confining their efforts to water-cooled engines. Two more recent English engine builders are now in production on small in-line air-cooled engines. In France and in Germany there appears to be equal effort spent in the development of both water-cooled in-line and air-cooled radial engines, while in Italy the only engines of merit are of the water-cooled type. Practically all engines now in use operate on the conventional four-stroke Otto cycle, but there are significant developments under way of engines operating on the Diesel cycle. In this country air-cooled and water-cooled Diesel aircraft engines are being experimented with, while in Europe several water-cooled Diesel engines are under development. Therefore, to make the discussion of present-day engine trends complete, developments must be considered in: (1) The radial air-cooled engine; (2) the in-line air-cooled engine; (3) water- or fluid-cooled engines; and (4) aircraft Diesel engines.

The radial engine will be considered first because it is the type now most extensively manufactured and used in this country. This type of engine was initially developed for military purposes because: Firstly, air-cooling made it less vulnerable to gunfire; secondly, the radial-engine arrangement gave a short engine length with concentration of mass, which is a desirable characteristic in combat airplanes designed for acrobatic maneuvers; and finally, the radial air-cooled power plant could be built at a lower installed weight per horsepower than the water-cooled power plant that it replaced. The first two advantageous characteristics of the radial engine from a military viewpoint have little if any influence on the commercial use of this type. The principal advantage of this type that must therefore be considered is that its development for military use has produced a power plant of low weight per horsepower. This is a most desirable engine characteristic if it is accompanied by engine life and reliability.

According to reports emanating from the Navy Department, which is a most ardent advocate of the air-cooled radial, long life and reliability are obtained by such engines in its airplane installations. However, commercial operators of these same engines do not apparently have equal satisfaction. The reason for this is at once evident to one experienced in the developments of air-cooled engines. The power output which such an engine will maintain without sacrifice of engine life and reliability depends upon the engine cooling. The cooling effect on an air-cooled engine is a function of the air velocity past the engine, which in turn depends on the speed of the airplane and the slipstream effect of the propeller on the engine. Therefore, every air-cooled engine developing its rated power has a certain minimum airspeed velocity which cannot be lessened without sacrifice of engine life and reliability.

Military Planes Have Good Cooling Conditions

If an air-cooled engine is designed for maximum performance in military use, it must be designed for the airspeed characteristics met in this service. These military planes, with power loadings of less than 10 lb. per hp. and with high speeds in excess of 175 m.p.h., have vastly superior cooling conditions for air-cooled engines than a modern economical commercial type airplane. An air-cooled engine, to be an efficient and reliable power plant, must have different cooling facilities, and must therefore be differently designed for every new type of airplane installation. When a military air-cooled engine designed for an airplane with a high speed of 175 m.p.h. and a power loading of 10 lb. per hp. is used in a commercial airplane with a high speed of 135 m.p.h. and a power loading of 15 lb. per hp., it should be expected that the rated engine horsepower would be greatly reduced for the engine of the commercial plane to obtain the same reliability that the engine has in the military airplane.

Partly to make up for the cooling deficiencies in the commercial airplane the carburetor mixtures can be enriched and the fuel consumption increased, thereby partly cooling the engine with excess fuel. Increasing the oil circulation and the oil consumption also helps in engine cooling. It is also possible to design the propeller section near the hub so that it throws a maximum slipstream velocity on the engine. Such a propeller characteristic is particularly advantageous during airplane climb where the engine operates at full throttle with low forward speed of the plane. Apparently increasing the slipstream on the engine decreases overall propeller efficiency, but it will undoubtedly be necessary to do this with air-cooled engines installed in airplanes that have slow climbing speeds.

Cowling May Effect Engine Cooling

One of the greatest problems encountered in the commercial installation of air-cooled engines is that an airplane builder by the choice of cowling and propeller can greatly improve airplane performance, but by doing so he may seriously affect engine cooling. Increasing airplane performance at a sacrifice of engine cooling and reliability is highly undesirable. Therefore, the only way to be sure of reliable engine operation with sufficient engine cooling in a new air-cooled engine installation is to run extensive cooling tests on the engine under all conditions of flight. This generally entails quite an elaborate test set-up,

because air-cooled engine temperatures vary among different cylinders and on different places on the same cylinder.

Another difficulty is the lack of general authentic data on the desirable maximum temperatures that can be employed. The author's experience has been that for reliable operation cylinder barrels should be kept below 300 deg. fahr. and cylinder heads below 600 deg. fahr. on the type of air-cooled cylinders now generally employed. When this cylinder-barrel temperature is exceeded, piston seizure may occur, and when the head temperature is exceeded, detonation and general engine overheating may ensue. There are times when engines have apparently operated satisfactorily with temperatures 50 deg. to 100 deg. higher than that stated.

Cooling tests on new air-cooled engine installations should be run with all standard cowling, exhaust manifolds, and carburetor settings. Engine temperatures should be determined in full throttle climb and level flight with the weakest carburetor mixtures and poorest fuels that will be used in service. It should also be considered that in extended service poor seating of the valves, bad spark plugs, and tight pistons may develop and that these also increase cylinder heating, so an additional margin of cooling should be allowed for these contingencies in an air-cooled installation in which reliability is aimed at. The importance of such engine-cooling tests cannot be overestimated and it may be desirable for the Department of Commerce to insist upon such tests in its regulations. When giving information on approved air-cooled engines, the cooling-air velocity and temperature, the grade of fuel, and the fuel consumption during the type tests should be published with the engine rating.

In multi-engine airplanes using air-cooled power plants, which are designed to fly safely when one of the engines is not operating, cooling tests are essential under such flying conditions. When one engine is out, it is generally essential to fly with the remaining power plants at full throttle. The airspeed of the plane is reduced under such conditions, and the cooling effect on the operating engines thereby reduced. Cooling tests on an air-cooled multi-engined airplane should show an adequate cooling reserve under such operating conditions if any increased airplane reliability is to be expected from the use of multiple power plants.

Cast-On Fins Being Used

Since engine cooling is a limiting factor in air-cooled engine development, continued progress should be expected along this line. The same type of air-cooled cylinder construction using steel-finned barrels with attached cast-aluminum heads has now been generally employed with but minor modification for the last eight years. It is known that better cylinder-barrel cooling than that obtained with steel fins will allow the development of greater engine-power output without requiring increased air-blast velocities. In France, Germany, and Czechoslovakia, air-cooled engine manufacturers are now using steel cylinder barrels with cast-on aluminum fins. Similar cylinder constructions were previously employed in this country with limited success. One serious difficulty formerly experienced was due to the large difference in coefficient of expansion between the aluminum and the steel which allowed the aluminum fins to expand away from the steel barrel when the cylinder heated up, thereby destroying the intimate contact required for satisfactory cooling. Recent metallurgical developments have shown the way to obtain special iron alloys suitable for cylinder-liner requirements that have expansion coefficients about equal to that of aluminum. It is therefore probable that satisfactory cast-aluminum cylinders with special iron or steel liners may be developed soon which will take full advantage of the higher heat conductivity, higher specific heat, and ability to get more cooling area per pound of metal than is possible with steel cooling fins.

Other methods of increasing cylinder cooling under critical cooling conditions, such as are obtained during climb and take-off in a commercial airplane with a heavy power loading, may also be developed. Placing the carburetor float chamber in front of the main discharge nozzles so that the carburetor mixture will automatically be enriched when the engine nose is pointing up would help cylinder cooling during climb without affecting fuel consumption in straight-away flight. Another method of obtaining added cooling is to arrange an auxiliary oil line for the purpose of spraying oil into the engine upon the inside of the pistons. This auxiliary oil system could be so arranged that it would operate only at full throttle when the engine nose was pointed up. The cooling effect of an increased oil circulation could thereby be employed without increasing oil consumption during level flight.

Improved Fuel Metering Desirable

Lean intake mixtures increase the amount of heat to be dissipated in the cylinder. The cooling characteristics of the engine determine the leanest mixture strength that it is advisable to employ. Where there is any unevenness in distribution, the carburetor must be set rich enough so that the cylinder receiving the leanest mixture will not overheat. Improvements in fuel metering and distributing systems are therefore highly desirable in air-cooled radial engines. It is particularly advantageous in these engines to get proper fuel distribution without heating the air-intake charge. Heating the air charge makes it necessary for the cylinder cooling fins to dissipate more heat per horsepower developed by the engine, both because of the added heat to the charge and also because it decreases the maximum allowable compression ratio. Cold carburetion methods should therefore be particularly attractive for these engines if they can assure perfect fuel distribution. Metering the fuel by means of separate fuel pumps for individual cylinders and injecting directly into the cylinder or intake port is a method now being developed for obtaining exact fuel metering and distribution without applying heat to the air-intake charge.

Means for controlling cylinder-barrel temperatures is probably most important for improving the life and reliability of air-cooled engines. Cold cylinder barrels increase engine friction and promote wear due to increase of oil dilution by gasoline. Overheated cylinder barrels not only increase wear by destroying cylinder lubrication but also are the general cause of piston seizure. To regulate cylinder-barrel temperatures, it is necessary first to determine them. The gaging of cylinder-barrel temperatures by oil temperature is neither exact nor reliable. This has led to the installing of one or more thermocouples on cylinder barrels in many airplane installations. Potentiometers or milliammeters used with these thermocouples are delicate instruments subject to numerous inaccuracies, and these installations show the temperatures of the cylinder at only a relatively few places.

Painting for Cylinder Barrels

As previously pointed out, temperatures vary considerably at different points on the different cylinder barrels. It has therefore been suggested that the cylinder barrels be painted or surfaced with some material which will show by change of color or some other effect when the safe temperature is exceeded. Paints which change color when the surface to which they are applied exceeds a certain temperature, and which regain their original color when the surface cools, have been developed. The general use of such a paint on cylinder barrels would simplify the conducting of cooling tests on air-cooled engines. By taking advantage of the temperature gradient that

exists along the length of the cooling fin, and by employing a paint that changes color at the mean temperature which it is desired to maintain, cylinder-barrel temperatures could be judged by observing the extent of change in the color of the paint.

A number of improvements that may be effected in air-cooled engines, particularly in regards to cylinder cooling, have been outlined. Commercial airplanes that climb at steep angles and low air speeds, that use propeller reduction gearing with large propellers and resultant low slipstream velocity on the engine, that use exhaust silencers with resultant increase of heat to be dissipated by the cylinders, that require engines of small frontal area and low head resistance, that necessitate quiet-running engines so that piston clearances cannot be excessive, and that require reliable power plants of low weight per horsepower, impose more strenuous engine-cooling requirements than the present engines are designed for. Whether air-cooled engines can be made fully to meet these requirements depends upon the development of better cylinder-cooling characteristics. It is evident that possibilities exist for considerable improvement over the present air-cooling arrangements. That this improvement will meet the more exacting requirements imposed by airplane builders and users cannot be conceded.

Air-Cooled-Engine Progress

The in-line air-cooled engine is now receiving considerable attention in this country. Its development is due to the desire to have air-cooling, and to overcome some of the objections to the radial engine. The lesser frontal area of the in-line engine is contrasted with the starfish arrangement of the radial with its large head resistance and its interference to visibility. It is claimed that the general users are more familiar with the in-line cylinder arrangement, and will therefore have more confidence and more ability to service this type in contrast with the radial engines. The in-line construction also overcomes other objections to the radial type such as the inability to obtain perfect inertia balance in the radial engine due to the distorted piston motion caused by the link connecting-rod arrangement. This articulated connecting-rod arrangement also increases the maximum piston side thrust in the radial engine about 50 per cent over that which would normally be obtained in an in-line engine with the same connecting-rod to crank-throw ratio, so that it should be possible to operate the in-line engine with higher cylinder-barrel temperatures without danger of piston seizure. The in-line engine also allows the use of overhead cam shafts so that the push-rod valve mechanisms used in radial engines with their higher inertia of moving parts and their variable tappet clearances due to cylinder expansion can be avoided. However, the in-line air-cooled construction is inherently heavier than the radial air-cooled arrangement because wide cylinder spacing must be allowed for the cooling fins in the in-line engine. There is also considerable controversy as to whether the in-line engine arrangement will allow as good cylinder air-cooling conditions as the radial type, particularly during take-off and climb when the propeller slipstream is most potent in cooling.

As in the radial air-cooled engine, the main problem in the development of the in-line engine rests upon further development in cylinder cooling. Improved cylinder construction and better mixture distribution will help. Considerable improvement in the cooling of the in-line air-cooled engine can also be effected by improved engine cowls and scoops. However, the problem of obtaining low weight per horsepower with the in-line air-cooled engine appears to be greater than with the radial type. The commercial in-line air-cooled engine is competing at present with the radial engine mainly in the smaller sizes. Due to their lower weight per horsepower, better airplane performance should be expected with the radial engines, but the in-line engine will have its adherents due to the fact that it provides a cleaner airplane installation, it allows a simpler exhaust-disposal system, and because it is probably better understood by the general public.

Comparatively little work has been done in this country during the past few years on the development of commercial water-cooled aircraft engines. Although considerable numbers of war-time water-cooled engines are in commercial use, practically all new commercial aircraft engines are of the air-cooled type. In comparing the two engine types, it is usual to compare the modern air-cooled engine with the war-time water-cooled engine, much to the disparagement of the latter. However, in Europe where the modern air-cooled engines were initially developed, it is found that equal effort is being spent on the development of the water-cooled types. Review of aircraft-flight records shows that most of the important world records in regard to speed, duration, and distance are now again held by water-cooled engines. These performances have been established abroad, and are causing American engine designers to reconsider the water-cooled engine types.

The popularity of the air-cooled engine was initially due to the fact that this engine could be built with the same weight per horsepower as the dry weight of the water-cooled engine. The use of air-cooling eliminated the weight of the water-cooling system and also the plumbing troubles that were experienced with it. The extended use of air-cooled engines has shown that this type also has its deficiencies, namely, the difficulty of determining and controlling engine temperatures. Recent developments in water-cooled engines have now brought the installed weight per horsepower of this type complete with cooling system down to that of the air-cooled engines. This decrease in weight per horsepower has been due mainly to an increase in engine speed and in brake mean effective pressure in the water-cooled engines. Equal effort has undoubtedly been spent to increase the power output of air-cooled engines, but the designers of water-cooled engines have met with more success because their problem was less difficult. Power output in an aircraft engine is now generally limited by thermal conditions, and with fluid cooling it appears to be easier to obtain the necessary increased cooling effect.

High-Temperature Coolants Developed

Aside from the influence due to the success of the water-cooled engines abroad, particularly in the last Schneider cup races, and in the establishment of numerous distance and duration records, the greatest factor in bringing attention again to liquid cooling in this country has been the development of high-temperature coolants for engine-cooling purposes by the U. S. Army Air Corps. Water-cooling had previously limited engine temperatures to around 180 deg. fahr. in order to prevent the loss of coolant by evaporation. By substituting a high boiling-point liquid, such as Prestone, it is possible to run coolant temperatures considerably higher. The development of the air-cooled engine has shown that satisfactory engine operation can be obtained with cylinder barrels at 300 deg. fahr. By raising the coolant temperature to this, a considerable reduction is possible in the size of the engine radiator over that which would be necessary for water. This reduction in radiator size means not only a reduction in weight but also in head resistance. Despite the more efficient streamline air flow that can be obtained through a radiator as compared with that through the fins of an air-cooled engine, the head resistance of the water-cooled engine and radiator was often equal to or greater than that of the air-cooled engine. However, with high-temperature liquid cooling,

it appears possible to bring the head resistance of engine and cooling system down to a minimum.

Since high-temperature liquid cooling improves airplane performance, it is reasonable to expect that liquid-cooled engines will be developed to use the new coolants. Engines developed initially for water-cooling probably will not prove satisfactory at the higher coolant temperatures, and the troubles that will at first be experienced with the new cooling fluid may discourage many from its use. With high-temperature cooling, new engine-design problems are introduced, the most vexing being due to the greater thermal expansion caused by the higher temperature range. In cylinder constructions using steel cylinder liners and aluminum water jackets, the difference in expansion of these two materials requires special provision for expansion in the original design when high cooling temperatures are employed. Rubber connections and soft-soldered joints do not stand up well under the higher temperatures, and other practices found satisfactory with water-cooling will need revision.

The trend in liquid-cooled engines is to construct in-line engines of maximum compactness, and to increase the horsepower developed per cubic inch of piston displacement by increasing both the brake mean effective pressure and the normal engine r.p.m. Due to the higher crankshaft speed, most of these engines are now fitted with propeller reduction gearing. Satisfactory propeller reduction gearing for liquid-cooled engines has been developed, and the use of such gearing presents no engine thermal problems with the liquid-cooled engine as it does with the air-cooled types.

Since the engine must be charged with the weight of the cooling system, and since the dependability of the power plant necessitates reliable cooling systems, it is likely that the manufacturers of liquid-cooled engines will undertake to supply engines complete with cooling systems to their customers. This will result in the centralized responsibility for all engine failures and should encourage more development in cooling-system reliability and efficiency. With high-temperature cooling systems, the radiators will be relatively small compared with their water-cooled predecessors. The high-temperature cooling liquids also hold possibilities for novel radiator designs of increased efficiency and reliability that will differ considerably in construction from present water radiators.

New Bearing Materials Allow Greater Bearing Loads

The horsepower outputs of the wartime water-cooled engines were generally restricted by limitations of bearing capacity, crankshaft critical speeds, and valve-cooling difficulties. Since the development of the modern air-cooled engines means of overcoming these limitations have been determined. The new bearing materials, particularly the high-lead bronzes, allow higher bearing loads and surface velocities, so that large-diameter and narrow-width main bearings can be employed. This has materially raised the engine speed at which bearing troubles and crankshaft critical vibrations occur. Improvements in cylinder and valve design together with the development of vastly superior valve materials have minimized the difficulties experienced with valve cooling. Improvements in manifolding and the use of supercharging have greatly increased the breathing capacities of the cylinders at high speed. All this has served to increase the specific output per cubic inch of piston development. With increased experimentation and development of engine details it appears that further improvement in this respect is still possible with the water-cooled engines, while in air-cooled engines the specific power output per unit of displacement appears to be limited by cylinder cooling. A large European water-cooled engine has already been constructed that develops a horsepower per 1.2 cu. in. of piston displacement and weighs about 0.80 lb. per hp. Similar results should be obtained in this country, which will require engine operating speeds of above 3000 r.p.m. with brake mean effective pressures exceeding 175 lb. per sq. in.

In the discussion of engine trends, the relative cost of the various engine types must be considered. From a superficial study of conditions it would appear that the air-cooled engine with a direct propeller drive is considerably simpler in construction and therefore necessarily cheaper in cost than a competitive liquid-cooled engine with its additional cooling system and propeller reduction gearing. However, a careful study of present aircraft-engine costs will show that the development, engineering, and servicing costs on airplane engines represent a large percentage of their total cost, so that an economical aircraft-engine design is one which a manufacturer can tool up on with assurance of little trouble in service and relatively few future design changes. The quantity of engines produced and the freedom of service troubles are of major importance in engine cost. When aircraft and engines of better reliability and performance are built, there will be a larger market for airplanes, and the increased quantity will reflect in lower costs. Therefore, engine reliability and performance will have more weight in ultimate price reductions than mere simplifications or cheap production methods that do not improve the product.

Use of Aircraft Engines in Other Fields

Since quantity of production has a most important influence on cost, it may be pointed out that it is of considerable advantage to build a type of engine for aircraft that can be used in other automotive fields. Because of their direct cooling systems the air-cooled engines can apparently only be employed for aircraft, since no other market has yet been found for the air-cooled aircraft engines now developed. However, liquid-cooled aircraft engines have been found to be excellent power plants for marine work, and all of the water-cooled engines have been employed in such service. Therefore, by designing a liquid-cooled power plant which can also be adapted to marine use, the manufacturer obtains a larger market for his product, with the resulting manufacturing economies. The similarity in construction of liquid-cooled aircraft engines to the usual automobile power plant allows the manufacturer to take more direct advantage of the experience and facilities that have been developed for the economical production of fine automobile engines.

It is now probably universally conceded that the large airplane engines of more than 600 hp. will be of the liquid-cooled type. Few aircraft engines of such power output have yet been developed in this country, although in Europe there are a number of such engines now in regular production. There appears to be no doubt that it will be necessary to equip future large transport airplanes with such power plants, and it should therefore be expected at this time that a more intense development of large liquid-cooled engines will take place in this country. These large engines will probably be of the V or W type of 12 or more cylinders with horsepower ratings of 600 to 2000. With such large power plants, it is now relatively easy to get weight-per-horsepower ratios of 1.5 to 2 lb. per hp. Lower weight than this is not as important for commercial aircraft use as improvements in fuel consumption, maintenance, and engine life. High-powered engines developed mainly for military purposes may not be entirely suitable for commercial service, with the result that there may be developments of two distinct types of large-sized aircraft engines. The military type will be of maximum compactness and minimum weight, designed to operate supercharged at highest speeds. The commercial type will be of more generous proportions designed for a service life of at least

1000 hr. between overhauls, and to operate at fuel consumptions of about 0.45 lb. per hp-hr.

Effect of the Diesel Cycle

The development of the Diesel cycle cannot be overlooked in predicting the engine of the future. The reduced fire hazard resulting from the use of a heavier fuel, and the improved economy, particularly at part throttle, which are obtained with the Diesel cycle, are important items in its favor. The problems of the high-speed Diesel engine are difficult of solution, so that it would be unwise to state when or if ever the Diesel engine will replace the conventional Otto-cycle engine for general aviation use. The Diesel cycle in its present state of development appears to be a relatively slow-speed cycle when compared with the Otto cycle. To replace the Otto cycle for aircraft use it will be necessary to find means of obtaining high b.m.e.p. at higher r.p.m. with the Diesel cycle. Direct injection of the fuel into the cylinder as provided in the Diesel cycle makes two-cycle engine operation entirely feasible in a liquid-cooled engine. This will allow such engines to operate at relatively low crankshaft speeds with high power outputs.

It is not unreasonable to expect that the present developments on Diesel engines will lead to the development of new engine cycles for aircraft that will combine the advantageous features of both the Diesel and the Otto cycles. Fundamental researches on engine cycles and improvements in engine fuels will undoubtedly show their effects in future engine designs. The new fuels that are being developed will allow the employment of higher compression ratios in the Otto cycle with the resultant increase in power output and fuel economy. The use of fuel-injection systems with the Otto cycle together with the perfection of heavy fuels that do not readily detonate will reduce the fire hazard caused by the light fuels now necessary with carburetor fuel-metering devices in the Otto-cycle engines. Two-cycle engine operation with increased power output, but without excessive fuel consumption, are also made possible with Otto-cycle engines by the use of fuel-injection systems. The development of fuel-injection mechanisms that have resulted from Diesel-engine development may therefore be of value to our future Otto-cycle engines.

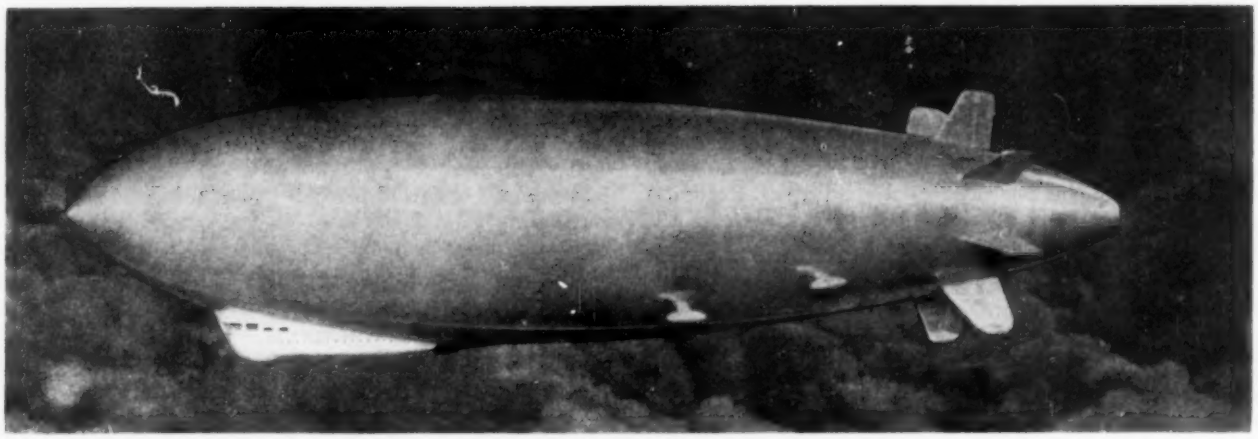


FIG. 1 PROPOSED COMMERCIAL TYPE OF METALCLAD AIRSHIP
(Gross lift 100 tons, maximum speed 100 m.p.h., volume 3,800,000 cu. ft., eight engines totaling 5000 hp., commercial range 2000 miles, pay load 50 passengers plus miscellaneous cargo of 12,500 lb. maximum.)

Some Economic Aspects of the Rigid Airship

BY CARL B. FRITSCHE,[1] DETROIT, MICH.



In this paper the author tells of the uses to which rigid airships may be put, the cost of constructing, operating, and maintaining, and the probable return on investment that can be expected. Two hypothetical airships are chosen for comparison, one of 100 tons gross lift and the other of 300 tons lift. These are compared with ocean craft and with naval vessels. In naval application it is shown that one airship will release three cruisers for other duty, that the cruisers consume more power, and are more vulnerable than the airship.

Military applications of the airship include army transport, reconnaissance, coast patrol, and many other important uses. Theoretical conditions are assumed and many problems are worked out with use of airships. Civil and commercial applications of the airship are discussed, and a comparison is made with the steamship Leviathan. Operating schedules are laid out for airships in service between New York, Philadelphia, Washington, Havana, and Panama, and performance data are given. Commercially the airship is shown supplementing present facilities, transporting passengers, mail, and express at high speed, doing more work per unit of energy, saving much time, and earning considerable profits.

DURING the first year of its demonstration commercial flights the *Graf Zeppelin* flew a total of 75,000 miles and carried 1575 pay passengers, plus a substantial load of mail and express. In addition, it has conquered the Pacific as well as the Atlantic and has flown entirely across every continent in the Northern Hemisphere.[2]

Undoubtedly such a record of performance justifies the conclusion that the utility of the airship has emerged from doubt and uncertainty to a condition of fact which challenges the attention of everyone.

ENERGY REQUIRED IS INDEX TO COST

As is always true when any new mode of transportation is born, three questions inevitably arise: (1) What is its application? (2) What will it cost? and (3) Will it pay?

With respect to the first question, in actual service three classifications naturally present themselves; they are (1) naval, (2) military, and (3) commercial.

The answer to the second question is, of course, problematical due to limited experience and to the absence of any record of a regularly operated commercial dirigible line. Therefore, in answering this question one must rely to a large extent on circumstantial evidence. By far the major operating expense in any form of transportation is the cost of the man-power plus the mechanical power required to negotiate a given journey or to accomplish a given purpose. Therefore, in this discussion an attempt will be made to compare the man-hours and horsepower-

---

[1] Vice-President, Detroit Aircraft Corp. and President of its subsidiary, Aircraft Development Corporation. Mem. A.S.M.E. Mr. Fritsche practiced civil and mining engineering until the outbreak of the World War, when he became interested in aeronautics although not actively engaged in the Air Service. In 1921 he initiated a preliminary investigation of the possibilities of all-metal construction of rigid metal airships. The next year he organized, with others, the Aircraft Development Corporation, and since that time has been engaged in lighter-than-air development exclusively, resulting in the construction and successful demonstration of the "ZMC-2" metalclad airship for the U. S. Navy. In 1925 he organized with others the National Air Transport, Inc., the first commercial air-mail transport line in the United States, and served as secretary of this company from 1925 to 1928.

Presented at the Fourth National Aeronautic Meeting, Dayton, Ohio, May 19 to 22, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] The recent flight to South America increased the commercial mileage to 93,000, pay passengers to 1730, and has added the conquest of the South Atlantic Ocean.

hours required for a given purpose with the energy required for existing modes of transport.

The answer to the third question, "Will it pay?" it is believed will follow as a natural deduction from the facts surrounding the answers to the first two questions.

For the sake of brevity, it is assumed that the airship does have a reasonable application for naval, military, and commercial purposes. Its graphic comparison with existing modes of transport which follows, it is believed, will justify this assumption as well as disclose a reliable index to probable costs.

Metalclad Compared With Ocean Craft

This comparison includes the estimated performance of two sizes of hypothetical airships. That both are of metalclad construction is because of the author's greater familiarity with this type and should not be interpreted as indicating any lack of interest or confidence in rigid airships covered with fabric. The

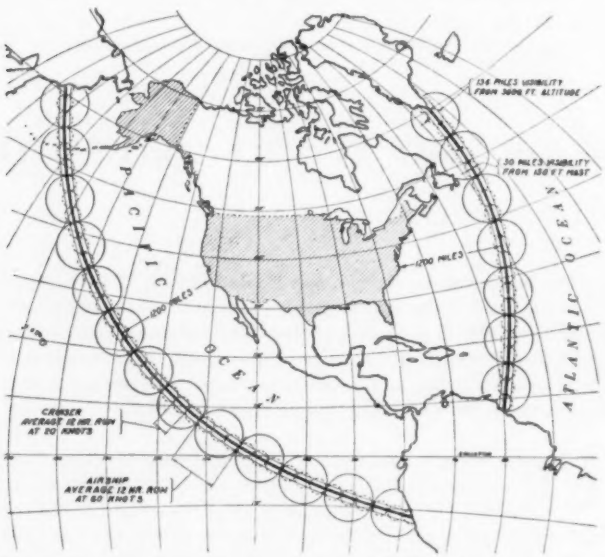


Fig. 2 Hypothetical Comparison of 300-Ton Airship Versus 10,000-Ton Cruiser on Scouting Duty

art is too young to permit of intolerance and unalterable opinions.

These two hypothetical airships, chosen for comparison with existing modes of transport, are as follows:

A 100-ton,[3] 100-m.p.h. metalclad, which it is proposed to build as the next succeeding unit following the successful construction of the ZMC-2, completed and delivered to the U. S. Navy in 1929.

A 300-ton,[3] 100-m.p.h. metalclad, which is of economic transatlantic size and should follow as a natural development within the next seven years.

It is proposed to equip each airship with sufficient engine power to provide a top speed of 100 m.p.h. This speed is considered of prime importance inasmuch as it assures ability of the airship to circumnavigate any severe storm area it may encounter. In this connection, it should be emphasized that while winds within a storm area may blow at a terrific speed, the storm area itself, whether it be a hurricane, typhoon, or cyclone, as a general rule, travels at a speed not in excess of 35 m.p.h. Therefore, properly warned by radio of the location and direction of any storm area, the airship can easily avoid it. Should the airship encounter the fringe of a storm area unexpectedly and without warning, it will possess sufficient speed to master the adverse wind currents, deflect its direction, and avoid the storm center.

[3] Long tons (2240 lb.). Air displacement, approximately 3,800,000 and 11,500,000 cu. ft., respectively.

The designation of the gross lift of these airships is expressed in long tons to facilitate comparison with ocean craft, with which it is proposed to enter them in competition from a performance standpoint. Pay load is expressed in pounds and in short tons in order to facilitate comparison, if any one is so disposed, with the performance of heavier-than-air craft attempting to engage on similar missions.

The general characteristics of a 100-ton metalclad airship are contained in Table 1, and those of a 300-ton airship in Table 2. Figs. 1 and 3 are the artist's conception of the commercial and military types of a ship of this size.

TABLE 1 100-TON METALCLAD AIRSHIP

General characteristics[a]

Gross lift		100 (long) tons
Hull volume		3,800,000 cu. ft.
Length		550 ft.
Maximum diameter		120 ft.
Engines (8) total		5,000 hp.
Maximum speed		100 m.p.h.
Cruising speed		70–80 m.p.h.
Commercial range		2,000 miles
Crew		35
Average fuel load		30,000 lb.
Pay load:		
50 passengers	12,500 lb.	
Express	2,500 lb.	
Mail	10,000 lb.	
Total pay load		25,000 lb.

[a] Based on 95 per cent inflation with 94 per cent pure helium at 60 deg. fahr. and 29.92 in. Hg lifting 62 lb. per 1000 cu. ft.

TABLE 2 300-TON METALCLAD AIRSHIP

General characteristics[a]

Gross lift		300 (long) tons
Hull volume		11,500,000 cu. ft.
Length		780 ft.
Maximum diameter		173 ft.
Engines		10,700 hp.
Maximum speed		100 m.p.h.
Crusing speed		70–80 m.p.h.
Commercial range (N. Y.–London)		3,470 miles
Crew		60
Fuel load		143,000 lb.
Pay load:		
Passengers[b]	50,000 lb.	
Express	60,000 lb.	
Mail	30,000 lb.	
Total pay load		140,000 lb.

[b] Includes additional weight of superior accommodations.

The 300-ton airship is presumed to be equipped with superior accommodations which first-class passengers are accustomed to find on palatial ocean liners. Hence the passenger capacity is limited to 100, although 200 could be carried, but with less comfort. The general-performance characteristics of these two airships applied to naval and military uses would, of course, vary to some degree due to the elimination of accommodations for passengers and commercial loads.

In the comparisons which follow, the performance data on naval and military craft are not guaranteed as accurate, but have been derived from Jane's handbook on the "Navies of the World," and from publicity releases growing out of the Washington conference in 1922, when the general specifications of a 10,000-ton cruiser were internationally defined. Such a cruiser is reported as being equipped with 107,000 hp.; has a maximum speed of 32.5 knots; consumes 0.75 lb. of fuel oil per hp-hr.; and carries a crew of 570 men.

Hence any citation of the performance of naval and military craft reveals no state secrets and for purposes of comparison should be treated in the nature of a supposition based on information which is already of public knowledge.

The performance characteristics of the Leviathan likewise are approximate, but are reasonably reliable, inasmuch as they represent an average of the several announcements and descriptions of this ship and its performance, which have been made public from time to time by the U. S. Shipping Board or have appeared ...

congressional hearings. Apparently they agree with the present capacity of the ship since it has been reconditioned for de luxe transatlantic service. It is realized that the indicated fuel consumption of the Leviathan is considerably higher than that now obtainable with more modern power-plant equipment. However, this fuel consumption is better than the average obtained with the general run of ocean liners now in service, and therefore affords a fair indication as to the relative cost of fuel per ton mile of pay load, as compared with the theoretical performance of airships.

In all of the comparisons which follow, it is assumed that the airships are traveling in still air and the ocean vessels, both military and commercial, are navigating in calm weather. Inasmuch as each type of craft carries 50 per cent fuel reserve to provide for adverse weather conditions, it is believed this assumption of ideal weather conditions for theoretical comparison is justified for the simple reason that extreme adverse weather will affect the speed and cruising radius of the ocean vessels approximately as much as it will affect the airship.

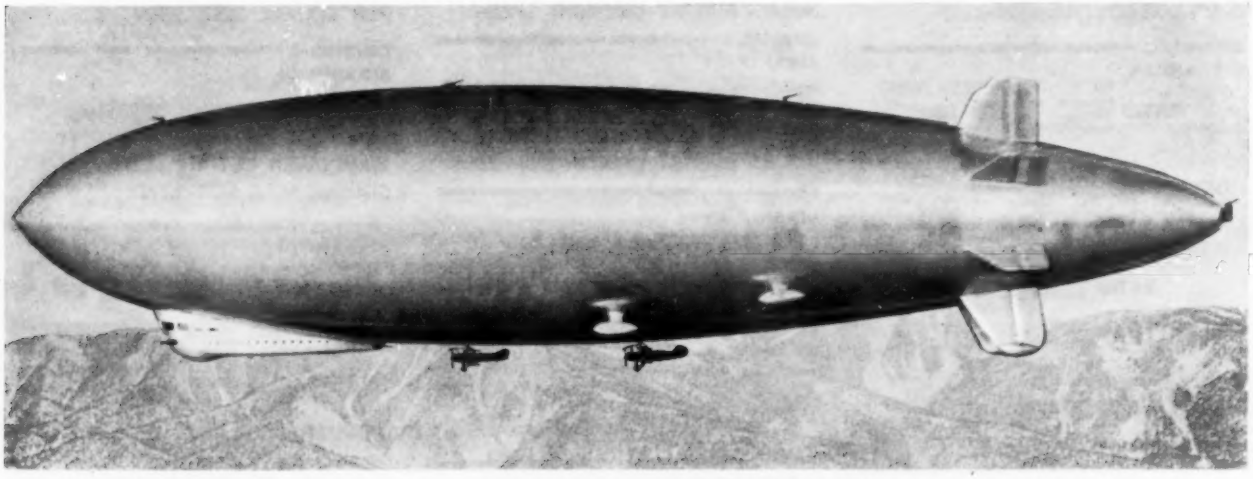


FIG. 3 PROPOSED MILITARY OR NAVAL TYPE OF METALCLAD AIRSHIP
(Gross lift 100 tons, speed 100 m.p.h., maximum range 4400 miles at 70 m.p.h.)

NAVAL APPLICATION

The following is a comparison of a 300-ton airship with a 10,000-ton cruiser engaged in scouting on the high seas. It is admitted the cruiser possesses certain valuable functions which the airship cannot perform. Therefore, the superiority of the airship on scout duty does not justify any conclusion that the cruiser is obsolete. On the contrary, it is assumed the cruiser is thus released for important duty elsewhere, which the airship is incapable of performing.

Fig. 2 is a map disclosing a theoretical defense of the east and west coasts of the United States, Alaska, and the Panama Canal against surprise attack by the enemy fleet. The succession of circles delineates the theoretical scout line located at an average distance of 1200 miles from the coast. This distance is chosen for the reason that in modern warfare it is recognized that the enemy fleet must, if possible, be intercepted and destroyed before its aircraft carriers get within bombing range of the thickly populated industrial centers along the seaboard. It is generally recognized that the best defense against an aerial attack on the Panama Canal or any large city is to destroy the enemy at its own base, namely, its aircraft carrier, before the air attack is launched, else any aircraft bombing squadron may take advantage of poor visibility and cloud formations to conceal its advance on its objective, and thus succeed with a disastrous surprise attack.

In the patrol path indicated on Fig. 2 the airships and the cruisers each are engaged solely in scouting duty. Their function is not to destroy but to discover the enemy and immediately radio to the high command of the main fleet, which is several hundred miles to the rear, full information with respect to the location, strength, and disposition of the enemy fleet. After supplying the main fleet with this vital information, the airships or cruisers have served their function well and ably, and their destruction by the enemy fleet will have little influence on the outcome of the approaching engagement.

With this vital information at its disposal, it is then the function of the main fleet to intercept the enemy fleet, if possible, before it gets within aircraft bombing range of any vulnerable point on the seaboard. Likewise, in the meantime, the probable objective of the enemy fleet will have been ascertained and the aerial defense (composed of army airplanes and airships)[4] along the seacoast, in close proximity to its objective, will be on the alert for any enemy bombing aircraft which may have escaped destruction by the main fleet. The above theory of aircraft defense is offered by the author, with apologies, as being modestly original and, so far as his knowledge is concerned, it does not disclose any military secrets of any nation.

ONE AIRSHIP RELEASES THREE CRUISERS

It will be observed from Fig. 2 that the effective cruising speed of the 10,000-ton cruiser engaged in scout duty is 20 knots (23 m.p.h.), whereas that of the airship is 60 knots (69 m.p.h.). Thus it is disclosed, when applied purely to scout duty, that one 300-ton airship is equivalent to three 10,000-ton cruisers in any range of action covering any definite time interval.

Fig. 4 graphically compares the complement and the estimated cost of these two types of craft. Fig. 5 discloses the superiority of the airship over the cruiser from the standpoint of visibility. From the top of a 150-ft. mast the visibility of the cruiser is limited to a radius of 15 miles. From an altitude of 3000 ft. the visibility of the airship increases to a radius of 67 miles. Engaged in a 12-hour run at 20 knots (23 m.p.h.) the cruiser can survey 8300 square miles, whereas the airship at 60 knots (69 m.p.h.) can survey 111,000 square miles, or over 13 times as much territory.

Referring again to Fig. 4 it is revealed that one cruiser has a

[4] Discussed in more detail under "Military Application."

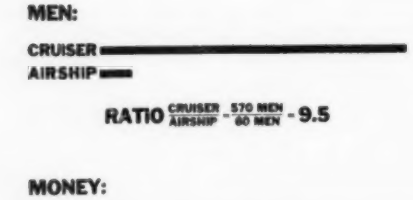


FIG. 4 COMPARISON OF COMPLEMENT AND COST OF AIRSHIP VERSUS CRUISER

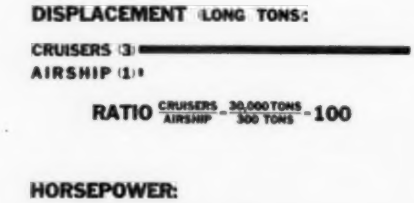


FIG. 7 DISPLACEMENT AND HORSEPOWER COMPARISON FOR AIRSHIP VERSUS THREE CRUISERS

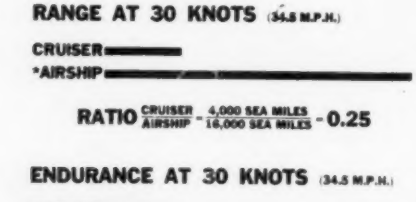


FIG. 10 COMPARISON OF TIME THAT CRUISER AND AIRSHIP CAN ENGAGE IN SCOUT DUTY

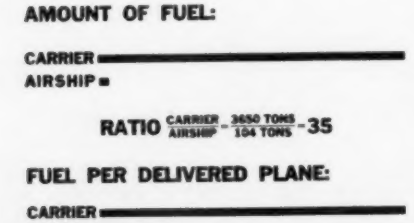


FIG. 13 FUEL COMPARISON FOR 300-TON AIRSHIP VERSUS AIRCRAFT CARRIER FROM SAN DIEGO TO HAWAII

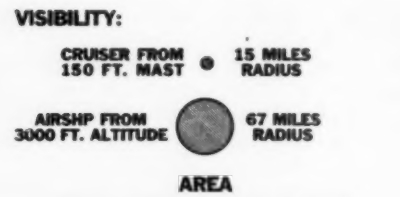


FIG. 5 COMPARISON OF VISIBILITY OF AIRSHIP VERSUS CRUISER

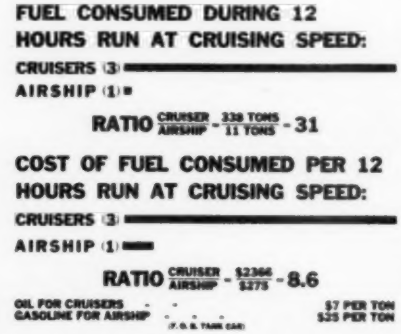


FIG. 8 COMPARISON OF FUEL CONSUMED ON 12-HOUR RUN AT CRUISING SPEED BY THREE CRUISERS AND ONE AIRSHIP

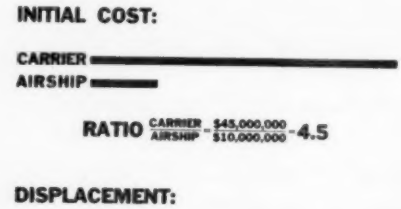


FIG. 11 COST AND DISPLACEMENT OF 300-TON AIRSHIP COMPARED WITH AIRCRAFT CARRIER

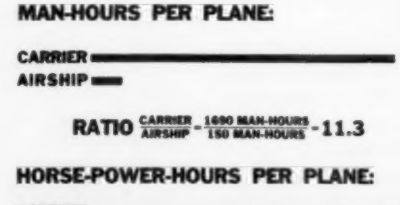


FIG. 14 MAN-HOURS AND HORSEPOWER-HOURS PER PLANE DELIVERED WITH 300-TON AIRSHIP VERSUS AIRCRAFT CARRIER FROM SAN DIEGO TO HAWAII

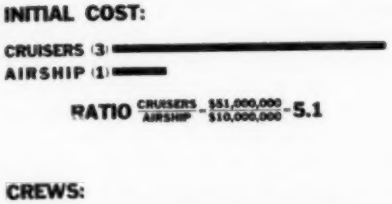


FIG. 6 COMPARISON OF COST AND CREWS FOR AIRSHIP VERSUS THREE CRUISERS TO ACCOMPLISH SAME MISSION

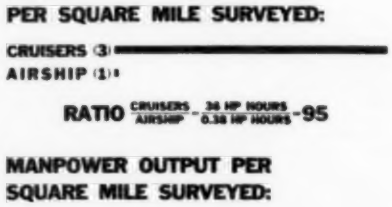


FIG. 9 AMOUNT OF ENERGY REQUIRED BY AIRSHIP VERSUS THREE CRUISERS TO PERFORM SAME MISSION

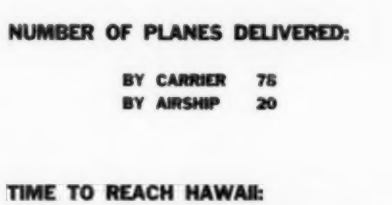


FIG. 12 NUMBER OF PLANES DELIVERED AND TIME SAN DIEGO TO HAWAII 300-TON AIRSHIP VERSUS AIRCRAFT CARRIER

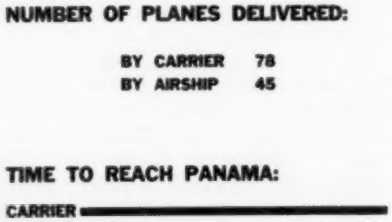


FIG. 15 NUMBER OF PLANES DELIVERED AND TIME PENSACOLA TO PANAMA, 300-TON AIRSHIP VERSUS AIRCRAFT CARRIER

complement of 570 men, whereas the airship has a complement of 60 men. Likewise the estimated cost of one cruiser is $17,000,000 and of one 300-ton airship, $10,000,000. Consequently in the event of the destruction of either type of craft by the enemy the loss in men for the cruiser is almost 10 times greater than that for the airship, and the loss in original cost of the cruiser 1.7 times that of the airship. Such economy in both men and money cannot be ignored in the planning of a naval defense and naval strategy adequate for the future.

Inasmuch as it is revealed that on scouting duty one 300-ton airship is equivalent to three 10,000-ton cruisers, it is of interest to indulge in a collective comparison of these two types of craft. Fig. 6 shows that to accomplish the same mission a capital investment of $51,000,000 is required to build three cruisers as against $10,000,000 for one airship, or over five times as much; also, that the complement of three cruisers is 1710 men as against 60 men for the airship, or 28½ times as much.

Fig. 7 reveals that the total displacement of three cruisers is 30,000 tons, as against 300 tons for the airship, or 100 times as much. Also that the total hp. required for the three cruisers is 321,000, as against 10,700 for the airship, or 30 times as much.

Fig. 8 shows that the fuel consumed on a 12-hour run at cruising speed by the three cruisers is 338 tons, as against 11 tons for the airship, or 31 times as much; also, that the cost of the fuel for the three cruisers is $2366, as against $275 for the airship, or more than 8 times as much.

Cruiser Requires 95 Times More Energy

Fig. 9 is perhaps the most important comparison of all. It discloses the amount of energy required to accomplish a given mission, this energy being measured in terms of both mechanical power and man power. The mechanical power output required per square mile surveyed by the cruiser is 36 hp-hr. as against 0.38 hp-hr. for the airship, or 95 times as much.

The man-power output required per square mile surveyed by the cruiser is 0.83 man-hours, as against 0.0065 man-hours for the airship, or 128 times as much.

Such startling disclosure of the superiority of the airship over the cruiser when engaged in scouting, it is maintained, presents circumstantial evidence of great preponderance which establishes the naval application of the airship, its economy in cost, and its efficiency in service.

At a first glance some will maintain, however, that the cruiser can engage in scout duty at sea for a longer interval of time than the airship. An examination of Fig. 10 reveals such is not the case. Based on the supposed fuel capacity of the cruiser and its theoretical consumption of fuel, it is estimated that the maximum range of the cruiser at 30 knots (34.5 m.p.h.) is 4000 sea miles, as against 16,000 sea miles (18,500 statute miles) for the airship, the superiority of the airship being four to one. With respect to endurance, the airship can cruise at a speed of 30 knots (34.5 m.p.h.) for 22 days, as against 5½ days for the cruiser, or 4 times as much.

It is realized that a low speed of 30 knots is uneconomical for the airship due to its propeller design being based on a high cruising speed, and that a 30-knot speed is too high to be economical for the cruiser. This speed is chosen in order to furnish a convenient yardstick to establish a comparison that is phenomenally interesting and which should convince those who doubt the superiority in range of action of the airship over that of the cruiser.

Airship Less Vulnerable

It will be maintained by some critics of the airship that it is too



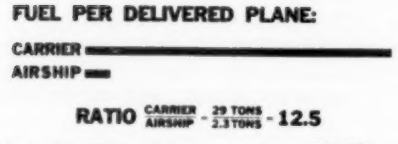
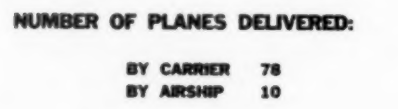


Fig. 16 Fuel Comparison for 300-Ton Airship Versus Aircraft Carrier From Pensacola to Panama



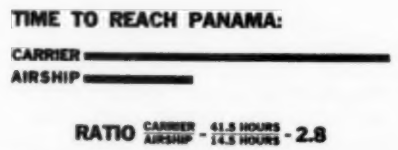
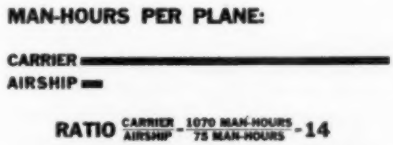


Fig. 19 Number of Planes Delivered and Time Miami to Panama, 100-Ton Airship Versus Aircraft Carrier



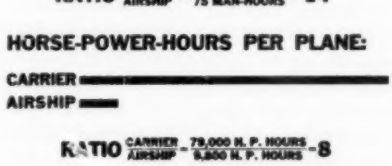


Fig. 17 Man-Hours and Horsepower-Hours per Plane Delivered With 300-Ton Airship Versus Aircraft Carrier From Pensacola to Panama

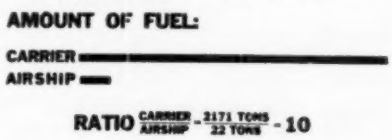
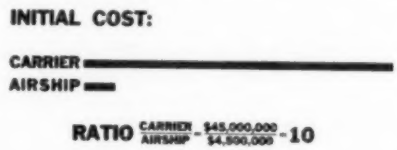




Fig. 20 Fuel Comparison for 100-Ton Airship Versus Aircraft Carrier From Miami to Panama



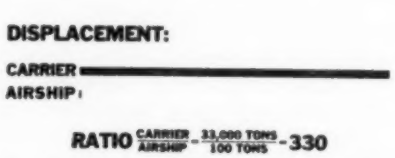
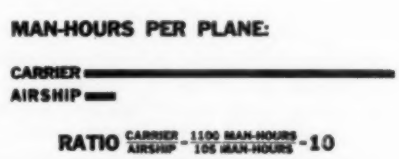


Fig. 18 Cost and Displacement of 100-Ton Airship Compared With Aircraft Carrier



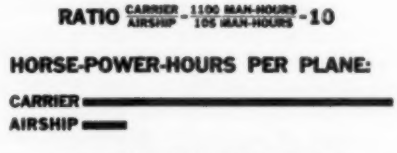


Fig. 21 Man-Hours and Horsepower-Hours per Plane Delivered With 100-Ton Airship Versus Aircraft Carrier From Miami to Panama

vulnerable and in close proximity to the range of anti-aircraft guns or when subjected to direct attack by airplane it will soon be destroyed. Granting this to be true, is it not also equally true that a 10,000-ton cruiser deployed several hundred miles in advance of the main fleet, when sighted by the enemy, will also in the majority of instances be destroyed because of the overwhelming superiority of the enemy fleet.

Comparing the two types of craft in this regard, however, the airship can sight the enemy fleet at longer range, determine its position and strength, reverse its direction and return to its main fleet with a speed at least three times that of the cruiser. Furthermore, it will be thoroughly practicable for a 300-ton airship to carry its own defense airplanes to aid in warding off enemy aerial attack. The airship it appears is vulnerable to attack principally from one source, namely, from the air, whereas the cruiser is subject to attack and destruction from a number of sources, including direct fire from the enemy fleet, attack by enemy bombing planes, attack by submarines, and destruction through contact with enemy mines. Certainly, it appears that the odds are in favor of the airship in so far as vulnerability is concerned.

Let it be recalled in conclusion that the experience of men and the function of equipment engaged in war is not to save themselves, but to spend themselves for the cause they serve. The airship, after discovering the enemy, its location, strength, and disposition, and having radioed this vital information to its own high command, has nobly served its function and what happens after will have no effect upon the outcome of the impending engagement between the two fleets. "Forewarned is forearmed" is the determining factor in a modern naval engagement.

MILITARY APPLICATION

The military application of the airship for army transport, reconnaissance, coast patrol, and other purposes is equally as important as is naval use. Perhaps the most vital problem concerning any army at any time is its service of supply. One phase of this problem takes first rank, namely, the quick transport of replacement airplanes to Panama and Hawaii in time of national emergency. It is unthinkable that sufficient airplanes could be maintained in storage to cope with the exigencies of war. The real test of effectiveness will be the quick construction and delivery of not only replacement planes, but the immediate concentration, at these strategic points, of aircraft units which normally have their base in the United States.

Obviously military planes cannot fly non-stop from California to Hawaii with any degree of certainty. Furthermore, present military planes cannot conveniently fly non-stop from Florida to Panama and still carry a full military load.

The delivery of replacement planes to Hawaii apparently must be made on board ocean vessels or aircraft carriers. The delivery of replacement and other military planes to Panama can be accomplished either by means of ocean vessels or aircraft carriers, or by two or three intermediate refuelings in Central America. However, without questioning the friendship of any of the countries to the south of the Rio Grande, it is obvious that military considerations demand that an alternative method of delivering airplanes to Panama be provided for in time of war other than by making use of refueling stations in Central America. The question of the violation of neutrality also arises. Obviously the most interesting alternative method is the use of aircraft carriers, both for Panama and for Hawaii. Therefore, it is proposed to compare the performance characteristics and respective efficiency of a 33,000-ton aircraft carrier with a 300-ton airship and with a 100-ton airship, each engaged in similar missions.

Table 3 gives certain performance characteristics necessary for this comparison.

TABLE 3 AIRCRAFT-CARRIER DATA

Displacement loaded	33,000 tons (long)
Horsepower	180,000
Cruising speed	30 knots (34½ m.p.h.)
Horsepower required (cruising)	140,000
Fuel oil ...ed per hp-hr	0.75 lb.
Crew (in ...uding airplane personnel)	1,900
Planes accommodated, assembled ready for combat	78

Figs. 11, 12, 13, and 14 give a graphic comparison of a 300-ton airship with an aircraft carrier, each engaged in delivering military airplanes from San Diego, California, to Hawaii. This comparison reveals an estimated cost of the carrier at $45,000,000 and of the airship, $10,000,000, or 4½ times greater for the carrier.

Convoy Duty, San Diego to Hawaii

The displacement of the carrier is 33,000 tons and of the airship 300 tons, or 110 times greater for the carrier. The carrier, it is presumed, will have a capacity for 78 airplanes ready for action, whereas the airship, by transporting 10 planes internally disposed and convoying 10 additional planes which are refueled periodically en route, can deliver 20 airplanes to Hawaii. The time required to reach Hawaii from San Diego, a distance of 2400 statute miles, at 70 knots (80 m.p.h.) is 30 hr., and for the carrier at 30 knots (34.5 m.p.h.) is 69.5 hr., or over 2 times greater.

The fuel required for the airship in still air is 104 tons and for the carrier, 3650 tons, or 35 times more. The fuel required per airplane delivered via airship is 5.2 tons and for the carrier, 46.8 tons, or nine times greater.

Again, the energy required for a given mission, which after all is perhaps the truest measure of efficiency, comes in for consideration. On the San Diego-Hawaii route, the man-hours required per airplane delivered by airship is 150 and via the carrier 1690, or over 11 times greater.

The horsepower-hours required per airplane delivered by airship are 15,720; via the carrier, 125,000, or 8 times greater. Four airships costing a total of less than the cost of one carrier can deliver the same number of planes from San Diego to Hawaii in less than half the elapsed time and at an expenditure of about ⅛ the energy.

Pensacola to Panama

Similarly the transportation of replacement military airplanes from Pensacola, Fla., to Panama via 300-ton airship and via carrier is graphically illustrated in Figs. 15, 16, and 17. Summarizing, two airships at a cost less than half that of one carrier can transport 12 more airplanes from Pensacola to Panama, a distance of 1530 statute miles, in half the time required for the carrier, and at an expenditure of only 1/14 the energy measured in man-hours and ⅛ the energy measured in horsepower-hours.

Miami to Panama

It is realized that the 300-ton airship still awaits future development and therefore it is important to compare the performance on a similar mission of a 100-ton airship (a size now practicable) engaged in convoying replacement airplanes to Panama from Miami, a distance of 1170 statute miles. This is a perfectly feasible tactical mission in so far as the 100-ton airship is concerned. Properly equipped with hook-on devices and cruising at a speed in excess of airplane stalling speed, it is capable of convoying 10 military airplanes, non-stop, by refueling en route in an elapsed time of 14½ hours.

A comparison of the performance of this 100-ton airship with a 33,000-ton airplane carrier is graphically described in Figs. 18, 19, 20, and 21. Summarizing, the energy required to transport one airplane from Miami to Panama by airship, measured in man power, is 105 man-hours and for the carrier 1100 man-hours, or over 10 times greater. Measured in mechanical power output

required, the airship expends energy equivalent to 11,000 hp-hr. per airplane delivered and the cruiser 74,500 hp-hr., or almost 7 times as much.

Long Distance Reconnaissance

While it is true in the above comparison the standard naval airplane carrier has been used, this is done solely for convenience, inasmuch as it is the fastest and most convenient ocean vessel available for such purposes. The designation of the naval carrier in this comparison, however, does not in any manner subtract from the army's responsibility to provide for alternative means for the quick transport of replacement airplanes from the United States to Panama in times of national emergency. It is believed that the use of the airship by the army for this purpose ranks equally in importance with any other military or naval use for which this type of craft might be employed.

Having convoyed airplanes to Panama, the airship is still available, because of its long endurance, to engage in scout duty and reconnaissance at considerable distance from the canal as a base. This is a service for which the airship is particularly well adapted in the rainy season, when, it is reported, military airplanes at Panama are often grounded a considerable part of the time on account of poor visibility and adverse weather conditions, which are very serious for the airplane, but do not interfere with the safe continuation of the airship on scout duty.

Some authorities maintain that it is impossible to defend the

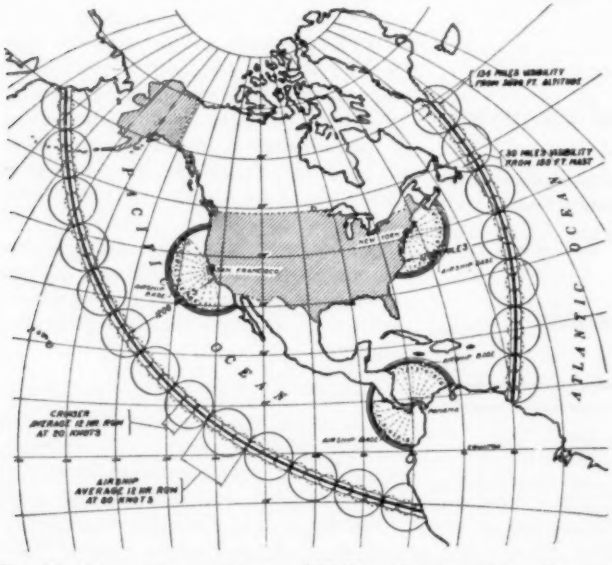


Fig. 22 Joint Coast Patrol 100-Ton Airship Plus Planes Versus Planes Alone

Panama Canal without having available an adequate number of airships for long-distance reconnaissance. In this respect, as an adjunct to the heavier-than-air force, they provide a most valuable auxiliary.

Airship Patrol Ideal for Panama

This is well illustrated in Figs. 22, 23, and 24, which describe the effectiveness of a 100-ton metalclad airship engaged in coast patrol in conjunction with observation airplanes.

Fig. 22 is a map on which are delineated four zones of activity, namely: Panama Canal, Atlantic side; Panama Canal, Pacific side; the eastern seaboard, with New York as the center of operations; and the western seaboard, with San Francisco as the center of operations.

Choosing the patrol problem of Panama on the Atlantic side for examination (see detail in Fig. 23), it will be observed that the airship is stationed 500 miles northeast of the canal. Its function is to cruise at low speed, serving as a mother ship and for refueling purposes for the eight observation military planes which are engaged constantly in patrol duty along an arc having a radius of 500 miles with Panama as its center, and extending from the shore of Nicaragua to the shore of Colombia.

Possessing an effective cruising speed of 125 m.p.h., it is feasible for a modern observation plane to negotiate the round trip from the airship to either Nicaragua or Colombia, and return in an elapsed time of 6 hr. At the conclusion of each round trip the plane hooks onto the airship for refueling and a 2-hr. rest, while simultaneously one of the reserve planes takes off to replace the returning plane. In this manner it is entire practicable, except when weather conditions are seriously adverse, to keep under constant observation every avenue of approach over water leading to the Panama Canal. A 100-ton airship is capable of an endurance of 48 hr. on this mission without the necessity of obtaining additional fuel for its own use and for the use of its observation airplanes. In this manner one airship plus eight observation planes should be able to establish and maintain an

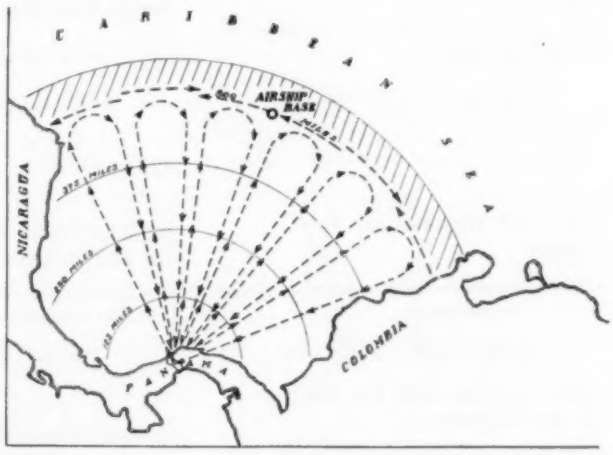


Fig. 23 Coast Patrol Summary

Zone	Observation planes required: Airship base	Land base
Panama (Atlantic side)	8	72
Panama (Pacific side)	8	72
New York	10	96
San Francisco	12	120
Total	38	360

almost air-tight defense against surprise attack over sea on the Panama Canal.

Radio Directs Attack

In actual practice when one of the observation planes has discovered the enemy, this information is relayed by a short-range radio to the airship and immediately transmitted over the powerful radio carried on the airship to the main aircraft base at the Panama Canal. Attack and bombardment squadrons then take off from Panama to intercept and destroy the enemy. Close watch of the enemy's maneuvers is maintained by the dirigible and its eight planes, and, by means of constant communication via radio with the airship, the attack and bombardment squadron commanders can direct their course to the enemy with precision.

The question will be raised, "Why establish a patrol at a distance of 500 miles from the canal?" The obvious answer is that the defense of the canal is thus made much more certain inasmuch as it is possible in this manner to discover the approach of the

enemy before its heavily laden bombing planes on board enemy aircraft carriers get within flight bombing range of the canal.

If an airship were not available for this purpose, naturally the observation planes would necessarily operate direct from a land base at the Panama Canal. To establish a patrol as frequent and as effective as with the airship as a base would require 72 instead of 8 observation planes. These planes would be divided into 6 squadrons of 12 units each, 10 planes being on active duty and 2 held in reserve. The units of each squadron would fly successively at one-hour intervals over the six loops, as indicated on Fig. 23. Having a 10-hour fuel capacity and flying at a speed of 125 m.p.h. each plane would require four hours to reach the patrol zone; would travel along the arc of the zone for one hour and require four hours to return to Panama for refueling. That this method would be exorbitantly expensive compared with the combination of one airship with eight observation planes is graphically described in Fig. 24.

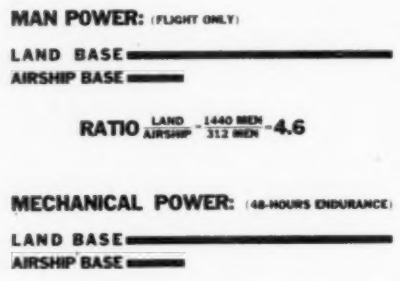


Fig. 24 Energy Required for 48-Hour Patrol With Land and Airship Bases

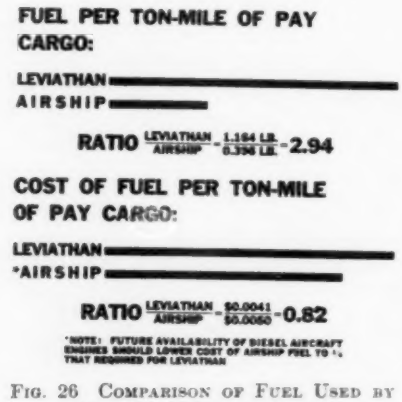


Fig. 26 Comparison of Fuel Used by Airship Versus Leviathan

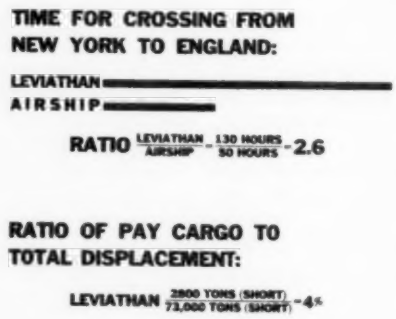


Fig. 25 Comparison of Time of Voyage and Pay Load for Airship Versus Leviathan

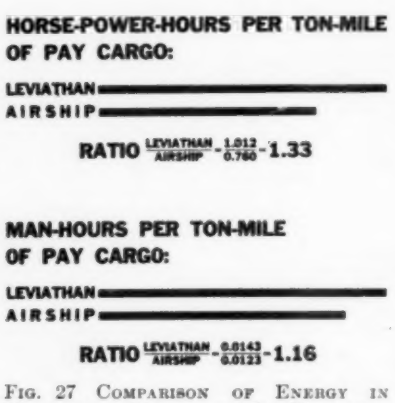


Fig. 27 Comparison of Energy in Horsepower-Hours and Man-Hours for Airship Versus Leviathan

ARMY OR NAVY

Obviously another question arises, "Is not this long-range patrol duty a function of the Navy rather than of the Army?" Apparently a reasonable answer to this question is that in times of national emergency no twilight zone or fine line of distinction will be drawn in determining whose duty it is to discover and destroy the enemy. In actual practice this is the obligation of every fighting arm of the government. The peril of air attack on vital centers such as the Panama Canal and elsewhere is such that both the Army and Navy must be in a position from the standpoint of equipment and experience to defeat the objective of the enemy, and for this purpose all forces immediately available and in close proximity to the point of attack must be and will be utilized.

Furthermore, some authorities maintain that the advent of the use of aircraft in modern warfare has automatically removed the zone of operations of the fleet from close proximity to the coast to a position located several hundred miles at sea. This is described in Fig. 22, where the outposts of the fleet on both the Atlantic and Pacific are located at a point 1200 miles from the coast. The main fleet cruises at a strategic distance behind the patrol zone ready to dash forward to intercept the enemy when warned of its approach. Immediately adjacent to the coast are the four theoretical zones of actual coast patrol radiating from New York, San Francisco, and the Panama Canal.

Airship Patrol More Efficient

Assuming, for purposes of comparison, that all four of these zones are patrolled simultaneously, it is shown in Fig. 23 that four airships, operating in conjunction with 38 observation planes, are able to accomplish as much work as 360 planes operating from land bases.

The energy required for a 48-hr. patrol mission is graphically described in Fig. 24. Using the airship as a base, 312 men (flight duty only) are required, as compared with 1440 men operating direct from land, or a ratio of 4.6 to 1 in favor of the airship. The mechanical power output required for the four airships and the 38 observation planes attached is 434,400 hp-hr. for a 48-hr. mission, while for the 360 planes operating from land bases is 1,944,000 hp-hr. or 4.5 times greater.

In the face of such an array of facts, no one can successfully dispute the military application of the airship, its economy in service, and its superiority for certain vital missions.

Other Military and Naval Uses

Other military and naval uses for the airship suggest themselves. For example, the 100-ton airship would be capable of transporting a 40,000-lb. military load from either Scott Field, Illinois, or Washington, D. C., to Panama in about 30 hr. It would also be valuable for the following purposes:

Protection of merchant marine
Destruction of enemy shipping
Convoy of troop ships
Direct reconnaissance
Training of personnel
Transportation of personnel and equipment
Observation of artillery practice
Patrol of Nicaragua Canal route
Quick transport of staff headquarters
Submarine search
Destruction of mines by direct gun fire
Courier service in the Caribbean
Patrol and direction of revenue cutter movement

Of these additional military uses, perhaps the most important are submarine search and destruction of mines by direct gun fire. The handicap of the airplane on such missions is due to its inability to cruise at low speed. On the other hand, the ability of

the airship to hover or fly at low speed enables it to search diligently the ocean depths for submarines, attack them independently or direct surface craft to their location, and thus accomplish their destruction with depth bombs or else destroy their will to come forth and do battle.

Airship Convoy War Record

It is a matter of official record of the last war that not one single American troop ship was successfully attacked by submarine while convoyed by airships. The record discloses that during the last year of the war American and British airships, engaged in convoy duty, flew a total of 2,000,000 miles, were in the air a sum total of 80,000 hr. and accounted for the destruction of 27 submarines. Certainly the superior visibility and adaptability of the airship in combating submarines, as compared to surface craft, is self-evident. Also, because of longer endurance the airship is superior from a tactical standpoint to the airplane. The necessity for precision in direct gun fire in destroying mines also gives the airship a distinct advantage over either surface craft or the airplane. This is true from the standpoint of both visibility and stability of the gun platform.

CIVIL APPLICATION

Peaceful missions of the airship, the accomplishment of which the Army and Navy are often charged with, are as follows:

River, harbor, and levee survey
Hydrographic survey
Topographic survey
Photography
Reclamation survey
Forest patrol and fire fighting
Medical relief in isolated sections
Bombing ice jams
Exploration

From an economic and humanitarian standpoint perhaps the most important of these have to do with Mississippi flood relief and control. For example, operating from Scott Field hangar at Belleville, Ill., a 100-ton airship engaged in flood relief in the Mississippi Valley can transport food and medical supplies to provide for 10,000 people each day. Possessing the ability to fly at low speed, of which the airplane is incapable, the airship can diligently search out refugees in isolated sections; cast supplies overboard via parachute and even lower Red Cross units to the ground by means of a practicable winch arrangement.

Aerial Survey More Economical

Such a ship would conveniently and economically expedite a detailed survey of the Mississippi Valley necessary for the planning of a comprehensive flood-control system. The enormous territory directly and indirectly affected by the Mississippi and its tributaries during flood periods comprises over 200,000 square miles (almost the size of the state of Texas) and lies in seventeen states, the respective adjacent areas of which are as follows:

State	Square miles
Alabama	1,000
Arkansas	36,000
Illinois	20,000
Indiana	7,000
Iowa	6,000
Kansas	3,000
Kentucky	10,000
Louisiana	37,000
Minnesota	2,000
Mississippi	23,000
Missouri	20,000
Nebraska	3,000
Ohio	3,000
Oklahoma	10,000
Tennessee	15,000
Texas	11,000
Wisconsin	3,000
Total	210,000

It is true that topographical maps and river surveys have been made of most of the Mississippi valley, but many of these maps, except the mosaics recently made via airplane, are a generation old and do not provide the detail essential for the planning of a successful flood-control system.

If it were undertaken to make a complete detailed survey on the ground of these 210,000 square miles, the map would never be completed up-to-date because it would take a generation to finish the survey. On the other hand, by undertaking such work to which a large airship having a complete photographic laboratory on board is adaptable, it could be completed within less than two years at approximately only one-tenth the cost. It would provide a map precise and superior in detail, giving a complete bird's-eye view of the whole situation for careful study by Army engineers charged with this responsibility.

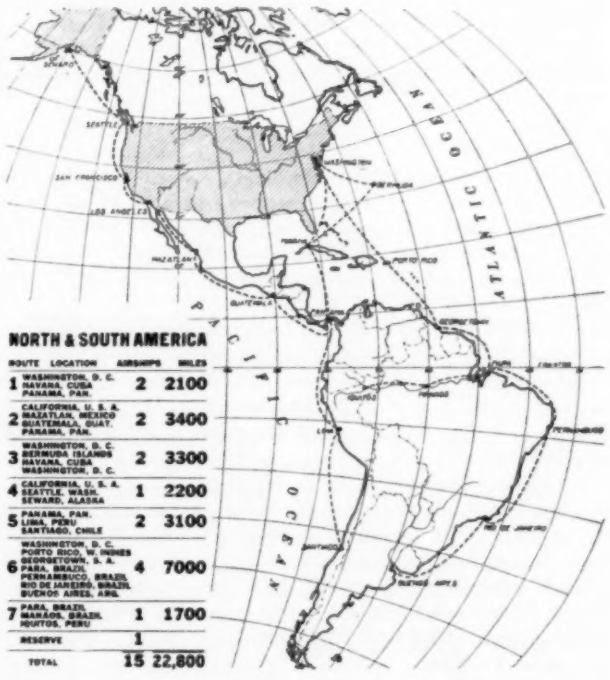


Fig. 28 Suggested Routes for Merchant Airships

Who can question that the foregoing uses of the airship engaged in either military or peaceful missions are worth while? Surely anything that will assist in diminishing the annual recurrence of devastating floods, costing thousands of human lives and the loss of millions in property, is of tremendous economic value.

COMMERCIAL APPLICATION

Most aeronautical engineers seem to be agreed that the airship rather than the airplane will prove to be more economical and satisfactory in long-distance heavy-cargo air transport. It is predicted that in the present generation airships will encircle the globe in regular operation, supplementing present land and water transport facilities, and, by shortening the line of communications, their use will expedite trade and thus increase the volume of commerce.

If this be true, the development of the airship is of vital economic importance to our foreign trade, which now annually represents a volume of 100,000,000 tons and a value of almost $10,000,000,000. Our annual shipping bill for commerce on the high seas for freight is $750,000,000, and for passengers $250,000,000, making a total of $1,000,000,000. To carry this enor-

mous volume of traffic requires 6000 vessels having a total displacement of 23,000,000 tons, and flying the flags of 28 countries. Of this volume of commerce, ships flying the American flag carry only 40 per cent in tonnage and only 34 per cent in value.

In foreign trade ready access to the source of supply has a profound influence upon a prospective customer. Therein lies the value of quick communication via airship. Consider the value of merchant airships flying between our eastern seaboard and Central and South America that would shorten the line of communications to Havana, Cuba, to only one day; from Washington, D. C., or from California to Panama, 2 days; to Lima, Peru 3 days; Santiago, Chile, 4 days; to Para (mouth of the Amazon), 3 days; Rio de Janeiro, 4 days; and to Buenos Aires, 5 days.

Certainly if this can be accomplished it will have an everlasting influence upon the prosperity of our trade with those countries, and upon the promotion of a friendly relationship between North and South America. Applied in a world-wide sense, the adaptation of American-built and American-owned merchant airships means American supremacy in the great international trade routes.

Airship vs. "Leviathan"

Any consideration of the employment of the airship engaged in international trade requires comparison with existing modes of transport. For this purpose a direct comparison of the performance of a 300-ton airship with the *Leviathan* engaged in trade between New York and England is graphically described in Figs. 25, 26, and 27.

Individual data for both the *Leviathan* and the airship are contained in Table 4. The airship traveling direct to London, England, would negotiate a journey of 3470 statute miles and the *Leviathan* from New York to Southampton, the nearest convenient port to London, would negotiate a journey of 3580 statute miles.

Engaged in this service, the airship is capable of transporting 70 tons of pay load, whereas the *Leviathan* carries only 2800 tons of pay load. This is graphically compared in Fig. 25, which discloses that the airship is capable of carrying 21 per cent of its gross lift in pay load, whereas the *Leviathan* can carry only 4 per cent of its gross displacement in pay load. This circumstantial evidence alone should establish the economic future of the airship as a merchant vessel.

TABLE 4 AIRSHIP DATA—METALCLAD CONSTRUCTION

Displacement (loaded)	300 tons (long)
Horsepower	10,700
Gasoline used per hp-hr	0.52 lb.
Cost of gasoline per ton[a]	$25
Crew	60
Effective crusing speed	61 knots (70 m.p.h.)
Hp. required	3,700
Passengers	100
Total pay load	70 tons (short)

"LEVIATHAN" DATA

Displacement (loaded)	65,000 tons (long)
Horsepower	90,000
Fuel oil used per hp-hr	1.15 lb.
Cost of fuel oil per ton[a]	$7
Crew	1,100
Effective cruising speed	24 knots (28 m.p.h.)
Hp. required	78,000
Passengers	2,700
Total pay load	2,800 tons (short)

[a] F.O.B. tank car.

In still air the airship would require 50 hr. for the crossing and the *Leviathan* about 130 hr., a difference of 80 hr., or over 3 days, the airship cruising at 61 knots (70 m.p.h.) and the *Leviathan* cruising at 24 knots (28 m.p.h.).

Fig. 26 shows that the airship requires 0.396 lb. of fuel per ton mile of pay cargo, whereas the *Leviathan* requires 1.164 lb. of fuel per ton mile of pay cargo, or 2.94 times greater.

Airship Requires Less Energy Per Ton Mile

Fig. 26 indicates also that even though the airship is burning gasoline, whereas the *Leviathan* is using cheap fuel oil, there is little difference in the actual cost of fuel per ton mile of pay cargo. The future availability of Diesel aircraft engines should lower the cost of airship fuel to considerably less than that required for the *Leviathan*.

It is startling to observe from Fig. 27 that the energy required to transport one ton of pay cargo from New York to London is less for the airship than for the *Leviathan*. This energy measured in horsepower-hours per ton mile of pay cargo for the airship is 0.76, and for the *Leviathan* 1.012, or 1.33 times greater. The man-hours per ton mile of pay cargo for the airship are 0.0123, and for the *Leviathan* 0.0143, or 1.16 times greater. This comparison of energy required is of vital consideration when its cost in dollars and cents is examined in Table 5, which is based on figures made public by the U. S. Shipping Board.

TABLE 5 "LEVIATHAN" OPERATING EXPENSES

(Fiscal year ending June 30, 1928)

Wages	$ 943,930
Fuel	2,004,652
Food and stores	1,409,594
Stevedoring	17,917
Other cargo expense	33,671
Port charges	497,483
Brokerage fees	1,251,767
Adjustments	75,025
Insurance	181,021
Maintenance	266,763
Betterments	76,382
Advertising	161,004
Administrative	136,484
Miscellaneous	470,223
Total expenses	$7,525,916

The preceding figures, based on Government operation, do not include depreciation and taxes. The total cost of energy measured in man-power and mechanical power for one year is as follows:

Wages	$ 943,930
Fuel	2,004,652
Maintenance of power plants (est.)	200,000
Total	$3,148,582

This cost of energy represents 42 per cent of all reported expenses. If depreciation of power plants were included, doubtless this cost of energy would represent approximately 50 per cent of all expenses. This justifies beyond all question the relative accuracy of the use of energy as a yardstick to determine the future economic value of airships, as compared to existing modes of transport.

In view of the foregoing circumstantial evidence, it is not too optimistic to predict that the large merchant airship of the future, irrespective of its saving in time, will be able to compete successfully with palatial ocean liners on a dollar-and-cents basis in the transport of first-class passengers, mail, and express.

The 100-Ton Airship

But the 300-ton airship must await future development. Hence, for immediate application, we must return again to the 100-ton airship which, comparatively speaking, can now be seriously considered in connection with the establishment of commercial airship lines. A ship of this size is too small for regular transatlantic service. However, it will carry an ample commercial load, including 50 passengers, to be profitable in developing the natural trade routes which lead from the centers of population on the east and west coasts to the Panama Canal and to South America. It is admirable for service to such important islands as Bermuda and Cuba.

Fig. 28 is a map on which are delineated seven routes to which

this size ship is adaptable and which represent a reasonable 10-year building program.

Washington-Havana-Panama Route

A detailed study of the proposed Washington-Havana-Panama route shows that the vicinity of Washington is chosen as the northern terminal instead of New York. This is considered desirable because meteorological conditions near Washington favor continuous operation with freedom from interruption by violent weather. Existing transport facilities, rail, and airplane, can be used to connect New York, Boston, Detroit, Philadelphia, Chicago, and other industrial centers with the Washington airship terminal. Furthermore, Washington is easily accessible to 80 per cent of the industrial population of the United States.

In addition, when, by using the airship, from two days to as much as two weeks' time is saved, it is not important that the northern terminal be located immediately adjacent to the largest city in the United States. A reasonable time table for this route is contained in Table 6.

TABLE 6 PANAMA OPERATING SCHEDULE

New York, Philadelphia, Washington, Havana, Panama

Southbound—Read down			Northbound—Read up	
Monday, Wednesday, Saturday	12:05 a.m. Lv.	Ar. 7:00 a.m. New York	Wednesday, Friday, Sunday
Monday, Wednesday, Saturday	2:10 a.m. Lv.	Ar. 7:00 a.m. Philadelphia	Wednesday, Friday, Sunday
Monday, Wednesday, Saturday	7:30 a.m. Lv.	Ar. 10:30 p.m. Washington	Tuesday, Thursday, Saturday
Monday, Wednesday, Saturday	10:00 p.m. Ar.	Lv. 8:00 a.m. Havana	Tuesday, Thursday, Saturday
Tuesday, Thursday, Sunday	7:00 a.m. Lv.	Ar. 8:30 p.m. Havana	Monday, Wednesday, Friday
Tuesday, Thursday, Sunday	8:30 p.m. Ar.	Lv. 7:00 a.m. Panama	Monday, Wednesday, Friday

Note: New York and Philadelphia via Pennsylvania Railroad to Washington Airship Terminal.

ELAPSED TIME SCHEDULE

	Havana hours	Panama hours
New York to	22	44½
Philadelphia to	20	42½
Washington to	14½	37
Havana to	..	13½

Stop over at Havana: Southbound, 9 hours; Northbound, 11½ hours.
Stop over at Panama: 10½ hours.

Performance Data

A general estimate of the performance of a 100-ton commercial airship is found in Figs. 29, 30, 31, 32, 33, and 34. In preparing these graphic charts, particularly with reference to operating costs, an attempt has been made to be conservative in order that the figures herein indicated may be accepted as reasonably reliable. These estimated costs are not guaranteed as perfect. There is no definite experience at this time on which to base an exact estimate. However, access has been had to the related experience to be found in the actual operation of the larger and more important air-mail transport lines in the United States. Therefore, no important essential items have been omitted, and while future experience may disclose that individual figures are in error, it is believed that the totals are reasonably certain for purposes of estimate.

Fig. 29 and Table 7, based on a cruising speed of 80 m.p.h., show the fuel load required with a 50 per cent reserve and the pounds of pay load carried over any given range.

Fig. 30 and Table 9, based on a full pay load and a cruising speed of 80 m.p.h., show the actual cost per passenger over any given range. Fig. 31 and Table 9, based on a full pay load and a cruising speed of 80 m.p.h., show the actual cost per pound of mail and express over any given range. Fig. 32 and Table 8, based on a cruising speed of 70 m.p.h., show the fuel load required with a 50 per cent reserve and the pounds of pay load carried over any given range.

TABLE 7 100-TON METALCLAD, 80 M.P.H. CRUISING SPEED

Range miles	Lb. fuel plus 50 per cent reserve	Pay load capacity lb. 50 passengers plus luggage	Mail and express
700	17,500	15,000	24,500
800	20,000	15,000	22,000
900	22,500	15,000	19,500
1000	25,000	15,000	17,000
1100	27,500	15,000	14,500
1200	30,000	15,000	12,000
1300	32,500	15,000	9,500
1400	35,000	15,000	7,000
1500	37,500	15,000	4,500
1600	40,000	12,000 (40 Pass.)	5,000
1700	42,500	9,000 (30 Pass.)	5,500
1800	45,000	9,000 (30 Pass.)	3,000

TABLE 8 PAY-LOAD CAPACITY 100-TON METALCLAD, 70 M.P.H. CRUISING SPEED

Range miles	Lb. fuel plus 50 per cent reserve	Pay load capacity lb. 50 passengers plus luggage	Mail and express
700	13,580	15,000	28,420
800	15,520	15,000	26,480
900	17,460	15,000	24,540
1000	19,400	15,000	22,600
1100	21,340	15,000	20,660
1200	23,280	15,000	18,720
1300	25,220	15,000	16,780
1400	27,160	15,000	14,840
1500	29,100	15,000	12,900
1600	31,040	15,000	10,960
1700	32,980	15,000	9,020
1800	34,920	15,000	7,080
1900	36,860	15,000	5,140
2000	38,800	12,000 (40 Pass.)	6,200
2100	40,740	12,000 (40 Pass.)	4,260
2200	42,680	9,000 (30 Pass.)	5,320
2300	44,620	9,000 (30 Pass.)	3,380
2400	46,560	6,000 (20 Pass.)	4,440
2500	48,500	6,000 (20 Pass.)	2,500

Fig. 33 and Table 9, based on a full pay load and a cruising speed of 70 m.p.h., show the actual cost per passenger over any given range. Fig. 34 and Table 9, based on a full pay load and a cruising speed of 70 m.p.h., show the actual cost per pound of mail and express over any given range.

These graphic charts and tables are based on full load in order to simplify their application to the study of any particular route. They also contemplate an operating minimum of 250,000 miles per annum. No profit has been included. Sleeping accommodations are provided for 50 passengers, and the free luggage allowance is 50 lb. per passenger.

TABLE 9 UNIT OPERATING COSTS—100-TON METALCLAD—BASED ON FULL PAY LOAD AND 250,000 MILES OPERATION ANNUALLY

Range, miles	Cruising 80 m.p.h. Per pass., inc. 100 lb. luggage	Cruising 80 m.p.h. Mail and express, per lb.	Cruising 70 m.p.h. Per pass., inc. 100 lb. luggage	Cruising 70 m.p.h. Mail and express, per lb.
700	$ 54.00	$0.18	$ 48.00	$0.16
800	66.00	0.22	57.00	0.19
900	78.00	0.26	66.00	0.22
1000	93.00	0.31	78.00	0.26
1100	111.00	0.37	90.00	0.30
1200	135.00	0.45	105.00	0.35
1300	159.00	0.53	120.00	0.40
1400	192.00	0.64	138.00	0.46
1500	231.00	0.77	159.00	0.53
1600	282.00	0.94	183.00	0.61
1700	351.00	1.17	210.00	0.70
1800	450.00	1.50	240.00	0.80
1900	.....	...	279.00	0.93
2000	.....	...	324.00	1.08
2100	.....	...	381.00	1.27
2200	.....	...	453.00	1.51
2300	.....	...	549.00	1.83
2400	.....	...	678.00	2.26
2500	.....	...	867.00	2.89

Of course it is realized that no commercial shipping line carries 100 per cent pay load on every trip, and, furthermore, in the operation of airships there will be certain short seasons when, because of adverse weather, certain scheduled trips will necessarily be cancelled. Some time will also be lost because of the necessity for overhauling and repairs.

Commercial Rate

Therefore, in using these tables the following formula applies:

$$\frac{\text{Unit cost per passenger or per pound}}{\text{Load factor} \times \text{Service factor}} + \text{Profit} = \text{Commercial rate}$$

To illustrate: Consulting Table 9 it is found that the cost per passenger on an 800-mile range is $66. Assuming that in actual operation this ship will carry on an average 75 per cent of its pay load capacity and will complete 80 per cent of its scheduled trips per annum, this will give a load factor of 0.75 and a service factor

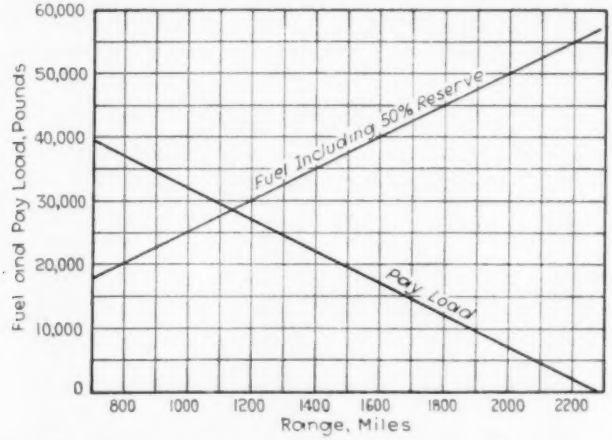


Fig. 29 Pounds of Pay Load and Fuel Required for 100-Ton Commercial Airship at 80 M.p.h. Cruising Speed

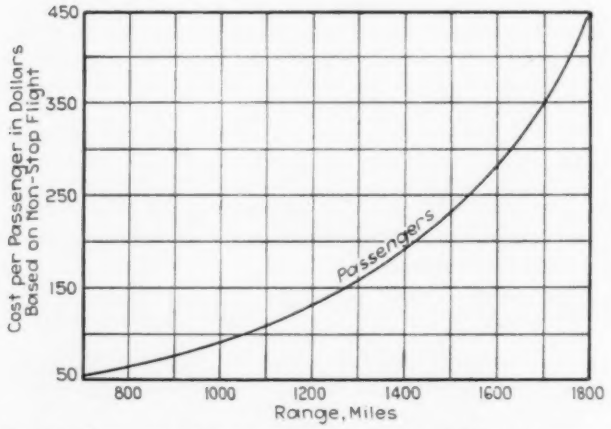


Fig. 30 Operating Cost per Passenger at 80 M.p.h. Cruising Speed, Based on 100 Per Cent Pay-Load Capacity

of 0.80. On this basis the actual cost per passenger is determined as follows:

$$\frac{\$66}{0.75 \times 0.80} = \$110$$

To this must be added a profit, which in turn will vary with the different routes and will be determined by the amount of capital required, including flying equipment and terminal facilities. It is believed that the pioneering hazard justifies the assumption of an annual profit of 20 per cent on the capital invested.

Table 10 sets forth in detail the estimated operating costs per mile based on a cruising speed of 80 m.p.h., a fuel reserve of 50 per cent, and the annual operation of 250,000 miles. In estimating the fuel cost it is assumed that 25 per cent of the reserve will be used in overcoming head winds. This table reveals in round numbers an operating cost of $10 per mile. Should the cruising speed be reduced to 70 m.p.h., the cost would be reduced to $9.83 per mile, because of the saving in fuel. Should two airships be operated over the same route, utilizing the same terminal facilities and each absorbing its share of overhead, the cost per mile would be reduced to $8.51. Additional units utilizing the same terminal facilities, repair shops, etc., would further reduce the cost.

It is interesting to observe that the much-heralded high cost of mooring airships has been grossly exaggerated. According to Table 10 this item represents only 10 per cent of the entire cost per mile. It is believed this terminal charge will be found to compare most favorably with present terminal expense involved in the operation of commercial airplanes.

In estimating the operating cost over any route, it is recommended that where the distance between the terminals does not exceed 1300 miles, the cost curves in Figs. 30 and 31, based on 80 m.p.h. cruising speed, should be used. Where the distance exceeds 1300 miles, the cost curves in Figs. 33 and 34, based on 70

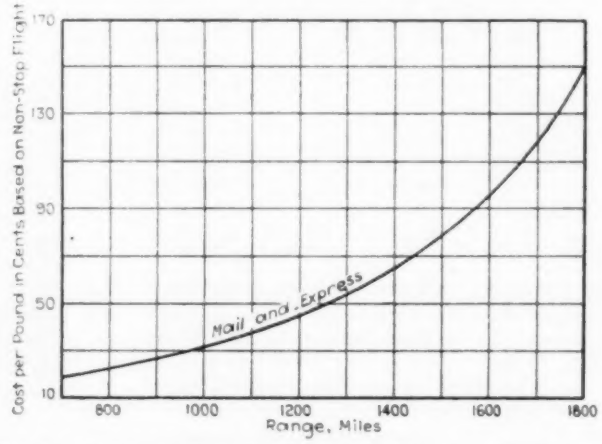


Fig. 31 Operating Cost Per Pound of Mail and Express at 80 M.p.h. Cruising Speed, Based on 100 Per Cent Pay-Load Capacity

m.p.h. cruising speed, should be used. In other words, on the longer ranges it is more economical to reduce the speed, thus reducing the fuel requirements and increasing the pay load accordingly.

Transportation Charges

On the basis of the preceding analyses of costs and assumed operation of two airships on the Washington-Havana-Panama

TABLE 10 OPERATING COST PER MILE

100-ton Metalclad—Sleeping accommodations for 50 passengers—Cruising speed, 80 m.p.h., Fuel reserve, 50 per cent—Cost based on operating 250,000 miles per annum

	Item		Cost per mile
1	Executive		$ 0.20
2	Administration		0.30
3	Traffic		0.30
4	Transportation:		
	(a) Fuel	$0.83	
	(b) Crew	0.58	
	(c) Helium	0.74	
	(d) Miscellaneous	0.20	
			2.35
5	Terminals		1.00
6	Maintenance		0.99
7	Communications		0.30
8	Insurance		1.00
9	Depreciation:		
	(a) Engines	$0.58	
	(b) Airships	2.00	
	(c) Buildings	0.59	
			3.17
10	Unforeseen items		0.40
	Total cost per mile		$10.01

line, it is believed that the transportation charges, as indicated in Table 11, are satisfactory for the purpose of preliminary estimate.

TABLE 11 WASHINGTON-HAVANA-PANAMA LINE—SCHEDULE OF AIRSHIP PASSENGER FARES

(Including railroad or bus transportation to airship terminal)

	New York	Philadelphia	Washington	Havana	Panama
New York	....	....	....	$215	$370
Philadelphia	....	....	....	$212	$367
Washington	....	....	....	$200	$355
Havana	$215	$212	$200	....	$155
Panama	$370	$367	$355	$155	....

Baggage arrangements: 50 lb. baggage per passenger carried free. Excess baggage charge 75 cents per lb. between Washington and Havana, and between Havana and Panama. Excess baggage $1.50 per lb. Washington to Panama.

Express rates: Express, exclusive of jewelry, bullion, and securities, transported at the following rates, including insurance:

Washington to Havana	$1 per lb.
Havana to Panama	$1 per lb.
Washington to Panama	$2 per lb.

Minimum mail rates: $0.0008 per lb. mile. Washington to Panama $1.64 per lb.

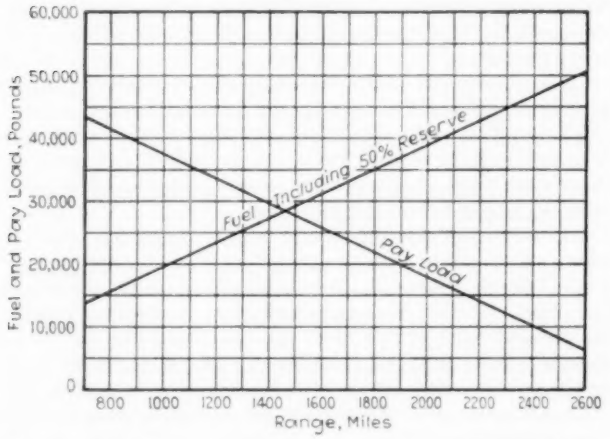


FIG. 32 POUNDS OF PAY LOAD AND FUEL REQUIRED AT 70 M.P.H. CRUISING SPEED

The estimated capital required for the establishment of service adequate for three round trips per week between Washington and Panama is given in Table 12.

TABLE 12 ESTIMATED CAPITAL REQUIRED

Two metalclad airships (100 tons gross lift each)	$ 8,000,000
10,000,000 cu. ft. helium at $40 per M. inc. freight	400,000
Spare parts, reserve engines, etc	200,000
One airship hangar 325 ft. wide by 700 ft. long 200,000 sq. ft. at $10	2,000,000
Four mooring towers, including fuel tanks and utilities	800,000
Helium purification plant and gasometers	400,000
Repair shops	150,000
Crews' quarters and miscellaneous buildings	100,000
Terminal buildings and equipment	300,000
Land	200,000
Electrical equipment, radio stations, etc	200,000
Unforeseen items	750,000
Working capital	1,500,000
Total capital investment required	$15,000,000

The annual mileage of two 100-ton airships operating on the Washington-Havana-Panama line would be as follows:

Air line distance, Washington to Havana	1,160 miles
Air line distance, Havana to Panama	930 miles
Total distance, one-way trip	2,090 miles

On the basis of a 100 per cent operating schedule each ship will make 1½ round trips per week. Two ships will make a total of 312 one-way trips per annum. 312 trips at 2090 miles equals 652,080 gross mileage per annum for both ships. Allowing for cancelled trips on account of weather, overhauling, etc., it is conservatively estimated that 80 per cent of the scheduled trips will be completed, thus giving a total gross mileage for both ships of 521,664 miles.

The revenue necessary per trip in order to earn a profit of 20 per cent on the investment of $15,000,000 is shown in Table 13.

TABLE 13 REVENUE NECESSARY PER TRIP

(Scheduled trips completed estimated at 80 per cent)

On the basis of operating two ships, the operating cost per mile is	$8.51
The total distance from Washington via Havana to Panama, is	2,090 miles
2,090 miles at $8.51 gives a total operating cost per trip of	$17,785
Add—Profit based on 20 per cent of total capital invested ($15,000,000) 2,090 miles at $5.75	12,018
Total revenue necessary per trip	$29,803

Revenue necessary per mile $\frac{\$29,803}{2,090} = \$14.26$

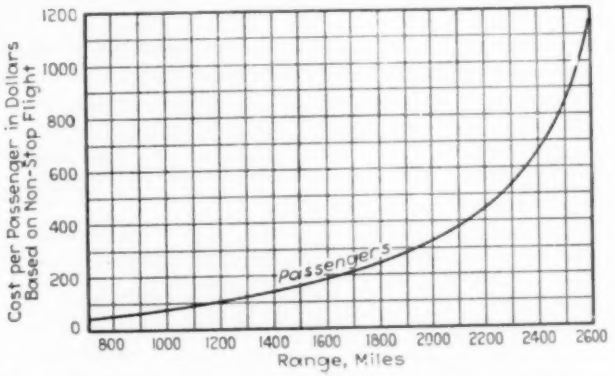


FIG. 33 OPERATING COST PER PASSENGER AT 70 M.P.H. CRUISING SPEED, BASED ON 100 PER CENT PAY-LOAD CAPACITY

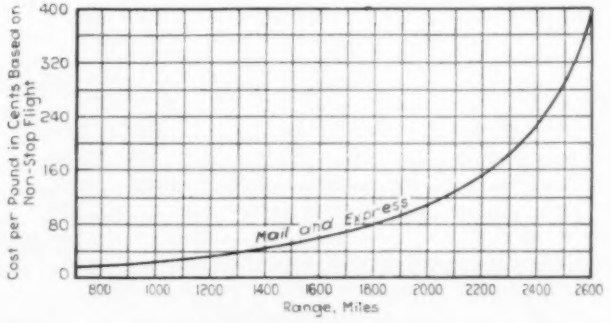


FIG. 34 OPERATING COST PER POUND OF MAIL AND EXPRESS AT 70 M.P.H. CRUISING SPEED, BASED ON 100 PER CENT PAY-LOAD CAPACITY

The anticipated revenue on the Washington-Havana-Panama line, assuming a 75 per cent pay load and a completion of 80 per cent of all scheduled trips, is given in Table 14.

TABLE 14 WASHINGTON-HAVANA-PANAMA LINE—ANTICIPATED GROSS REVENUE PER ANNUM

(Assuming 75 per cent payload and 80 per cent schedule)

75 per cent of 50 = 37½ passengers at $355	$ 13312	
75 per cent × 2500 lb. = 1875 lb. express at $2.00	3750	
10,000 lb. mail at $1.64	16400	
Total revenue per trip	$ 33,462	
80 per cent of 312 trips = 250 trips per annum × $33,462.00 equals gross revenue per annum of		$8,365,500
$17,785 × 312 (total possible number of trips per annum) reveals annual maximum operating cost of	$5,548,920	
Less saving in fuel, crews' pay, helium and depreciation per 62 trips cancelled	268,240	
Total operating cost per annum		5,280,680
Conservative anticipated annual profit		$3,084,820

This represents about 20 per cent profit on the total capital invested ($15,000,000).

TEN-YEAR PROGRAM OF DEVELOPMENT

The suggested program for future development which follows

may sound ambitious, yet it is thoroughly practicable from the standpoint of the development and commercial application of merchant airships. Certainly it will be a tremendous stimulus to international trade to shorten the distance between Washington and eastern industrial centers and Havana to only one day; Porto Rico, one day; from Washington and California to Panama, two days; the mouth of the Amazon, three days; Rio de Janeiro, four days; Santiago, four days; Buenos Aires, five days; the headwaters of the Amazon River, five to six days.

South and Central America during the coming century are destined to duplicate the marvellous development which during the past century has made the United States the most prosperous nation in the world.

All that is needed is improvement in transportation facilities and the shortening of the line of communications.

TEN-YEAR PROGRAM WITH SUGGESTED ROUTES

ROUTE NO. 1 WASHINGTON TO HAVANA-PANAMA

Distance	2,100 miles
Capital required	$15,000,000
Round trips	3 per week
Equipment required	2 airships
Beginning of operations	1934
Maximum annual mileage	655,200 miles

ROUTE NO. 2 CALIFORNIA TO MAZATLAN, MEX., GUATEMALA, PANAMA

Distance	3,400 miles
Capital required	$15,000,000
Round trips	2 per week
Equipment required	2 airships
Beginning of operations	1935
Maximum annual mileage	707,200 miles

ROUTE NO. 3 WASHINGTON TO BERMUDA-HAVANA-WASHINGTON

Distance	3,300 miles
Capital required	$13,000,000
Round trips	2 per week
Equipment required	2 airships
Beginning of operations	1936
Maximum annual mileage	686,400 miles

ROUTE NO. 4 CALIFORNIA TO SEATTLE-SEWARD, ALASKA

Distance	2,200 miles
Capital required	$ 5,000,000
Round trips	1 per week
Equipment required	1 airship
Beginning of operations	1937
Maximum annual mileage	228,800 miles

ROUTE NO. 5 PANAMA TO LIMA, PERU, SANTIAGO, CHILE

Distance	3,100 miles
Capital required	$15,000,000
Round trips	2 per week
Equipment required	2 airships
Beginning of operations	1938
Maximum annual mileage	644,800

ROUTE NO. 6 WASHINGTON TO PORTO RICO-GEORGETOWN, PARA, PERNAMBUCO, RIO DE JANEIRO, BUENOS AIRES

Distance	7,000 miles
Capital required	$27,000,000
Round trips	2 per week
Equipment required	4 airships
Beginning of operations	1939
Maximum annual mileage	1,428,000 miles

ROUTE NO. 7 AMAZON RIVER VALLEY-PARA, MANAOS, IQUITOS, PERU

Distance	1,700 miles
Capital required	$ 6,000,000
Round trips	1 per week
Equipment required	1 airship
Beginning of operations	1940
Maximum annual mileage	176,800 miles

RESERVE EQUIPMENT

One airship:	
Capital required	$ 4,000,000

RECAPITULATION, TEN-YEAR PROGRAM

Total number of routes		7
Total route mileage—one way		22,800 miles
Maximum annual mileage		4,527,200 miles
Equipment required		15 airships
Capital required:		
(*a*) 15 Airships	$60,000,000	
(*b*) Terminals	30,000,000	
(*c*) Working capital	10,000,000	
Total		$100,000,000

CONCLUSION

The author recognizes the legitimate right of any one to interpret differently the facts presented herein and to question any conclusions that may be reached. Circumstantial evidence is not infallible. Experience should confirm the qualitative analysis, but quantitatively the results no doubt will vary.

The theoretical application of the airship for naval, military, civilian, and commercial uses appears so promising that the expenditure of both public and private funds for actual demonstration is well justified. The cost for certain missions for which the airship is peculiarly adapted should unquestionably be less than for existing forms of transport. The question as to whether it will pay can be answered only with reference to specific application of the airship.

With respect to naval and military uses, neither the value of adequate preparation for the national defense, nor the tragedy of a poisonous gas attack on a metropolitan center can be measured in dollars and cents. Long distance reconnaissance, coast patrol, and transportation of personnel and supplies appear to be the most useful application of the rigid airship in this regard. In any well-balanced program of national defense, the function of the airship is not to replace, but rather to release, certain existing facilities and equipment for other important duties for which they are peculiarly fitted, thus aiding in the efficient disposition of the armed forces.

Commercially the airship will supplement present facilities, not replace them. The airship will transport first-class passengers, mail and express at higher speed, do more work per unit of energy, save more time, earn more profits. It has power in abundance, long range, safety, comfort, dependability, and economy in operation and maintenance.

It is destined to play an important role in the general advancement and prosperity of our country. Its future is as brilliant as that of the first steamship. As a means of promoting the peaceful pursuits of commerce, as an instrument for bridging quickly the physical barriers that often are obstacles to better understanding among nations, it is an international asset. Because of its speed the airship will expedite trade and thus increase the volume of commerce. Therefore its general application to world trade routes is inevitable.

In 1900 Thomas A. Edison was asked if the automobile would ever pay. His reply is historic as to the automobile and prophetic as to the airship. "A great invention which facilitates commerce enriches a country just as much as the discovery of vast hordes of gold."

Discussion

GEORGE V. WHITTLE.[5] While the significance of the comparisons given in the paper between surface ships and airships may be questioned, as may also some of the suggested uses of the airships, these are details which do not affect the author's general conclusions regarding the utility of the airship, which are endorsed.

Airships have limitations which should be recognized, and claims for them restricted accordingly. Those who have studied the possibilities of the airship will, in the main, agree that its economic field is over the sea, where it has both naval and commercial uses. For long-distance operation over water the airship offers great carrying capacity, with comfort and safety, at a speed not attainable by water transportation. For short-distance travel over water, the flying boat seems best; and over land, the airplane appears economically supreme for high-speed transportation. It is generally understood that air transport does not compete directly with the older means of transportation, but rather supplements the latter and scores where time is of great importance. As indicated the airship and airplane will not be direct competitors, as each has a sphere of usefulness peculiarly

[5] Lieutenant (CC) U. S. N.

its own. In the highly developed air-transportation systems of the future, it seems that airships will be used on the long ocean routes, and will be served at the terminals by airplane and flying-boat lines serving as feeders to the main stem.

The spectacular flights of the *Graf Zeppelin*, and to a lesser extent, the numerous but not so much advertised flights of the *Los Angeles*, have focused public attention on the rigid airship as a means of transportation. The technical skill and manufacturing ability necessary to produce airships far superior to either of these exist in this country today. The new Navy airships, each of which will be roughly $2^1/_2$ times the volume of the *Los Angeles*, will be much superior in performance to either the *Los Angeles* or the *Graf Zeppelin*, due to the technical progress of the art and the gain which comes from increased size. The first of these airships should be ready for trials in 1931. Invention and refinement can be expected to contribute in a large way to the advancement of the art for many years to come. In this connection, attention is invited to the pioneering work being done at the Naval Air Station, Lakehurst, N. J., in the mooring and ground handling of airships in which mechanical equipment is being substituted for manual handling, which will ultimately effect considerable economies in operation.

Every means of transportation has requirements which must be met, if it is to function properly. In the case of the airship, one of the most important requirements is good meteorological information, which the government should provide commercial-airship operators in a manner analogous to the aids to navigation now provided for surface vessels.

Airships are here to stay. The technical, constructional, and operational talent is available for commercial-airship enterprise and awaits the call of the airship transport operator.

T. G. W. Settle.[6] Estimated performance and other data of two hypothetical metalclad airships of 100 and 300 tons displacement respectively are used in the study because, as the author states, he is more familiar with that type of vessel than with the Zeppelin type of ship. He very generously and properly indicates that this is not to be taken in any sense as a disparagement of the Zeppelin in favor of the metalclad. And the paper is to be construed as a presentation of the case of "The Airship" rather than that of a particular kind of airship, despite the fact that the data presented are labeled "metalclad." It is probable that the airship of the future will be neither strictly "Zeppelin" nor "metalclad" in the present rather narrow interpretations of these terms; and it is to be hoped and expected that present-day performance forecasts of future airships will be exceeded. The paper does not discuss nor compare the estimated technical characteristics of the large metalclad with Zeppelin-type vessels.

The author at the beginning briefly reviews the recent convincing "full-scale" demonstrations by the *Graf Zeppelin* of the commercial utility and practicability of the airship, and very justifiably takes the position that the airship has passed through the stage of "doubt and uncertainty" and has focused upon itself the attention of the world as a practical and useful vehicle for transport.

The point is well taken that engine power sufficient for higher speeds is most desirable, particularly with a view to enabling the airship to take best advantage of good weather conditions, or least disadvantage from adverse ones. Dr. Eckener, referring to the portion of his world cruise in the vicinity of Japan, said that he "played with the typhoon instead of allowing it to play with him." However, to make this possible in the bigger and better airships of the future, we must have airship engines (compression-ignition type, sooner or later), and in much larger power units than any available today. Probably the most "lagging" major feature of airship development at the present time is the engine. There is only one fully developed and proved airship engine in the world today, and that a gasoline (or gaseous fuel) 560-hp. engine. It is feared that airship-engine progress is doomed to lag behind structural, operational, and handling development for some time to come, and to react unfavorably, of course, on operational technique.

In the more detailed discussions of the several functions of the airship and analyses of costs in money and energy, in the pages of the paper, quite a mass of figures, statistics, and ratios are presented. In reading this over, it comes to one's mind that figures and statistics, like a good lawyer, can be made to "prove anything." Many of the assumptions and data on the airship side are hypothetical, or otherwise open to argument, while the corresponding data for the surface carrier are more largely based on actual experience. However, it seems to the writer that the bulk of the comparative "figures" are valid qualitatively if not always quantitatively, and that the airship side of the comparison is, in general, presented in the proper sense and conservatively.

Naval Application

The author compares the airship scout cruiser (300 tons) with a surface scout cruiser (10,000 tons) and very properly indicates that the former does not replace the latter in naval strategy, but does form a most valuable addition on the "scouting line" and can perform certain functions much more efficiently than the surface cruiser. However, the author displays a serious misconception of naval strategy in a major campaign. The scouting line is disposed not with reference to any coast line, but with reference to our "main body," that is, our "battle fleet," which in turn is governed in its movements by the disposition of the enemy and considerations of the "grand strategy" of the war, rather than by simple geography; a fleet is a mobile force which must not be "coast bound," unless forced to be so by enemy action and force. However, the above does not affect the main point sought to be made by the author in this section, that is, that the airship is a most valuable and essential type of fleet-scouting vessel, and one that for certain purposes is the equivalent of several surface cruisers.

Fig. 5 does not present altogether a true picture, since both the airship and surface cruisers will carry planes, which will often extend the effective visibility of each considerably; of course, the airship is a much better plane "carrier" than is the surface cruiser, in point of "getting the planes back;" the surface ship has to heave to, to hoist the plane aboard.

The simple and unqualified statement that an airship is equivalent to a certain number of surface cruisers is unfortunate, since no such simple equivalency can exist where there are such a large number of more or less indeterminate factors entering into the case. It would be as erroneous to state simply that a certain number of planes were equivalent to a certain amount of railroad rolling stock in the field of land transport.

The author has done well to emphasize the enormous cruising range and "sea-keeping" ability of the airship. It is believed that the airship has a very high degree of defense against and invulnerability to enemy action as is indicated by the author although, again, any numerical "odds" in regard to vulnerability of the two types of craft is without justification in so complex a problem.

The author could well have mentioned several legitimate secondary functions of the airship cruiser, that is, convoy escort, bombardment of enemy territory, enemy commerce destruction, dispatch, and maintenance of "our" line of communications.

Military Application

While the writer agrees very largely with the paper in the

[6] Lieutenant, U. S. N.

naval application of the airship, he cannot do so in the case of military application. It is believed that the Army's proper relation to and utilization of the airship will very closely parallel its relation to and utilization of other ocean-going vessels (surface).

Possibly in peace-time and probably in certain wars, the Army will need to utilize airships, in addition to surface ships, for overseas transport of men and munitions, but these airships, just as such surface vessels now and in the past so utilized, will be merchant ships, bought, chartered, or commandeered by the Army (or the Navy) and manned by merchant (or Navy) crews. In time of war, the available merchant airship tonnage will be largely divided between the Navy, for auxiliary cruisers (and transports), and the Army for transports, the allocation ratios depending upon the exigencies of the particular war, but probably always leaning heavily toward the Navy.

The comparison of an airship with a surface ship plane-carrier for transport of planes seems rather futile, since it is inconceivable that a surface carrier would be so used in a major war. However, for overseas transport of planes and other munitions, the merchant airship would probably compare very favorably with the surface cargo ship under many sets of circumstances.

It cannot be seen that the Army has any proper use whatever for the airship (large, rigid) for reconnaissance or coastal patrol. For reconnaissance the blimp (non-rigid or small pressure airship) may have some very infrequent application, but in general it seems that the airplane and cavalry are the Army's proper and usual arms for this function. For coastal patrol, that is, such coastal patrol functions as properly come under the cognizance of the Army (coast artillery), the blimp, without doubt, has a legitimate role, along with planes and Army mine craft in certain harbors.

The items in the list of "other military uses" for the airship have either already been discussed or should properly be handled by blimps, planes, or Navy agencies.

The last paragraph in this section deals with the wartime use of blimps by the Allies (Navy forces) for convoy escort duty, and the reason cannot be guessed for its inclusion at this point, in this paper on the "airship" (which throughout the paper is taken to mean the large rigid airship). A proper secondary function of a naval airship cruiser is as a convoy "ocean escort," as pointed out above.

Peaceful Application (Army)

The items listed under this heading, except "exploration," it would seem should properly be handled by blimps or planes, some under the cognizance of civilian departments of the government; at any event, the Army use of large rigid airships for these missions is not believed to be justifiable. Just what is meant by "exploration" is not clear, but it would seem that any exploration mission of a character such as to dictate the use of a rigid airship, should be carried on by a civilian department of the government or by private enterprise, and utilizing a commercial airship fitted up for the specific purpose.

Commercial Application

In this section the writer can again heartily agree with the author in his optimistic view of the future of airships in the commercial field. The point might well have been more clearly and definitely made that there is no competition between the airship and the airplane; that they rather have a complementary relationship to each other, much as the steamship and railroad feed into and serve each other.

As before, the writer does not put too much faith in the actual figures quoted, but believes that a good general comparative picture is presented which argues the case of the airship conservatively. The airship lines, of course, must have adequate financial backing and administrative organization, and necessary port works, just as steamship lines must have the same things; airship lines must have proper meteorological service just as surface ocean systems must have "hydrographic office" service and advices.

Rather than competing with the steamship, the airship will take the "cream" of passenger, mail, and cargo traffic from transoceanic surface carriers, in an analogous fashion to the airplane taking the cream of overland and coasting traffic from the railroad and coastwise steamers.

Conclusion

Paralleling the author's conclusion, it is believed that the practical application of the airship for naval and commercial uses has been fully demonstrated in full scale by the long list of successful airships which have been built and operating during the past score of years; the theoretical economic proofs, at least qualitatively, are very well handled in this paper.

However, it cannot be agreed that the airship has either a theoretical or a demonstrated "military" application; the airship's only logical relationship with an army is exactly that of the ocean-going surface craft. In this connection, the experience of the Germans during the war in connection with attempted military utilizations of airships is very enlightening, and is believed to fully uphold the views of the writer on this phase of the question.

In regard to questions "What will it cost?" and "Will it pay?" it is believed that the author has handled these very excellently in his paper; he has taken performance and estimated data at hand, and has made many instructive and interesting analyses of the economics of the airship versus surface carrier. While he admits the fallibility of this "circumstantial" evidence, and while the figures, curves, and numerical ratios given are not to be taken too literally, nevertheless, these all constitute what seems to be a good and conservative qualitative picture. This paper, and its analyses, will serve to crystallize thought in channels investigated; it will encourage healthy argument among "professionals," and will be very illuminating to laymen. While not agreeing at all points with the author, it is nevertheless believed that he has handled a difficult and involved subject excellently; his paper is a valuable addition to current airship literature, and he well deserves the appreciation of the airship "profession" for his painstaking and effective work.

AER-53-4

Rubber in Airplane Construction

By C. SAURER,[1] AKRON, OHIO

There are a large number of uses to which rubber can be put in airplane construction. Its ability to dissipate shocks and vibrations makes it useful for engine mountings, shock absorbers, and propeller insulators. It can also be used to produce a leakproof tank lining, for seat cushions, and as a lining for airplane cabins to insulate the passengers from outside noise and changes in temperature. The paper describes several types of construction in which rubber is used to advantage.

RUBBER at the present time, regardless of its form, shape, or size, is generally thought of in terms of devices for absorbing shocks and energy. This mental association is made because it has become well known from experience that the major physical characteristic of rubber is its ability to receive shocks and vibrations of varying intensities without passing them on to contacting objects, or in other words, its ability to dissipate energy. It is not claimed that rubber insulation, even when used to its best advantage, will dampen out all vibratory and shock movements, but rather that it is a most ideal substance for the absolute cushioning of these movements.

Rubber in its various compounded forms has other remarkable and interesting characteristics such as its ability, when acted upon by force, to change its form, either in the direction of compression or elongation, returning practically to its original form when the force is removed. Also its peculiar characteristic of almost equally resisting movement in all directions makes it fit in ideally in shock-insulation work. Tensile strength as high as 5000 lb. per sq. in., permanent set in the direction of compression as low as 7 per cent, hysteresis control, and variations in hardness ranging from soft, putty-like consistency to horny, hard rubber are all commercial possibilities available to present-day designers. However, regardless of what may be the other characteristics of rubber, it is its ability to dampen out shocks and vibrations that should most interest the airplane designer.

Vibrations are necessarily attendant to motor action, and it is the designer's and builder's job to reduce these vibrations to a minimum before being passed on to the fuselage and occupants of the plane. This is necessary not only to preserve the plane itself, but also to add to the comfort and safety of air transportation. For proof that rubber is and has been doing a wonderfully successful job in this capacity, one has only to refer to what has been accomplished in the automotive industry. The comparatively recent development wherein an effective bond can be secured between soft rubber and steel has put into the hands of the automobile designer one of the most potent tools for construction and performance improvements in recent years. Torque insulators smoothing out shocks in the transmission shafts, clutch rubber assemblies toning out jerks caused by sudden action of clutch engagement, rubber spring shackles, and engine-motor mountings eliminating 4/5 of motor vibrations, are all materially adding to the remarkable smoothness of automobile operation. One normally thinks of an automobile as a steel-fabricated vehicle, and yet a well-known manufacturer recently announced that its car contained over 400 separate rubber parts. This in itself is conclusive proof that rubber is doing an excellent job in the automotive field.

The particular underlying principles promoting these remarkable improvements in the automotive field are in the same way even more applicable to the airplane industry. In fact, it is largely due to the entirely satisfactory performance of rubber in automobiles that has prompted rubber manufacturers to lay out development programs for the extensive use of rubber in airplanes.

Rubber in Airplane-Engine Mountings

Of the 400 or more rubber parts used on an automobile, the most important, aside from tires, are the rubber motor supports. They have done more for the absorption of vibration and the elimination of noise than any other device.

In the airplane they are of even greater importance, as they retard and prevent crystallization of otherwise affected metal structure, thus prolonging its life. Much thought has been given to the design of airplane-motor supports, the requirements being very different from those of an automobile. In an automobile motor, the reciprocating action of the pistons sets up vibrations which are primarily vertical.

Of the many forms of rubber insulators the most successful ones have applied the principle of vertical vibration elimination, depending upon an efficient bond of rubber between steel parts. Figs. 1 and 2 show two such installations. Any desired degree of cushion in the vertical plane is obtainable without the ill effects of a substantial engine movement.

These motor supports are directly applicable to airplane motors of the "in-line" type. It is recommended that the support shown in Fig. 2 be used, because of its greater factor of safety. With the radial-type motor the problem becomes quite different as three major sets of vibrations are encountered. First, there are radially disposed vibrations in a vertical plane set up by the motor. Second, there are propellor vibrations, mostly running in opposite rotation to the motor vibration, but in the same vertical plane. Third, there are vibrations from the propeller thrust; these are at right angles to the vertical-plane vibrations and are radially disposed. Besides these three majo vibrations, many other factors, such as gravity, torque reaction, gyroscopic action, etc., have to be considered.

The best solution for these varied problems of vibration elimination on radial-type motors is to have radially installed motor supports, as shown in Fig. 3. These motor supports are not intended to absorb all vibration, but to absorb and reduce periodic vibration and to check a large part of the greater shocks. Safety, light weight, suitability for torque, torque reaction, and gyroscopic action were deciding factors in the adoption of this design. It is hardly possible to insulate the motor from the rest of the airplane in such a way that no vibrations are transmitted. To absorb all the vibrations completely an amount of material would be necessary which would so increase the weight of the plane as to make it impractical from a weight and cost standpoint.

---

[1] Firestone Tire and Rubber Company. Mr. Saurer was born in Switzerland. He studied in Germany and France, and was connected with the Saurer Truck Company. He came to America in 1919, and was Assistant Engineer of the Mack International Company; from 1922 to 1929 was chief engineer of the U. S. Rubber Company on automotive development; from 1929 on development and Sales Engineer for the Firestone Tire and Rubber Company. He has been instrumental in the introduction of many new rubber articles in the automotive field and lately has developed and studied new articles for airplane and railroad construction.

Presented at the Akron Meeting, Akron, Ohio, Oct. 21 to 23, 1929, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Rubber in Airplane Shock Absorbers

The shock to be absorbed by an airplane shock absorber should be not less than four times the static weight of the plane. To dissipate the shock of the landing impact, the travel of the shock

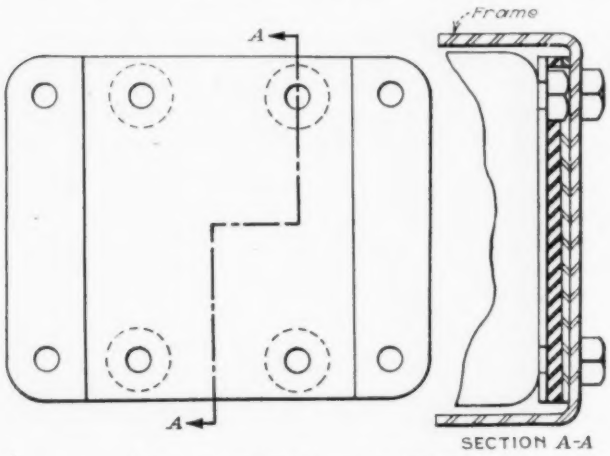


Fig. 1 Adhesion Type of Support

absorber should range over a long enough period of time to bring this weight slowly to rest. This has been accomplished by air, hydraulic, steel-spring, or rubber devices, or combinations of these. The light weight and strength of rubber have made the compression-disk and tension-cord shock absorbers popular with the airplane designer. The comparatively low cost and easy maintenance of such an installation have been an added reason for their use. The tension-cord shock absorber is undoubtedly the cheapest and lightest device to bring the falling weight of the plane to rest. A steel spring designed to absorb the same energy and weight in the same length of travel would be approximately 75 times heavier than the shock cords. According to the designer's views, the shock cords are used in various ways; they are laced between spokes and cross each other; they may be used straight and wound parallel. The differences in spacings and windings are many. The shock cords have a maximum elongation of 10 to 11 times the original length; in use they seldom exceed $^1/_2$ the total elongation possible. A cable of $^1/_2$ in. diameter carries approximately 220 lb. for every inch of elongation. Tests made with tension-cord shock absorbers show that approximately 14 per cent of the energy is absorbed by them, the variety in designs or spacings having very little effect on this.

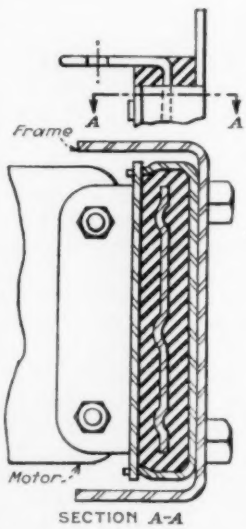


Fig. 2 Support for Airplane Motors of the In-Line Type

The compression-disk shock absorbers vary somewhat from the tension-cord shock absorber, in so far as total travel is concerned, the travel being considerably less under impact than on the tension cords. As the load increases on the rubber disks, the travel reduces rapidly. Therefore, special care has to be taken in the design to prevent the shock of the impact from being transmitted to the fuselage. The energy absorbed in the compression-disk device is approximately 17 per cent. The performance of rubber shock absorbers can be improved by rebound checks, preventing the plane frame from bouncing.

Other uses of rubber in airplane construction are leakproof tanks. These are especially valuable in military airplanes. The construction of this tank lining is very similar to puncture-proof inner tubes. Several layers of rubber are put together in such a fashion that they are in compression and will seal holes of small size automatically. These leakproof tanks are also some insurance against crashes.

The time for general adoption in airplane designs of air-filled rubber-seat cushions should not be very distant. The light

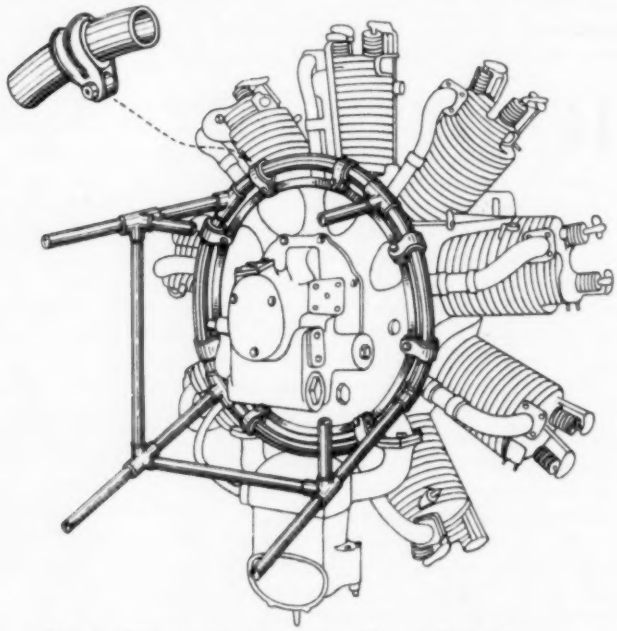


Fig. 3 Support Insulator for Radial-Type Engines

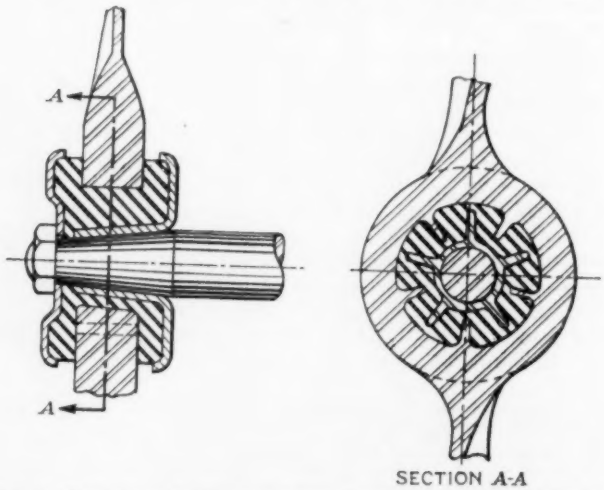


Fig. 4 Propeller Insulator Interposed Between the Motor Drive Shaft and the Propeller Hub

weight and wonderful comfort derived from such a seating arrangement, coupled with the life-saver qualities for seaplanes, will certainly open a field both for the airplane designer and the rubber manufacturer.

Another article which is now being developed in cabin planes is a rubber lining which will insulate the passengers from outside

noises and changes in temperature. Experiments and developments are conducted to reduce the weight and cost of such linings to make them practical for installation on planes.

A propeller insulator would also greatly help in the reduction of vibration. This would be a device similar to an automobile's torque joint, and would be interposed between the motor drive shaft and the propeller hub, as shown in Fig. 4. The long-desired rubber fender for automobiles might become a reality on airplanes as a streamline housing for wheels.

Aside from insulation of gasoline lines, cables, grommets, handle bars, and so on, where rubber is used on airplanes, there are many other points where it might be used, be it in solid or movable joints, in insulation of vibration, or for any other problems such as where steel is subject to crystallization from changeable stresses.

AER-53-5

Airplane Tires and Wheels

By H. F. SCHIPPEL,[1] AKRON, OHIO

This paper tells of some of the mounting conditions of airplane tires on wheels, and of the relation between static load, ultimate cushioned load, and energy capacity. Static and dynamic loading curves and corresponding hysteresis loops are also presented, and the forces and power losses are resolved horizontally and vertically. Results of extensive wind-tunnel tests show the effects of tire size, wheel size, safety-tread design, streamlining, and tire misalignment.

TIRE EQUIPMENT and performance cannot be discussed without including the wheel, and even the shock absorber, for these three principal parts of the airplane landing gear are closely interrelated. This paper discusses some of the accepted practices in connection with the application of pneumatic tires to airplanes and the performance of airplane tires.

Tire Mounting Conditions

The straight-side tire has now without question been accepted as the standard for the industry, although the changeover from clincher to straight-side tires has not yet been completely accomplished. The rim equipment is of the drop-center type, which has been selected on account of its strength and lightness having eliminated the necessity of a felloe band. A typical tire mounting is shown in cross-section in Fig. 1. The bead of the tire registers on the rim ledge, which has a 5-deg. slope and makes a close fit so that the inner tube can come in direct contact without danger of injury. The necessity of a flap is therefore eliminated. However, particular attention is paid to the design of the tire bead so that it does not overhang the side of the well, as shown in the center illustration, Fig. 1. This condition may possibly arise when oversize tires or extra-ply tires are used, due to the greater bead width, but usually there is no overhang. Whenever such overhanging exists, tube chafing is likely to occur, and this may eventually lead to a leaky tube and a flat tire. In cases where bead overhang is unavoidable, the tire may be furnished with bead flaps, as shown at the bottom of Fig. 1. These flaps are not entirely desirable, however, because they interfere with mounting of the tire.

Airplane wheels are of the wire-spoke and disk types. In the former the spokes are laced into the drop-center rim, the heads passing through the rim either on the bead ledge or in the well, or both, and sometimes on the side of the well. The ends of the spokes and nipples may be ragged, and it is important that the tube be kept from contact with them. For this purpose a rim band is provided, as shown in Fig. 1, which successfully covers the spokes on the bottom of the well, but not on the side of the well unless the rim band is made unusually wide, in which case it cannot with certainty be maintained in correct position due to possible disturbance when mounting the tire. An alternative solution lies in the use of bead flaps, but this practice, as mentioned before, is not wholly desirable.

Rim bands are not required on disk wheels, there being no danger in allowing the tube to make complete contact with the rim. It has been the practice of airplane-tire manufacturers to furnish rim bands with each tire, whether sold for original equipment or replacement, but by a new agreement among the tire and wheel manufacturers, these rim bands are furnished by the wheel manufacturers when necessary. A further advan-



Fig. 1 Arrangements of Tires on Rims

[1] Design Engineer, The B. F. Goodrich Co. Received degree of B.S. in Electrical Engineering, McGill University, in 1912. Entered service with Canadian General Electric Co. and in 1914 became lecturer in electrical engineering at McGill University. From 1917 to 1925 he served as Research Engineer, Canadian Consolidated Rubber Co.; Research Engineer, Ames-Holden-McCready; and Development Engineer, Ames-Holden Tire Co. Joined The B. F. Goodrich Co. in 1925.

Presented at the Akron Meeting, Akron, Ohio, Oct. 21 to 23, 1929, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

tage of this change in practice occurs in the case of oversizing, the rim band according to old practice being too large for the undersize rim.

Tire Load Ratings

The method of rating airplane tires for static load capacity is very similar to that used for passenger-car tires. The rating is made on a basis of radial deflection, calculated by the method

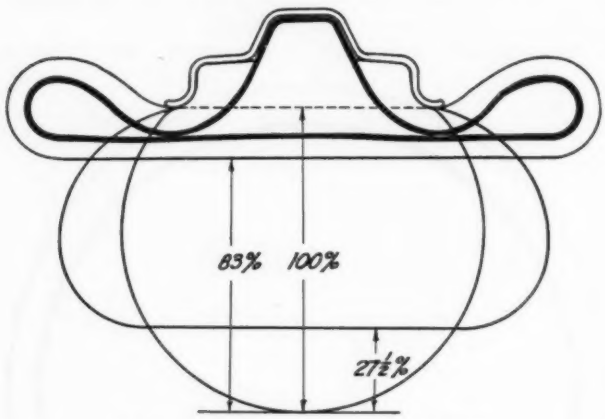


Fig. 2 Deflection of 30- by 5-In. Tire

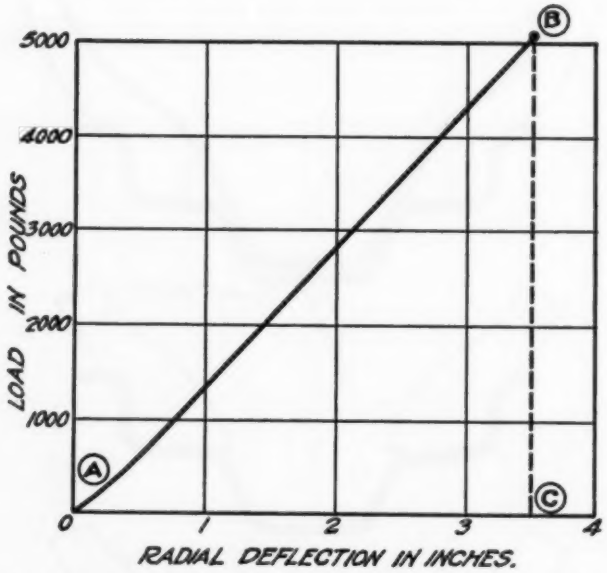


Fig. 3 Graph of Deflection of 30- by 5-In. Tire on 30- by 5-In. Rim at 50 Lb. Inflation

shown in the army specifications. This method rates tires according to a definite radial deflection taken as a percentage of the cross-sectional height of the tire measured from the point of extreme overall diameter to the top of the rim flange. The present line of tires is rated on the basis of a radial deflection equal to 27½ per cent of this distance. On this basis the tire cannot deflect 100 per cent, because twice the thickness of tire and tube intervenes between the rim flange and the ground at complete collapse of the tire. Fig. 2 shows this clearly. In the case of the 30- by 5-in. tire the maximum possible deflection is about 83 per cent. Fig. 3 shows a typical loading curve for an airplane tire, loaded up to the extreme deflection noted above, or collapse point. It will be noted that this tire, which is rated at 1600 lb. static load, has a maximum cushioned load capacity of 5100 lb. This indicates that the ratio of maximum to static load rating is 5100 ÷ 1600 = 3.2. But the maximum design load for the plane which takes the 30- by 5-in. tire is about 6 times 1600 lb. or 9600 lb. It is therefore quite obvious that under the conditions of maximum allowed impact of the plane upon the ground, the tire is not only completely flattened, but is pinched between the rim and the ground. This is not a highly desirable condition, and is only tolerated because apparently the tire has the ability to withstand this punishment, or because such a severe landing occurs but infrequently. The ideal condition for the tire would be to have its collapsing load equal to the static load multiplied by the load factor, but this would reduce the static rating of the 30- by 5-in. tire, for example, from 1600 lb. to 5100 ÷ 6 = 850 lb., or approximately one-half its present rating. An alternative and equally undesirable method of equalizing this condition would be to increase the

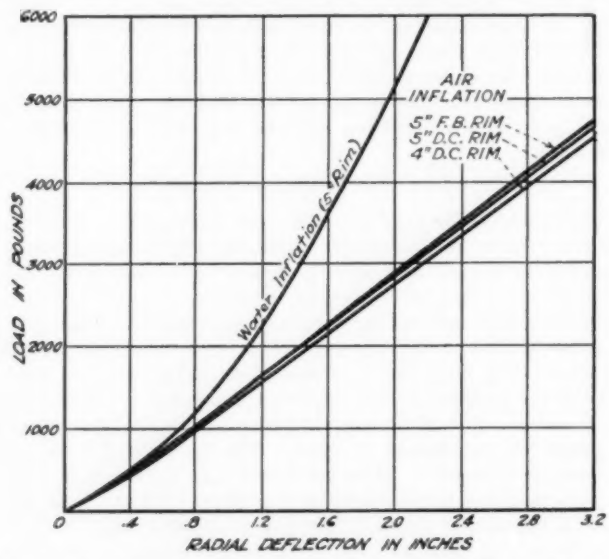


Fig. 4 Variation of Radial Deflection With Load—30- by 5-In. Tire on 30- by 5-In. Rim at 50 Lb. Initial Inflation—Air vs. Water

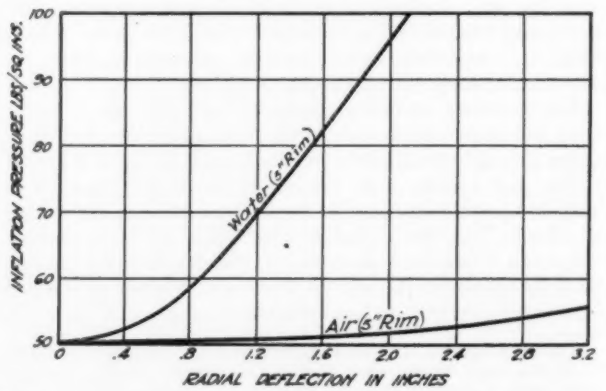


Fig. 5 Variation of Radial Deflection With Inflation—30- by 5-In. Tire on 30- by 5-In. Rim

collapsing load to the required value by increasing the inflation pressure to about 100 lb.

There seems to be an impression that the drop-center rim increases the cushioning properties of a tire because of the extra amount of air volume in the drop-center portion of the rim. This impression probably arises from the fact that the capacity of a

tire is often spoken of as proportional to its air volume. Although this is true in an approximate way, yet it is the active air space only that is of consequence in relation to tire cushioning, because it is only that air space that substantially affects the loading characteristics of the tire. Figs. 4 and 5 show in a diagrammatic way the significance of this statement. Fig. 4 shows the relation between load and radial deflection for a 30- by 5-in. airplane tire mounted on the regular drop-center rim and on a hypothetical flat-base rim having the same dimensions as the drop-center rim, except for the absence of the well portion. The latter curve was obtained by calculation on the following basis: When a tire is loaded, the portion of the tire in contact with the ground suffers a decrease in internal volume, and this produces an increase in inflation pressure. For example, a 30- by 5-in. tire, mounted on a 5-in. drop-center rim, and inflated to 50 lb., shows a pressure rise of 5 lb. when loaded to 5100 lb. The curve of pressure rise is shown in Fig. 5. The drop-center portion of the rim represents only 5 per cent of the total volume of air in the inner tube, and hence the pressure rise in the 30- by 5-in. tire when mounted on a hypothetical flat-base rim would be $5^1/_4$ lb., which would increase the ultimate load-carrying capacity of the tire by approximately 25 lb., which is almost

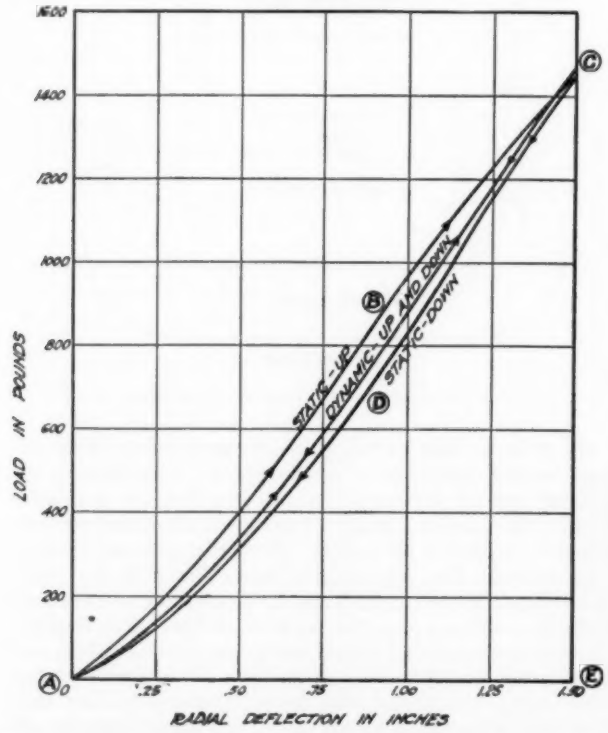


FIG. 6 STATIONARY AND ROLLING LOADING CURVES FOR 30- BY 5-IN. AIRPLANE TIRE ON 29- BY $4^1/_2$-IN. PASSENGER-CAR RIM—ROLLING TEST AT 50 M.P.H.—TIRE LOADED AGAINST DRUM 67 IN. DIAMETER

insignificant. However, the results serve to indicate that the increased air volume of the tire on the drop-center rim decreases rather than increases the load-carrying capacity, through the effect that air volume has upon the pressure rise under load. In order to show the full extent to which one can increase load-carrying capacity by decrease in air volume, a test was made with the air entirely replaced by water. The water being incompressible, the cushioning property of the tire was entirely dependent upon the elasticity of the cords in the tire carcass. The pressure rose from 50 lb. at no load to 96 lb. at 5100 lb. load, but at full radial deflection the tire was capable of supporting 10,000 lb., at which load the inflation pressure would have been 150 lb. The load capacity of a tire is also affected to a small extent by the width of the rim on which the tire is mounted. Fig. 4 shows the loading curves for the 30- by 5-in. tire on the 4-in. and 5-in. airplane rims.

ENERGY CAPACITY AND ABSORPTION

The simple and approximate method of measuring the energy capacity of a tire under radial deflection will be considered first. This method is the static test, in which the stationary

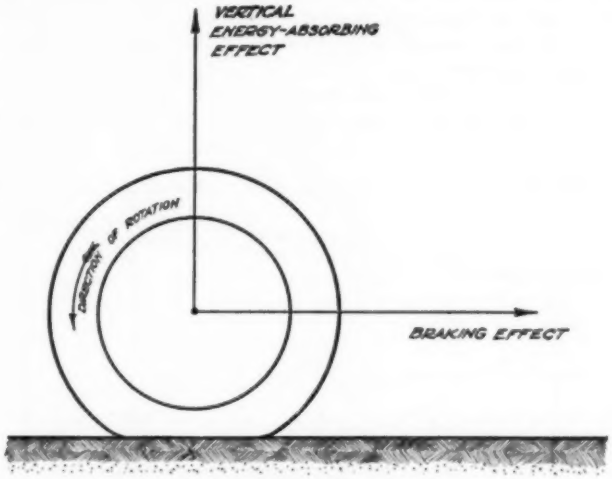


FIG. 7 ENERGY LOSS IN TIRE WHEN ROLLING ALONG THE GROUND

tire is loaded against a flat plate in a loading machine, and deflected until the tire is compressed between the rim and the contact plate. The curve in Fig. 3, for a 30- by 5-in. tire, was obtained by this method, the energy stored being represented by the area *ABC* lying under the loading curve. This energy is 8950 inch-pounds, and represents the ability of the tire to absorb energy in a vertical direction when the airplane makes a landing. It should more correctly be stated, however, that this only approximately represents the vertical energy capacity, because the curve was obtained by a loading test that was comparatively slow compared to the rate of deflection under service conditions. It is quite possible that the energy capacity may be greater under the condition of rapid loading. Unfortunately, the necessary equipment was not available to make the rapid loading test. However, a careful study of the conditions under which a tire is deflected against the ground in actual service will disclose the fact that even this rapid loading test would not give the true figures.

In actual service, the tire is not stationary, but rolls along the ground at a rapid rate, and the length of tire carcass loaded is not represented by the small section in instantaneous contact with the ground, but by the length equal to the distance traversed by the plane over the ground during the interval of contact. This condition was simulated in the laboratory by loading the tire against a large wheel revolving at a speed representative of the initial landing speed of an airplane, viz., 50 m.p.h. On account of the curvature of the drum it was necessary to obtain a new static loading curve for comparison with the dynamic curve. Fig. 6 shows the results obtained. The curve *ABC* represents the static test results for increasing load and *CDA* the curve for decreasing load, and the area *ABCDA* represents the static hysteresis loss for the complete loading cycle. It will be noted that the tire was not loaded to complete collapse, because at a speed of 50 m.p.h. the temperature rise due to internal friction in the tire during the length of time

required to make the test would have affected the inflation pressure and also the results. The static curves show that the amount of energy stored in the tire upon increasing load was 1047 in-lb., and the amount returned during decreasing load was 919 in-lb. The area of the hysteresis loop is therefore 125 in-lb., and so it may be said that the efficiency of the tire statically as an energy dissipator is 12.2 per cent. However, static loading tests on a Riehle machine have shown this efficiency to be only 6 per cent; so it is feared that the above static loading test may be in error on account of friction of the carriage on which the tire was mounted. However, the results are at least of qualitative interest. Fig. 6 also shows the results of the dynamic loading test at 50 m.p.h. The curve falls inside the static hysteresis loop, and it shows no hysteresis loop, since the curves of increasing and decreasing loads coincide. It may therefore be concluded that the efficiency of the rolling tire as an energy dissipator in a vertical direction is zero.

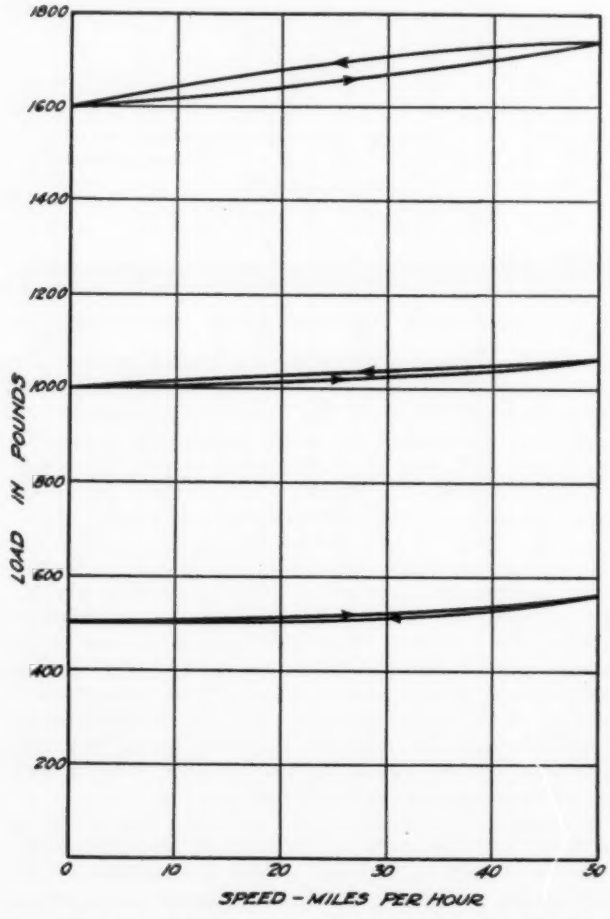
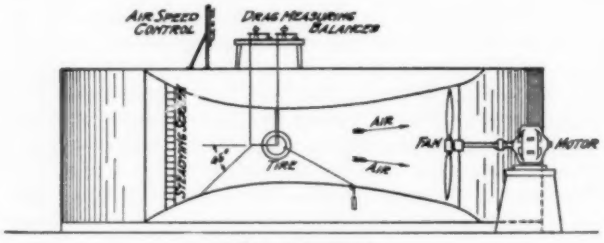


FIG. 8 DYNAMIC LOADING—30- BY 5-IN. AIRPLANE TIRE AT 50-LB. INITIAL AIR PRESSURE AND CONSTANT VERTICAL DEFLECTION

But since, during the interval of the single high-speed impact of the plane on the ground, the tire may have traveled over 100 ft. along its outside periphery, there must necessarily have been a considerable amount of energy absorbed due to the hysteresis of the tire carcass. If, then, there is all this absorption of energy why does the tire show no hysteresis loop on the dynamic loading curve? The answer is, that it is necessary to distinguish between effective braking action and "shock-absorbing" action in a tire, or horizontal and vertical components of energy absorption. (See Fig. 7.) In other words, the energy loss in a tire when rolling along the ground appears as a net horizontal, or braking action, and not vertical or "shock-absorbing" action. It is unfortunate that the search for the good qualities in airplane tires should have exposed such gross incompetence of the tire in the vertical direction. But so long as this creates a demand for rubber shock absorbers, the tire cannot be condemned for failure to support the rubber industry.



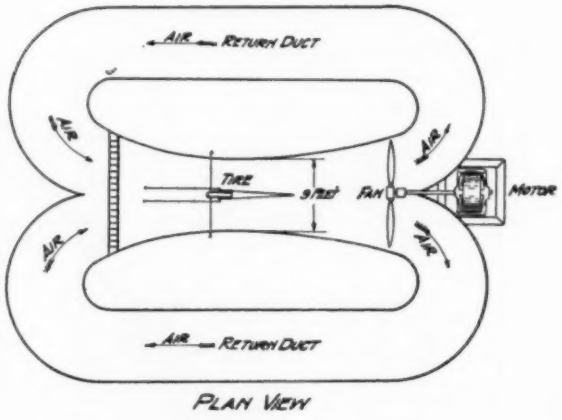


FIG. 9 DIAGRAMMATIC VIEWS OF WIND TUNNEL

Fig. 8 throws light on this particular performance of the tire from another angle. Here are presented curves showing the apparent increase in vertical reaction of a tire due to rotative speed. The increase is really a centrifugal effect, and could be expressed in terms of equivalent inflation-pressure increase, if so desired. The upper curve shows that if at the beginning the load is 1600 lb. with the tire at rest, this load increases to 1740 lb. at 50 m.p.h., or 8.6 per cent increase. This centrifugal effect increases the vertical energy capacity of a tire, and this increase would be very apparent if the dynamic loading curve in Fig. 6 is plotted with the point of zero load set back of the zero for radial deflection by $^3/_{32}$ in., which is the increase in the tire height at 50 m.p.h.

AIR DRAG

On account of the increasing use of anti-skid tires on airplane landing wheels, it is important that their comparative air-drag values become definitely known. It is also important that information be made available on other factors in connection with tire and wheel design, such as the efficiency of streamlining of the wheels and tires, and the effect of toe-in or out-of-alignment of the plane of the tire and wheel with respect to the line of flight.

With these objectives in view, The B. F. Goodrich Company, under the personal direction of Prof. Alexander Klemin, conducted tests in the 9-ft. wind-tunnel at the Daniel Guggenheim

School of Aeronautics of New York University, on various combinations of tires and wheels, as follows:

1. 14- by 3-in. smooth airplane tire on 14- by 3-in. wheel
2. 20- by 4-in. smooth on 18- by 3-in. wheel
3. 20- by 4-in. safety tread on 18- by 3-in. wheel

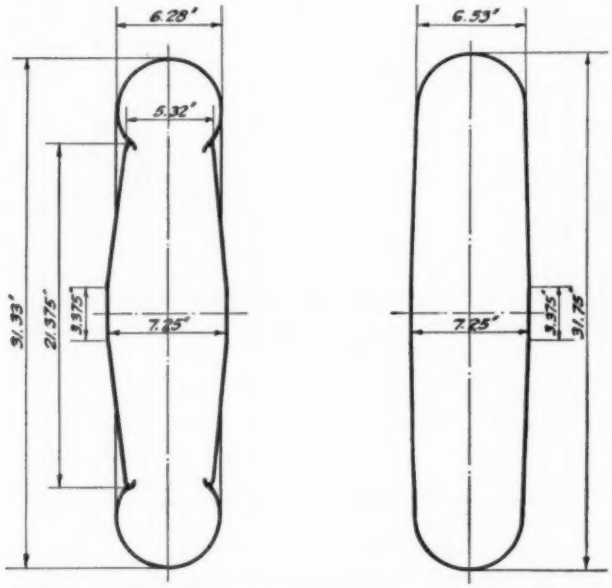


FIG. 10 AT LEFT SMOOTH AIRPLANE TIRE AND AT RIGHT SMOOTH STREAMLINE AIRPLANE TIRE 32- BY 6-IN. ON 32- BY 6-IN. WIRE WHEEL WITH METAL SHIELDS

4. 30- by 5-in. smooth streamline tire on 28- by 4-in. wheel
5. Same as 4 but toe-in 2 deg. 2 min. (0.35 in.)
6. Same as 4 but toe-in 4 deg. 4 min. (0.70 in.)
7. Same as 4 but toe-in 6 deg. 6 min. (1.05 in.)
8. 30-by 5 in. smooth tire on 28- by 4-in. wheel
9. Same as 8 but toe-in 2 deg. 2 min. (0.35 in.)
10. Same as 8 but toe-in 4 deg. 4 min. (0.70 in.)
11. Same as 8 but toe-in 6 deg. 6 min. (1.05 in.)
12. 30- by 5-in. safety tread on 28- by 4-in. wheel
13. 32- by 6-in. safety tread on 28- by 4-in. wheel
14. 32- by 6-in. smooth streamline tire on 32- by 6-in. wheel
15. 32- by 6-in. smooth on 32- by 6-in. wheel
16. 32- by 6-in. safety tread on 32- by 6-in. wheel
17. 6.00–20 (32 × 6.00) Silvertown passenger-car balloon tire on 32- by 6-in. wheel
18. 32- by 6-in. smooth airplane tire on 32-by 6-in. wheel without streamline disks on wheel
19. 36- by 8-in. smooth on 36- by 8-in. wheel
20. 40- by 10-in. safety tread on 36- by 8-in. wheel

Each combination was tested at 40, 50, 60, 70, 80, and 90 m.p.h., and the drag in pounds calculated for each air speed, as well as the drag coefficients based on actual cross-sectional area through the center of the axle, and on the projected area which is slightly greater. Fig. 9 shows diagrammatic plan and elevation views of the wind tunnel.

The wheels used were of the wire-spoke type, and sheet-metal fairings were used except where noted above. Detailed results are shown in the curve sheets, Figs. 11 to 16, inclusive, in which the numbers shown refer to the tests listed. The drag given is the net drag on tire and wheel, correction being made for the drag on the axle and connecting wires.

In order to calculate drag coefficients, the projected areas for each tire must be computed similar to the method shown on the attached sample drawing Fig. 10 for the 32- by 6-in. tire. The actual areas, as viewed in an aerodynamic sense, comprise the complete projection of each unit, and these areas are slightly larger than the cross-sectional areas. The method of calculating the drag coefficient is as follows:

$$\text{Drag coefficient} = \frac{\text{Drag in pounds}}{\text{Projected area in sq. ft.} \times \text{m.p.h.}^2}$$

The drag coefficient therefore numerically equals the drag per square foot of projected area for an air speed of 1 m.p.h. In the case of streamlined wheels the projected area was computed disregarding the increase in area due to the shields, as this makes the comparison of these wheels with ordinary ones clearer. No allowance for increase in area was made in the cases of toe-in.

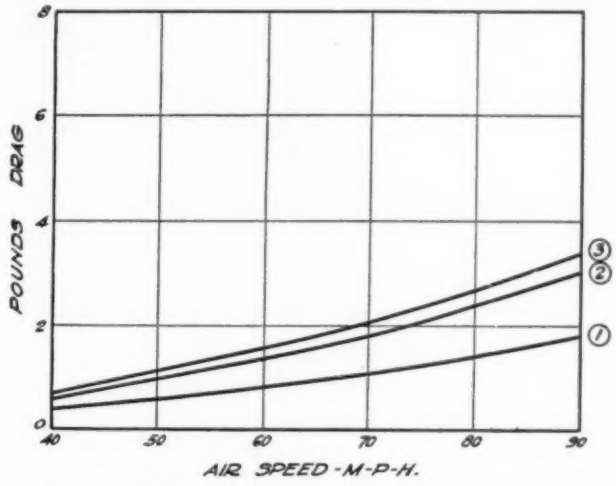


FIG. 11 AIR DRAG OF AIRPLANE TIRES NOS. 1, 2, AND 3

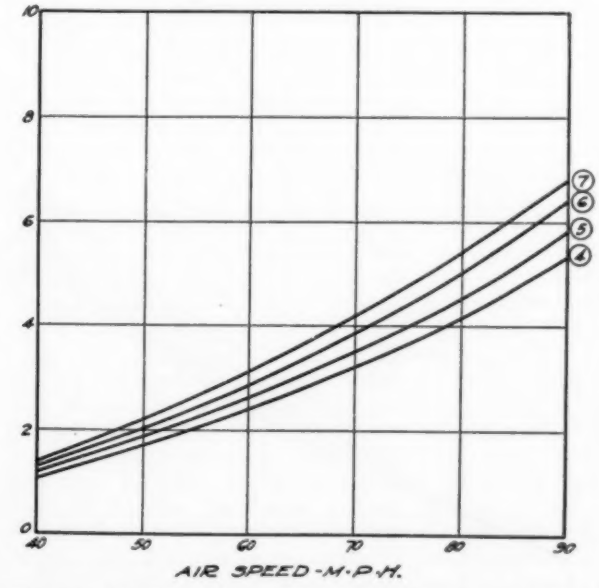


FIG. 12 AIR DRAG OF AIRPLANE TIRES NOS. 4, 5, 6, AND 7

DISCUSSION OF RESULTS

The drag coefficients have minimum values for 30- by 5-in. tires mounted on 28- by 4-in. wheels and slightly increase both

for smaller and larger wheels. The scale effect is the most pronounced in the case of smooth tires mounted on streamline wheels; there is a decrease of about 8 per cent in the value of drag coefficient from 40 m.p.h. to 90 m.p.h.

A comparison of drag on tires of different tread shows that the safety-tread tires have 5 per cent to 10 per cent greater drag than smooth ones. There is a saving in resistance of about the same magnitude by adoption of streamline tires. Streamline tires are still more advantageous in case they are set with toe-in, as the increase in drag due to the toe-in of 6.1 deg. is 45 per cent in the case of non-streamline tires, and only about 25 per cent in the case of streamline tires. This fact is of importance when it is considered that perfect alignment of the plane of the wheel with the path of flight may not always be easy to maintain in manufacture or service.

The test on the Silvertown 6.00–20 passenger-car tire shows that substitution of this kind of tire for an airplane safety-tread tire does not make a great difference from a drag standpoint, but the weight penalty is considerable. The resistance

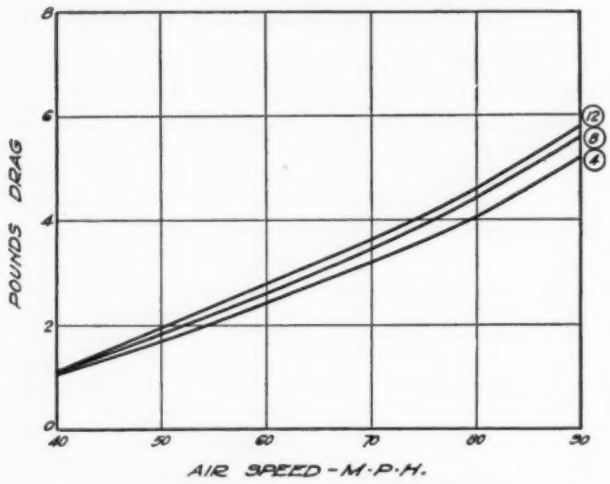


FIG. 13 AIR DRAG OF AIRPLANE TIRES NOS. 4, 8, AND 12

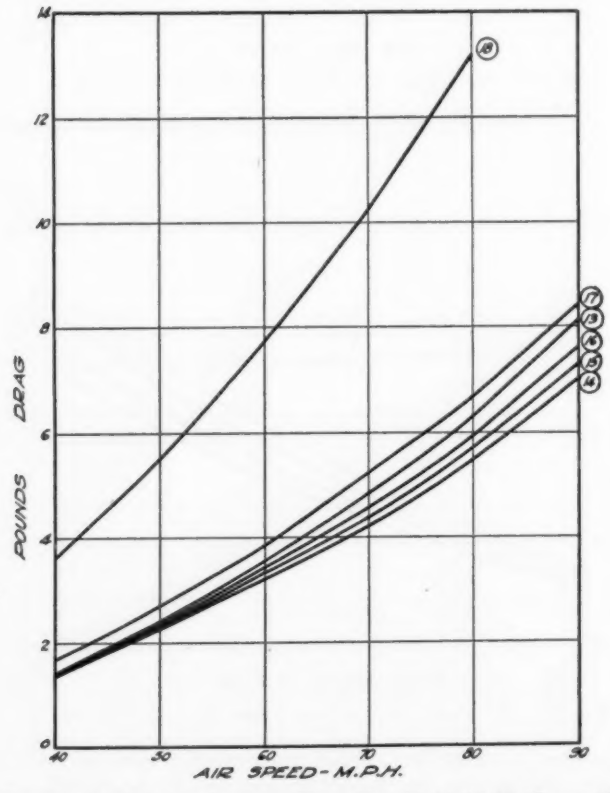


FIG. 15 AIR DRAG OF AIRPLANE TIRES NOS. 13, 14, 15, 16, 17, AND 18

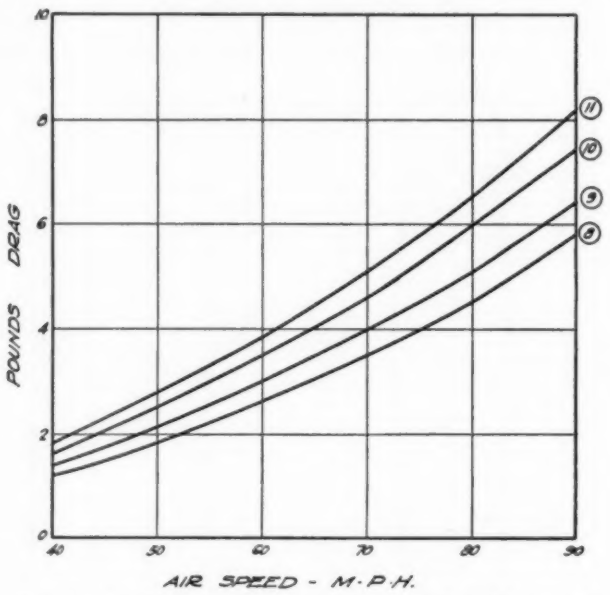


FIG. 14 AIR DRAG OF AIRPLANE TIRES NOS. 8, 9, 10, AND 11

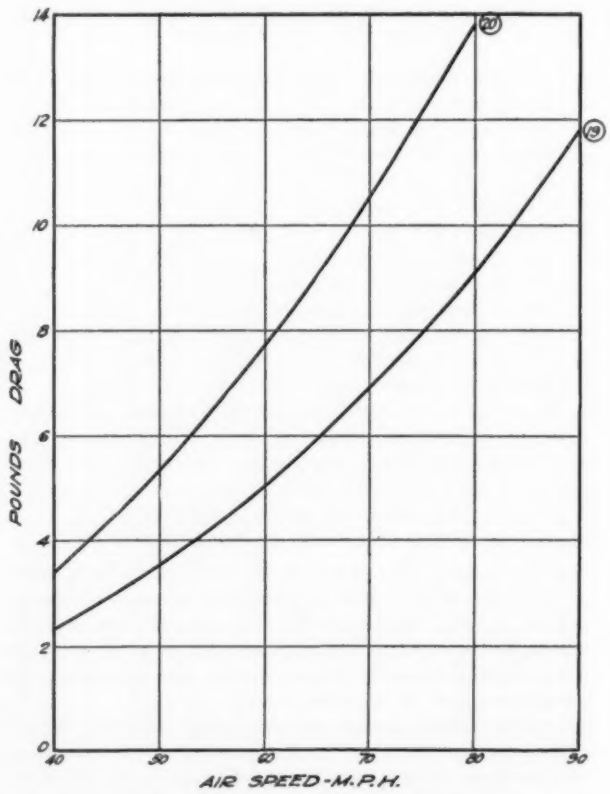


FIG. 16 AIR DRAG OF AIRPLANE TIRES NOS. 19 AND 20

of tire and wheel with the disks removed is about twice as great as the resistance with the disks on.

It is interesting to note the effect of oversizing upon the drag. For example, at 90 m.p.h. the 30- by 5-in. safety-tread tire on a 28- by 4-in. wheel shows a drag of 5.81 lb., while the 32- by 6-in. safety-tread tire on the same wheel shows a drag of 8.17 lb., an increase of 41 per cent. It is also interesting that the 32- by 6-in. safety tire has less drag when mounted on a 32- by 6-in. wheel than when on a 28- by 4-in. wheel.

Discussion

A. L. KIMBALL.[2] The writer does not agree with the author's conclusion that the hysteresis action is zero for a tire rolling on a surface for up-and-down vibratory motion. Although his dynamometer tests give a zero hysteresis loop, for the tire rolling against a revolving drum, it is to be noted that measurements are made when the up-and-down motion is zero. If measurements could be made while the tire was in motion during its up-and-down path a hysteresis loop would be obtained.

Referring to Fig. 17, when the tire is rolling against a moving surface as shown, the rubber in contact with the ground along AC, to the right of the vertical OA, is in the act of being compressed, while that along AB is in the act of expanding. This means that the frictional force which resists deformation produces an increased reaction on the ground from A to C, but a decreased reaction from A to B. This results in the couple shown by the pair of small arrows, which has to be overcome during rolling motion and accounts for the power absorbed in the tire as it rolls.

Assume, however, that a downward component of motion is superimposed on the horizontal. The median line OA then swings to the left slightly to a position OA'. The point of maximum compression is no longer directly under the hub at A, but swings over to A' due to the superimposed downward component of motion. This means that the zone of increasing compression is $A'C$ instead of AC, and that of expansion is $A'B$ instead of AB. Since more rubber is being compressed upward than is expanding downward, the right-hand member of the couple is greater than the left-hand, so that work must be done during the downward motion because of a resultant upward force to be overcome.

The following is an approximate calculation of the energy dissipation due to up-and-down vibration during rolling.

Assume, as an approximation, that when the tire is not rolling, but only moves up and down, the total hysteresis force is F. The work dW_v during an up or down motion through a distance dy is Fdy.

When the tire is rolling, a motion through this same element dy results in the work

$$dW_r = Fdy\left(\frac{2AA'}{BC}\right)$$

where as an approximation the force F is supposed to be uniformly distributed.

But $\frac{A'A}{AO} = \frac{V_y}{V_x}$ approximately, where V_x is the constant forward motion and V_y the instantaneous motion along dy.

Let the tire vibrate up and down according to the law $y = y_o \sin \omega$ where y_o is the half amplitude.

Then $V_y = \frac{dy}{dt} = y_o\omega \cos \omega t$

Also $dy = y_o \cos \omega t \, d(\omega t)$

[2] Research Engineer, General Electric Co., Schenectady, N. Y.

Substituting in equation for dW_r

$$dW_r = \frac{2Fry_o^2\omega \cos^2 \omega t \, d(\omega t)}{lV_x}$$

where $r = AO$ and $l = BC$.
Integrating over one cycle

$$\Delta W_r = \frac{2Fr}{lV_x} y_o^2\omega \oint \cos^2 \omega t \, d(\omega t) = \frac{2\pi Fry_o^2\omega}{lV_x}$$

which is approximately the energy dissipated during one up-and-down vibration. The energy dissipated during one cycle for the static test with no wheel rotation $= \oint F dy = 4Fy_o$ approximately, since the amplitude is assumed relatively small so F is approximately constant, but reverses in sign at the top and bottom of the cycle. The symbol y_o represents the half amplitude as before.

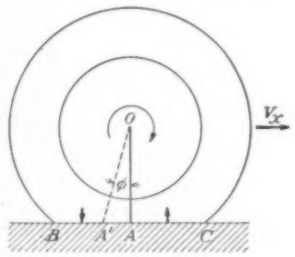


FIG. 17

The ratio of the energy dissipated during up-and-down motion in the static case to that in the case of the rolling wheel is:

$$\frac{\Delta W_v}{\Delta W_r} = \frac{lV_x}{\pi^2 r y_o f}$$

where $2\pi f$ is substituted in place of ω, f being the frequency of the up-and-down vibration.

If $l = 5$ in., $V_x = 500$ in. per sec., $v = 15$ in., $f = 10$ cycles per sec. and $y_o = 1/7$ in.

$\frac{\Delta W_v}{\Delta W_r} = \frac{20}{3}$, that is, the energy dissipated due to up-and-down motion during rotation is only about $1/7$ that without rotation.

The formula shows that ΔW_r increases with the frequency so the above ratio becomes smaller as the frequency of up-and-down vibration increases. ΔW_r decreases, however, with V_x, giving a correspondingly larger value of the ratio.

A remarkable feature of this result is the increase of W_r with frequency. The law of hysteresis assumed is that the dissipation per cycle per unit volume of the rubber is independent of frequency, that is, that a viscosity which would cause a viscous reaction which increases with velocity of strain, is not present. The hysteresis cycle is like that of magnetic hysteresis in iron, where frequency does not enter. It is for this reason that the force F is taken as a constant. A consideration of the fundamentals employed above will show that an equivalent rectangular loop is employed. Since we are interested in loop area, the shape is immaterial for the purpose of analysis given.

This law of independence of loop area of frequency has been studied by the writer and has been found to hold for a variety of solids, including various metals, wood, celluloid, and rubber.[3] It is possible, however, that in the case of rubber, with strain velocities as high as they are in a rapidly rolling wheel, a strain velocity effect may enter, which increases the loop area. Experiment shows that loop area depends upon the strain amplitude squared. In the above case, where during the up-and-down vibration the strain is relatively small compared with the entire range, the force F is assumed constant and independent of amplitude, which gives a loop area proportional to the first power of the amplitude.

Although this discussion is rather inadequate, it at least gives an idea of what happens, and brings out the important

[3] A. L. Kimball and D. E. Lovell, "Internal Friction in Solids," *Physical Review*, Dec., 1927.

fact that the higher the frequency of bouncing the greater the dissipation and the higher the speed V_x the less the dissipation for a given amplitude. A complete analysis would be very complicated. It is thus seen that with very high rolling speeds indeed, the hysteresis dissipation due to up-and-down vibration may be relatively small, although it can never be actually zero.

Aviation Natural Gasoline

By R. C. ALDEN,[1] BARTLESVILLE, OKLA.

The usual natural gasoline of commerce is an extremely volatile product and quite unsuitable for direct use as a motor fuel in the conventional fuel systems of today. By careful fractionation, however, it has been found possible to produce "aviation natural gasoline," a fuel that has made a splendid record commercially during the last three years. A considerable amount of research work has been done with regard to this product, and the author discusses it under the following heads: (1) Initial flight tests to determine specification bases for commercial purposes; (2) effects of changes in specifications for aviation natural gasoline; (3) survey of commercial aviation gasolines; (4) flight tests to determine actual temperature in airplane fuel systems; and (5) other investigations. An extensive table of test results of commercial aviation gasolines is included, as well as a bibliography of articles having a bearing on aviation natural gasoline.

NATURAL gasoline is the name applied to the gasoline recovered from natural gas. Formerly, the product was called "casing-head" gasoline from the fact that the first natural gasolines were recovered entirely from the gas withdrawn through the casing heads of oil wells.

The present daily production of natural gasoline amounts to more than six million gallons, which represents approximately twelve per cent of the total gasoline supply. The natural gas from which natural gasoline is obtained may contain before treatment as little as one-tenth to as much as twenty gallons of commercial natural gasoline per thousand cubic feet. The average richness of gas treated for natural gasoline is about one and one-quarter gallons per thousand cubic feet.

Natural gasoline has been used almost exclusively as a blending agent, which, when added to the refinery product in suitable proportions, greatly improved the ease of starting and accelerating characteristics of the finished motor fuel. For this purpose high volatility of the natural gasoline was required, inasmuch as the product had to be shipped from its points of manufacture to the refinery or other point of use, thus increasing its costs.

During the past five years developments in the refining industry have made it possible for the refiners to obtain increased volatility without the use of natural gasoline. The use of vapor-recovery systems and the oil-cracking process has been responsible for this condition. Fortunately, for the natural-gasoline industry, competition in the marketing end has been so keen that this progress has been passed on to the public in the form of better motor fuels. It is safe to say that if the motor-fuel standards of 1923 were in effect today there would be very little market for natural gasoline.

An interesting sidelight on this development of gasoline quality is shown in Table 1, which shows the "front end" volatility of the composite gasolines reported in the Semi-annual Motor Gasoline Surveys of the U. S. Bureau of Mines.

Although Table 1 is a study of motor gasolines, it has a very definite bearing on the subject of aviation gasolines. The trend toward more volatile fuels is quite evident, and a knowledge of the economic conditions surrounding the petroleum industry leads to the conclusion that these volatile fuels have come to stay for a considerable length of time. It therefore follows that fuel systems for automotive and airplane engines should be designed with this trend in mind.

Because refineries are making such rapid progress in imparting volatility to their finished gasolines in their own processing, the more progressive manufacturers of natural gasoline have undertaken extensive research programs in order to assure markets for their product. One of the most interesting of the developments resulting therefrom has been aviation natural gasoline.

The usual natural gasoline of commerce is an extremely volatile product, quite unsuitable for direct use as a motor fuel in the conventional fuel systems of today. A typical Grade AA natural gasoline has a vapor pressure at 100 deg. fahr. of 28 lb. per sq. in. abs. It contains about 1 per cent by volume of propane and 30 per cent of butanes. The vapor pressures at 100 deg. fahr. of propane, isobutane, and normal butane are 186, 74, and 53 lb. per sq. in., respectively. When these volatile constituents are removed, the residual product has a vapor pressure at 100 deg. fahr. of approximately 11 lb. per sq. in. Further fractionation of the "butane-free" product, at the expense of the pentanes content, can be employed to reduce the vapor pressure further. This residual kettle product, resulting from the careful fractionation of natural gasoline, composes the product known as aviation natural gasoline.

TABLE 1 "FRONT END" VOLATILITY OF COMPOSITE GASOLINES REPORTED IN THE SEMI-ANNUAL MOTOR-GASOLINE SURVEYS OF THE U. S. BUREAU OF MINES

	Winter					Summer				
Year	Per cent recovered	First drop, deg. fahr.	10% condensed, deg. fahr.	10%[1] evap., deg. fahr.	Reid[2] vapor pressure	Per cent recovered	First drop, deg. fahr.	10% condensed, deg. fahr.	10%[1] evap., deg. fahr.	Reid[2] vapor pressure
1920	96.6	119	172	161	6.85	96.7	130	182	173	5.80
1921	95.9	113	168	165	7.90	96.7	125	177	167	6.15
1922	96.0	102	168	154	8.20	96.8	121	176	165	6.55
1923	96.0	107	167	153	8.10	96.8	125	179	170	6.20
1924	96.2	96	164	150	8.95	96.0	107	167	153	8.15
1925	96.2	95	161	147	9.30	96.2	108	169	156	8.05
1926	96.3	93	155	142	9.40	96.7	100	162	152	8.55
1927	96.5	92	149	137	9.65	96.4	102	159	147	8.65
1928	96.5	90	146	134	9.95	96.1	100	156	143	9.00
1929	96.1	87	143	131	10.50	96.1	102	155	143	8.70
1930	96.6	93	139	129	10.25					

[1] A constant distillation residue of 1.0 per cent was assumed.
[2] Graphically determined by method described in the *Oil & Gas Journal*, May 15, 1930, p. 40.

The equipment used in the manufacture of aviation natural gasoline is the same as that used in the manufacture of commereial propane, butane, pentane, and the refinery grades of natural gasoline. The "wild" condensate extracted from the natural gas is fed into one column, where conditions of temperature, pressure, and rate of reflux are maintained to give the desired kettle product. The overhead vapors are liquefied and subjected to further refractionation.

Aviation natural gasoline thus produced is an exceptionally well-fractionated, or "stabilized," product. With the average raw material it is necessary to eliminate a portion of the pentane content in order to reduce the vapor pressure to the desired limits. For example, Table 2 can be taken as illustrative of the composition changes which take place in the processing of the "wild" condensate extracted from natural gas in the production of aviation natural.

[1] Assistant Director, Phillips Petroleum Co.

Presented at a joint meeting of the Chicago Section and Aeronautic Division of the A.S.M.E., Chicago, Ill., August 25, 1930.

TABLE 2 COMPOSITION OF FEED AND KETTLE PRODUCTS IN THE FRACTIONATION OF AVIATION NATURAL GASOLINE

(Composition in terms of volume per cent)

	Initial condensate	Grade AA natural	"Butane-free" natural	Aviation natural
Propane and lighter	14.0	1.0	..	..
Butanes	32.0	30.0	1.0	0.3
Pentanes	28.0	31.0	46.0	31.7
Hexanes	14.0	15.0	25.0	31.0
Heptanes and heavier	12.0	15.0	28.0	37.0

It is interesting to note in regard to the processing of aviation natural gasoline that the emphasis in fractionation is placed on the light ends. In other words, very careful control is maintained in regard to the vapor pressure of the finished product. This, of course, is due to the fact that the possible contaminants of the finished product are more volatile than the finished product itself. This is in direct contrast with usual refinery practice, where the possible contaminants are less volatile than the finished product, with the result that the emphasis in fractionation is placed on the heavy ends. With the ever-increasing supply of volatile hydrocarbons, it is becoming increasingly necessary to exercise careful control of what might be termed the "front-end" volatility of fuels for internal-combustion engines.

Aviation natural gasoline is a comparatively new product. In its present form it has been on the market a little over three years. In this short time it has made a splendid record commercially. It is estimated that in 1929 between 15 and 20 per cent of the gasoline used in commercial flying was aviation natural. Among the more spectacular achievements to its credit can be cited its use by Col. Arthur Goebel in winning the Dole race from Oakland, Calif., to Hawaii, and the 1929 endurance flight of the *St. Louis Robin*.

A considerable amount of research work has been done in behalf of aviation natural gasoline. This work can be divided into five headings, each of which will be briefly discussed.

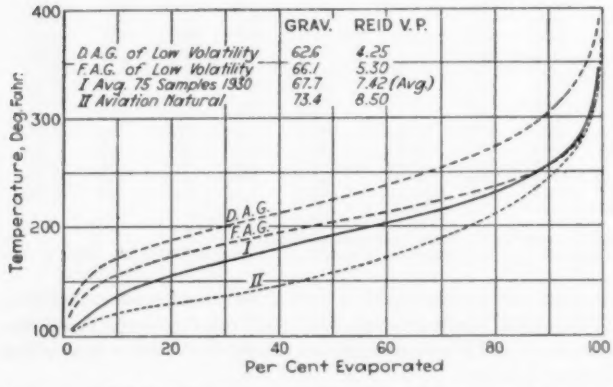


FIG. 1 A.S.T.M. DISTILLATION CHARACTERISTICS OF AVIATION GASOLINES

1 INITIAL FLIGHT TESTS TO DETERMINE SPECIFICATION BASES FOR COMMERCIAL PURPOSES

Fig. 1 shows the A.S.T.M. distillation characteristics of refinery products which meet the requirements of domestic and fighting grades of aviation gasoline as compared with a typical aviation natural. The significant characteristics to note are:

a The remarkably low mid-portion of the distillation curve of aviation natural. This signifies superior distribution to the cylinders of the engine and excellent cold-weather performance.

b The low 10 per cent evaporated temperature, which signifies easy starting, particularly in cold weather.

c The flat "front end" of the curve. This signifies the absence of light hydrocarbons and means that the product has a very low vapor pressure as compared with products not having this characteristic.

Because of these differences, each manufacturer who has launched an aviation natural program has considered it advisable to conduct actual flight tests prior to placing his product on the market. In the flight tests, a wide variety of gasolines has been used, ranging from Grade AA natural in extreme volatility to very deeply cut butane-free naturals. The most rigorous flying

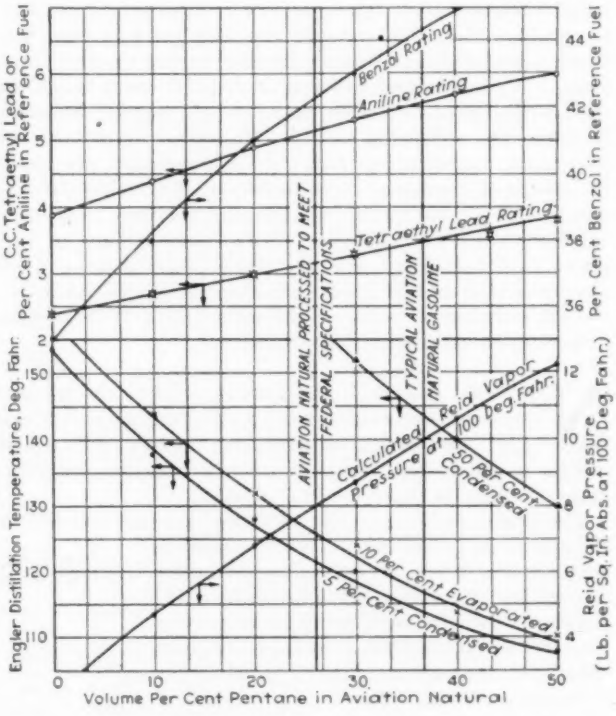


FIG. 2 EFFECT OF PENTANE CONTENT ON SEVERAL SALIENT CHARACTERISTICS OF NATURAL GASOLINE

conditions were imposed, particularly in regard to extreme ground temperatures, rate of climb, and sustained flight.

In this test work it was found that even the more volatile products could be used under many conditions. In order to be on the safe side and to avoid undue handling losses, most manufacturers established a maximum Reid vapor pressure limit of 10.5 lb. per sq. in. at 100 deg. fahr. Inasmuch as this is a shipping specification, the vapor pressure at points of consumption will be somewhat lower than 10.5 lb. per sq. in. In terms of the A.S.T.M. distillation, this means that the 5 per cent condensed temperature will vary through the approximate range of 112 to 119 deg. fahr. and the 50 per cent condensed temperature will vary through the approximate range of 140 to 154 deg. fahr.

It is worth noting at this point that the existing specifications for F.A.G. (Fighting Aviation Grade) and D.A.G. (Domestic Aviation Grade), formulated eight years ago, require a minimum 5 per cent condensed temperature of 122 deg. fahr. Consequently, in this one respect the conventional aviation natural does not meet the federal requirements. On the other hand, the maximum temperatures at 50 per cent condensed for D.A.G. and F.A.G. are 221 and 203 deg. fahr., respectively. Consequently, in this regard, aviation natural is much superior to the fuels which barely meet federal requirements.

Besides the specifications which have resulted from the initial flight tests, several facts were developed of general interest. No noteworthy differences were found as to fuel consumption from

a volumetric viewpoint as compared with other gasolines. The consumption when expressed in terms of weight did, however, show an advantage in favor of the lighter aviation natural, this difference being worthy of consideration.

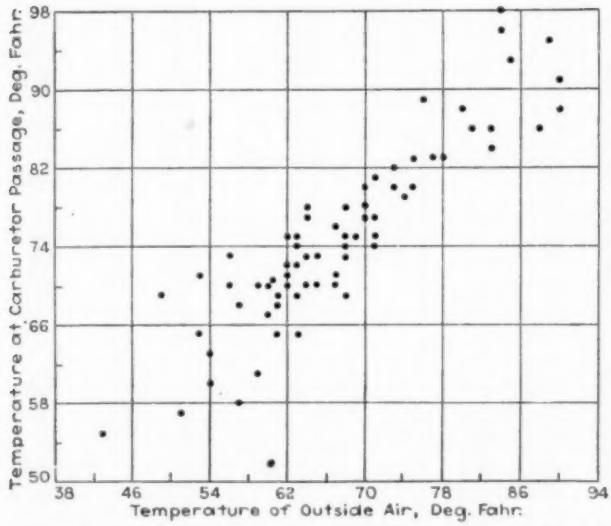


FIG. 3 FUEL-SYSTEM TEMPERATURE IN WASP-ENGINED TRAVELAIR MONOPLANE

2 EFFECTS OF CHANGES IN SPECIFICATIONS FOR AVIATION NATURAL

As has been stated, aviation natural is a natural gasoline product with the butanes and lighter eliminated and some of the pentanes removed. In readjusting the product to consider other specification bases than those ordinarily in practice, the control is effected by varying the pentane content.

In Fig. 2, the pentane content of a typical Mid-Continent base material is the abscissa. The effect of this variable on several salient characteristics is represented by the various curves. The most important fact to be noted from Fig. 2 is the decrease in anti-knock rating with decreased volatility as measured by the several methods depicted.

From Fig. 2 the conclusion is readily reached that any volatility limitation for aviation gasoline that is not founded on the soundest engineering data causes a needless sacrifice of anti-knock value.

For purposes of comparison of anti-knock ratings it can be stated that the reference fuel used in the tests shown on Fig. 2 required 7.3 per cent aniline to equal the "Ethyl Standard" for motor gasoline prior to July 1, 1930.

3 SURVEY OF COMMERCIAL AVIATION GASOLINES

Since November, 1929, the Research Department of the author's company has conducted a monthly survey of aviation gasolines in order to determine the quality of the products in commercial use. The samples for this survey have been collected at widely scattered airports. Extreme care was exercised in sampling and shipping the products. Upon arrival at the central laboratory the following tests were made: A.S.T.M. distillations, composition analyses, Reid vapor pressures, and anti-knock determinations.

For the benefit of those who may desire to study the results of this rather comprehensive survey, a detailed tabulation is presented in Table 3.

Some of the interesting facts developed in the course of the survey are as follows:

a Approximately half the samples examined are considerably more volatile at the A.S.T.M. distillation 50 per cent point than is required by federal specifications.

b From a composition viewpoint, the essential difference between aviation naturals and other aviation gasolines is that the latter usually have considerably more in the butane range and less in the pentane and hexane ranges.

c Several products not aviation naturals are almost identical in test characteristics with aviation naturals.

d A considerable proportion of the fuels examined failed to meet federal specifications in regard to the A.S.T.M. distillation 5 per cent condensed temperature.

e Products having Reid vapor pressures as high as 11 lb. per sq. in. at 100 deg. fahr. are on the market. Smooth curves drawn through a plot of Reid vapor pressure against the 5 per cent condensed temperature show that it is easily possible to procure gasolines of 10.5 lb. vapor pressure, and even as high as 11.0 lb., under existing F.A.G. and D.A.G. specifications.

f The 10 per cent evaporated temperature in the A.S.T.M. distillation is not indicative of the Reid vapor pressure.

g Although there is a lack of uniformity among the several marketers of aviation natural gasoline, as to volatility specifications, one manufacturer has demonstrated a decided ability to maintain his product at a satisfactory high level of uniformity.

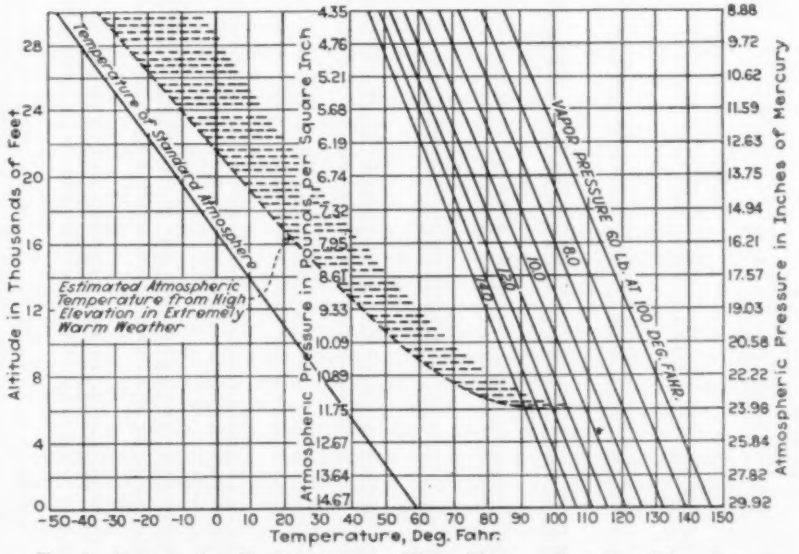


FIG. 4 ALTITUDE AND TEMPERATURE AT WHICH VARIOUS GASOLINES WILL BOIL

In order to demonstrate the rapid increase in the average volatility of aviation gasolines, Curve I was added to Fig. 2. This curve represents the composite gasoline resulting from the survey described. It is to be noted that the average commercial aviation gasoline, as revealed by the survey, has considerably lower boiling points, that is, more volatility than required by D.A.G. or F.A.G. specifications. When it is recalled that a few years ago practically all marketers of commercial aviation gasoline adhered very closely to the maximum boiling points of federal specifications, it will be realized that the period has witnessed a considerable change in the quality of aviation gasolines. Curve II of Fig. 2 permits a comparison of aviation natural gasoline with past and present practice.

This trend toward greater volatility is naturally of importance

TABLE 3 TEST RESULTS OF SURVEY OF COMMERCIAL AVIATION GASOLINES

	A.S.T.M. Distillation Data[2]												Composition[5]				
	Temperatures, deg. fahr.					Percentages			Evap. at	Gravity °A.P.I. at	R.V.P.[3] at	A.K. Rating[4]		Volume per cent			
Identification[1]	1st drop	5% cond.	10% evap.	90% evap.	End-point	Rec.	Res.	Loss	212° F.	60° F.	100° F.	212° F.	350° F.	C_3 and light	C_4	C_5	C_6	C_7 and heavy
1-17-12	110	140	151	289	358	98.3	0.6	1.1	50.0	64.2	6.2	8.7	...	...	2.2	12.2	21.3	64.3
1-17-7	108	140	151	288	357	98.0	0.8	1.2	50.0	64.2	5.6	9.6	8.7	...	...	...	...	...
1-4	108	140	148	288	350	98.1	0.7	1.2	51.4	64.2	5.9	9.0	...	...	3.2	11.4	19.1	66.3
2-20-11	108	129	137	243	334	98.5	0.5	1.0	77.0	69.8	7.8	5.8	...	0.3	3.4	16.9	30.4	49.0
2-20-2	104	133	139	244	370	97.9	0.5	1.6	73.0	68.6	7.9	4.7	...	0.3	3.0	9.8	24.5	62.4
2-5-12	114	140	146	219	282	98.0	0.3	1.7	84.9	68.0	6.8	5.7	...	...	3.0	12.1	18.4	66.5
2-5-4	112	138	146	218	290	98.5	0.6	0.9	87.0	68.2	6.8	5.5	...	...	2.9	12.4	25.1	59.6
3-14-11	90	120	137	325	368	98.0	0.6	1.4	36.0	61.7	8.4	5.4	...	trace	6.1	12.2	14.1	67.6
3-14-12	92	123	136	323	362	97.9	0.5	1.6	35.3	61.8	7.9	5.3	...	...	5.5	11.5	13.8	69.2
3-14-3	96	124	137	328	384	97.4	1.1	1.5	33.5	61.4	7.9	5.0	...	0.3	5.6	10.4	13.0	70.7
4-14-11	95	128	140	304	372	98.0	0.7	1.3	42.0	64.9	7.3	3.8	...	0.3	5.3	9.9	17.5	67.0
4-14-1	100	134	143	302	389	97.4	0.5	2.1	44.0	65.2	8.0	3.2	...	...	5.3	10.0	19.4	65.3
4-14-5	104	132	140	254	324	98.0	0.5	1.5	74.0	69.7	7.4	4.4	...	...	4.1	13.8	24.4	57.7
4-14-7	107	141	154	350	402	98.0	0.6	1.4	31.5	60.9	5.8	3.1	3.0	...	...	...	...	...
5-20-1	91	118	124	336	396	95.3	0.6	4.1	40.0	62.3	10.4	6.4	..	0.9	6.7	11.7	14.8	65.9
5-20-4	97	122	130	284	362	97.5	0.7	1.8	58.0	68.6	8.0	3.0	...	...	3.0	17.8	19.8	59.4
5-20-7	110	140	152	325	382	98.0	0.7	1.3	36.0	63.5	5.9	2.0	2.0	...	...	...	...	...
6-8-11	91	119	130	266	330	97.2	0.6	2.2	54.0	70.1	11.2	4.4	...	0.8	7.9	12.8	14.5	64.0
6-19-12	102	127	135	232	303	98.1	0.2	1.7	78.0	69.9	8.3	3.7	...	0.6	4.2	12.2	18.3	64.7
6-19-2	114	144	151	243	324	98.0	0.7	1.2	71.0	68.3	5.9	4.1	...	...	2.1	8.4	30.6	68.9
6-19-4	98	130	138	240	332	98.0	0.5	1.5	74.0	69.5	8.0	4.2	...	...	4.4	14.8	21.8	59.6
6-19-6	99	121	127	238	324	97.0	0.6	2.4	75.0	71.0	8.6	3.9	3.9	...	4.7	13.9	24.9	56.5
7-5-12	109	138	145	236	339	98.1	0.6	1.3	77.4	67.3	6.8	5.2	...	...	3.2	12.5	19.8	64.5
7-5-3	112	139	148	286	387	98.6	0.7	0.7	64.0	65.1	5.9	4.6	...	0.2	2.4	11.7	25.1	60.6
8-25-11	105	120	124	245	375	98.8	0.7	0.5	77.5	71.9	8.2	5.9	...	...	1.3	31.2	29.0	39.5
8-14-2	126	160	170	304	390	98.0	0.8	1.2	40.0	62.6	4.0	2.3	...	...	1.1	6.0	9.9	83.0
8-14-5	114	145	153	276	380	98.0	1.0	1.0	62.0	66.9	5.2	3.2	...	...	1.9	10.4	24.6	62.1
9-13-11	93	106	109	237	342	98.0	0.5	1.5	84.0	75.5	11.6	6.2	...	...	5.2	33.6	28.1	31.1
9-19-1	92	105	109	223	330	98.7	0.2	1.1	86.6	76.8	11.8	5.9	...	...	4.4	39.9	26.7	29.0
9-19-3	101	117	122	244	354	97.9	0.9	1.2	78.0	72.8	8.8	5.1	...	...	2.2	28.7	22.2	46.9
9-19-5	97	118	124	245	342	97.9	0.8	1.3	78.0	71.9	8.3	4.9	...	...	• 3.1	23.9	30.0	43.0
9-19-7	98	115	120	233	329	98.0	0.7	1.3	83.0	74.5	9.2	5.2	5.1	...	...	...	...	...
10-12-1	102	130	140	279	356	97.8	0.5	1.7	54.0	67.8	8.0	3.9	...	...	5.4	12.6	20.7	61.3
10-12-4	100	138	150	334	396	97.5	0.8	1.7	35.0	62.2	5.9	3.2	...	...	4.0	8.4	14.4	73.2
10-12-6	98	120	124	317	372	95.4	0.9	3.7	46.0	65.3	9.7	5.0	5.0	...	5.6	13.9	15.5	65.0
11-1-2	97	115	120	246	372	98.0	0.9	1.1	79.2	74.1	8.8	4.8	...	...	1.3	29.7	25.9	43.1
11-1-3	106	120	124	242	373	98.0	0.8	1.2	80.1	74.3	8.5	4.7	...	...	1.4	30.5	27.7	40.0
11-1-4	102	119	123	247	380	98.1	0.9	1.0	79.3	73.5	8.2	4.1	...	...	1.4	29.4	30.4	38.8
11-1-6	103	118	122	251	372	98.0	0.4	1.6	79.8	73.4	8.5	4.4	4.4	...	1.2	29.6	26.0	43.2
11-1-7	101	120	126	264	399	98.2	1.1	0.7	75.5	62.6	8.4	8.2	8.1	...	...	...	...	...
11-25-6	104	120	124	235	330	97.2	1.3	1.5	83.0	74.6	8.6	8.9	8.7	...	1.0	30.8	28.3	39.9
11-25-7	106	122	126	230	314	97.5	1.2	1.3	84.0	74.9	8.4	9.0	8.7	...	...	...	...	...
12-8-11	116	154	168	285	368	99.0	0.5	0.5	41.0	60.5	5.6	6.1	...	...	2.3	8.2	19.0	70.5
12-8-12	98	134	145	274	346	97.1	0.7	2.2	50.6	62.6	8.5	6.4	...	...	4.3	10.5	14.2	71.0
12-8-3	102	140	151	281	354	98.0	0.6	1.4	49.0	61.9	6.9	6.5	...	0.8	3.3	9.0	12.7	74.2
12-20-11	102	136	150	282	355	98.5	0.6	0.9	48.0	62.3	8.4	6.5	...	0.8	4.6	8.5	17.7	68.4
12-20-2	100	142	151	278	350	97.0	0.7	2.3	48.0	61.5	8.0	6.7	...	0.8	3.1	2.7	18.2	75.2
12-20-5	100	142	151	276	354	97.2	0.6	2.2	48.5	61.1	6.8	6.4	...	...	4.2	8.2	16.5	71.1
12-5-2	104	142	155	275	337	98.1	0.3	1.6	47.7	61.7	6.9	6.3	...	...	4.1	7.5	16.1	72.3
12-5-4	120	156	167	278	363	98.5	0.6	0.9	44.6	65.2	4.9	6.2	...	...	2.9	5.6	14.3	77.2
12-5-7	115	150	164	281	354	98.4	0.6	1.0	45.0	60.7	6.2	6.5	5.3	...	...	...	...	...
12-22-6	101	124	133	282	358	97.2	0.8	2.0	53.0	61.5	8.1	6.4	6.0	...	5.2	8.5	13.6	72.7
13-9-11	121	157	169	280	340	99.0	0.3	0.7	44.5	61.2	5.0	6.4	...	...	2.8	5.2	15.6	76.4
13-9-12	119	159	171	270	333	98.5	0.5	1.0	44.6	61.4	5.2	6.3	...	...	2.7	4.9	16.6	75.8
13-9-2	118	158	168	272	333	98.1	0.5	1.4	47.0	61.3	4.9	6.3	...	...	2.7	6.3	9.8	81.2
13-9-4	120	160	172	272	332	99.0	0.6	0.4	43.0	61.1	4.9	6.5	...	...	2.9	5.2	14.9	77.0
13-26-6	126	161	173	270	345	98.6	0.6	0.8	42.5	66.2	5.8	6.3	6.3	...	2.2	7.1	10.9	79.8
14-20-11	101	133	149	276	330	97.8	0.5	1.7	44.0	64.9	7.4	4.2	...	0.4	4.2	9.8	18.3	67.3
14-20-2	102	137	147	276	346	97.6	0.6	1.8	47.0	65.2	7..	4.0	...	0.5	4.4	8.6	11.9	74.6
14-28-11	106	138	149	288	360	97.5	0.6	1.9	44.0	64.9	6.0	4.2	...	0.5	4.0	10.5	17.2	67.8
14-13-12	105	138	149	279	350	97.8	0.6	1.6	46.6	65.2	7.1	3.5	...	0.5	4.1	9.3	16.5	69.6
14-13-6	110	146	157	317	392	97.6	1.0	1.4	37.0	62.5	6.1	3.7	3.3	...	3.6	7.8	13.1	75.5
14-7-1	104	140	150	274	336	98.1	0.2	1.7	46.0	64.0	6.7	4.3	...	trace	4.6	9.3	17.5	68.9
14-7-7	109	141	152	272	287	97.9	0.5	1.6	46.0	65.4	6.9	3.9	4.1	...	...	...	...	...
14-16-1	100	132	141	301	380	97.4	0.3	2.3	42.0	64.5	8.3	4.0	...	0.6	5.3	10.1	15.6	68.4
14-16-5	97	135	147	276	338	97.9	0.6	1.5	47.0	64.9	6.7	4.0	...	...	4.8	9.0	15.7	70.5
14-19-1	104	141	152	281	352	98.0	0.5	1.5	44.0	64.3	6.9	3.7	...	...	4.5	8.5	17.2	69.4
14-15-3	111	137	145	282	345	97.1	0.5	2.4	46.3	65.0	5.9	3.6	...	...	3.6	8.9	14.1	73.4
14-14-4	104	140	151	277	344	97.9	0.5	1.6	45.0	64.2	6.3	3.9	..	0.4	3.4	8.1	15.2	72.9
14-5-5	103	138	151	273	338	97.9	0.6	1.5	48.0	64.3	6.5	4.0	...	...	4.4	9.0	16.9	69.7
15-19-1	117	137	141	214	330	98.7	0.4	0.9	88.0	70.8	6.7	4.7	...	...	0.9	16.9	39.6	43.4
15-19-3	123	137	142	231	322	96.8	0.5	0.7	84.4	71.6	6.4	4.2	...	...	0.4	18.3	29.9	51.4
15-19-5	108	129	134	233	322	98.2	0.8	1.0	84.6	72.2	7.4	4.2	...	...	1.3	20.7	40.4	47.6
15-19-7	112	132	137	236	331	98.2	0.7	1.1	81.5	72.3	7.2	4.3	4.0	...	...	...	...	...
16-8-11	94	116	126	232	312	98.5	0.5	1.0	76.5	71.8	11.4	5.6	...	0.5	7.3	17.0	21.5	53.7
16-8-12	92	115	122	228	306	97.7	0.4	1.9	80.6	72.0	11.0	...	...	1.0	7.4	17.0	26.5	48.1
16-8-3	96	116	121	222	274	96.7	0.7	2.6	83.5	72.8	10.4	4.7	...	0.6	6.3	15.7	22.2	54.2
16-27-11	104	137	145	217	288	98.8	0.4	0.8	83.5	68.9	4.5	5.8	...	0.2	3.5	10.6	26.3	59.6
16-7-11	106	126	130	202	282	98.5	0.3	1.2	94.0	74.3	8.9	5.7	...	...	2.0	24.8	44.3	28.9
16-7-1	108	127	130	206	296	98.8	0.1	1.1	92.0	73.9	8.5	5.1	...	...	2.4	23.8	41.2	32.6
16-7-4	104	126	131	250	370	97.6	0.7	1.7	82.5	71.8	7.7	4.7	...	...	2.1	20.5	35.6	41.8
16-14-1	120	151	160	295	378	98.0	0.6	1.4	46.6	62.9	5.4	4.3	...	...	2.5	8.6	18.0	70.9
16-14-4	122	156	166	306	394	98.1	1.0	0.9	42.0	61.9	4.5	3.8	...	...	1.7	7.1	9.0	82.2
16-14-7	114	138	144	227	318	98.3	0.5	1.2	83.0	68.4	6.7	5.2	5.5	...	...	...	...	...
16-20-2	93	116	123	232	295	97.1	0.4	2.5	79.0	71.8	9.9	5.0	...	0.6	6.1	12.6	21.8	58.9
16-20-5	89	115	122	217	300	97.4	0.4	2.2	87.5	72.4	10.1	4.7	...	0.7	6.5	17.2	25.3	50.3
16-5-6	122	151	160	278	364	98.4	0.9	0.7	55.0	63.5	4.5	4.4	4.3	...	1.3	8.3	16.9	73.5
16-22-6	102	126	131	222	290	97.0	0.6	2.4	84.0	71.7	9.3	4.5	4.3	...	3.3	17.5	27.5	51.7
17-30-11	98	127	140	269	344	98.5	0.5	1.0	55.5	66.9	8.8	6.0	...	0.5	4.6	12.3	20.9	58.5
17-3-1	98	126	134	261	328	98.1	0.2	1.7	60.0	68.2	8.9	6.3	...	0.3	5.2	13.4	19.0	62.1
17-3-3	100	126	134	261	326	98.0	0.6	1.4	61.3	68.2	7.4	6.5	...	...	4.7	13.4	16.9	65.0
17-3-7	104	131	140	256	325	98.0	0.5	1.5	62.0	68.4	8.2	6.8	6.6	...	...	...	...	...
17-24-6	102	131	139	347	390	96.1	0.9	3.0	32.0	58.0	8.2	5.9	5.0	...	4.4	9.8	9.9	75.9
18-11	102	128	136	260	352	98.8	0.6	0.6	71.5	67.9	7.8	6.2	...	...	3.6	15.8	29.7	50.9
18-4-12	118	141	149	288	388	98.2	0.9	0.9	62.0	66.9	5.4	3.5	...	...	1.2	13.3	23.0	62.5
18-18-2	102	126	134	272	360	98.2	0.5	1.3	70.5	68.3	6.9	3.5	...	...	2.3	14.4	14.9	68.4
18-2-3	94	117	124	250	334	97.9	0.6	1.5	75.7	71.7	8.6	4.6	...	...	4.9	15.3	24.7	55.1
18-18-5	94	118	125	233	312	97.8	0.6	1.6	81.6	72.2	9.5	4.9	...	...	5.8	18.8	28.1	47.3
18-21-6	124	154	160	270	336	97.1	0.9	2.0	56.0	63.4	4.9	9.3	8.7	...	1.7	7.5	16.2	74.6
19-6-11	117	139	142	221	308	98.5	0.5	1.0	86.0	70.7	6.1	5.2	...	...	1.5	15.7	59.4	23.4
19-6-12	116	139	145	223	310	98.5	0.4	1.1	85.3	70.3	6.2	4.6	...	...	1.7	15.3	35.0	48.0
19-6-2	118	140	144	222	303	98.1	0.4	1.5	85.0	70.1	5.6	5.0	...	...	1.1	13.6	21.8	63.5

TABLE 3 (*Continued*)

Identification[1]	A.S.T.M. Distillation Data[2] Temperatures, Deg. Fahr. 1st drop	5% cond.	10% evap.	90% evap.	End point	Percentages Rec.	Res.	Loss	Evap. at 212° F.	Gravity °A.P.I. at 60° F.	R.V.P.[3] at 100° F.	A.K. Rating[4] 212° F.	350° F.	Composition[5] Volume per cent C_3 and light	C_4	C_5	C	C_7 and heavy
19-6-4	116	136	142	221	300	98.4	0.5	1.1	86.0	70.6	6.4	4.3	...	...	1.1	16.6	28.8	53.5
19-6-6	116	142	150	276	340	98.0	1.0	1.0	61.0	66.7	5.8	3.8	3.5	...	1.5	11.5	25.2	61.8
20-20-11	112	137	143	257	344	98.0	0.8	1.2	73.0	65.6	6.4	7.1	...	...	2.3	13.9	30.4	53.4
20-20-1	92	111	116	245	336	97.8	0.1	2.1	78.3	71.6	10.9	6.1	...	0.9	6.7	19.6	31.1	41.7
20-20-4	114	137	143	251	341	98.7	0.5	0.8	76.0	65.9	6.1	7.0	...	...	2.3	12.5	25.6	59.6
20-20-7	115	141	147	252	338	98.1	0.5	1.4	75.0	65.9	6.5	7.5	6.7	...	...	...	...	...
20-14-12	93	122	130	264	342	97.9	0.4	1.7	65.3	68.8	8.8	5.5	...	0.4	5.5	13.9	20.3	59.9
20-14-3	100	126	133	277	362	97.9	0.6	1.5	69.3	67.7	7.7	6.9	...	1.3	3.7	14.3	24.4	57.3
20-14-6	110	130	134	252	342	97.4	0.8	1.8	75.0	68.5	7.8	7.8	6.5	...	3.0	17.0	29.0	51.0
20-10-2	100	123	128	252	349	98.0	0.6	1.4	75.0	69.3	8.3	7.2	...	...	3.1	15.4	32.1	49.4
20-10-6	102	138	152	377	424	96.8	1.1	2.1	76.1	55.4	7.3	5.5	5.5	...	4.7	6.9	9.0	79.4
20-5-2	100	122	128	253	340	97.9	0.6	1.5	76.0	69.4	8.8	7.2	...	...	4.6	17.5	18.5	59.4
20-5-4	99	121	129	246	341	98.5	0.5	1.0	77.0	69.6	8.4	6.8	...	...	4.4	18.5	28.7	48.4
20-5-7	107	130	137	267	345	98.0	0.8	1.2	70.0	67.2	7.3	7.2	6.2	...	...	...	...	...
20-2-3	100	123	129	252	349	98.1	0.5	1.4	75.0	69.5	8.0	7.3	...	...	4.3	17.7	27.4	50.6
21-14-11	107	134	145	292	360	99.0	0.5	0.5	51.0	66.8	6.5	4.0	...	...	3.6	11.8	21.4	63.2
21-14-2	103	134	142	273	356	97.7	0.7	1.6	61.0	67.6	6.5	3.6	...	...	3.7	12.1	21.6	62.6
21-14-5	106	136	145	265	358	98.0	0.7	1.3	65.0	67.9	6.1	4.5	...	...	2.6	13.6	19.9	64.0
22-5-6	107	132	139	236	357	97.8	0.6	1.6	78.0	66.8	6.8	5.5	5.4	...	3.4	11.3	24.4	60.9
23-20-6	108	132	141	258	332	97.6	0.9	1.5	60.5	67.3	7.7	10.2	9.3	...	4.7	12.0	17.1	66.4
24-23-6	104	132	139	315	368	96.8	0.8	2.4	49.0	55.3	8.2	9.2	7.7	...	4.8	9.4	14.8	71.0
25-18-6	124	144	148	227	342	97.2	0.8	2.0	84.6	49.7	5.5	9.9	8.0	...	...	12.9	16.9	32.0[7]
26-2-7	116	147	156	256	326	98.3	0.4	1.3	59.0	66.1	5.3	5.9	6.1	...	...	...	...	...
27-29-11	104	129	140	288	354	98.8	0.6	0.6	47.5	61.9	7.5	6.6	...	...	5.0	13.2	18.7	63.1

[1] First number refers to brand of gasoline (9, 11, and 15 are aviation naturals); second number refers to the city in which the sample was collected; third number refers to the month of the year in which the sample was collected.

[2] "Cond.," "Evap.," "Rec.," and "Res." are abbreviations for "condensed," "evaporated," "recovery" and "residue," respectively.

[3] "R.V.P." signifies Reid vapor pressure at 100 deg. fahr. in pounds per square inch absolute.

[4] "A.K.Rating" means anti-knock ratings which are expessed in terms of per cent of aniline in reference fuel described in connection with Fig. 2. The temperatures are those of the cooling medium employed.

[5] "C_3," "C_4," etc. signify the paraffin hydrocarbons having the number of carbon atoms designated, values given include hydrocarbons other than paraffin, which have boiling points close to those of the paraffin hydrocarbons.

[6] 3.4 per cent benzene.

[7] 38.2 per cent benzene.

to the designer of aviation motors, as it minimizes the problem of securing complete vaporization and equalized distribution of fuel to the various cylinders. In the experience of the company with which the author is associated, using a large variety of motors and of gasolines, it has been found that exhaust-heated manifolds or hot spots in carburetors are entirely unnecessary with these modern fuels, and, under some conditions, decidedly injurious as they reduce the volumetric efficiency of the engine, increase detonation possibilities, and result in a loss in r.p.m. of the motor. On the other hand, the low manifold temperatures resulting from the rapid vaporization of these gasolines increases the tendency of the carburetor to frost up, and become clogged with ice. This condition can be eliminated by a controllable device for moderately heating the air at the carburetor intake. In one instance these low manifold temperatures have been successfully utilized for cooling the hot oil from the motor.

4 Flight Tests to Determine Actual Temperature in Airplane Fuel Systems

Under the direction of Dr. G. G. Brown of the University of Michigan, the Natural Gasoline Association of America has sponsored a research program which has for its primary object the determination of temperatures in the fuel systems of airplanes in actual flight. This investigation is now under way. Thus far two ships, one a Hornet-engined Boeing biplane, the other a Wasp-engined Travelair monoplane, have been equipped with instruments to determine the temperatures in the fuel system.

The investigation is not far enough along to provide a basis for definite conclusions. However, in the work with the wasp-engined Travelair monoplane, several interesting observations have been made.

The maximum temperature in the fuel system, exclusive of the fuel tank, has never exceeded the outside air temperature by more than 20 deg. fahr. At the higher air temperatures (60 to 80 deg. fahr.) the difference has not exceeded 15 deg. fahr. Fig. 3 shows the relation between the temperature of the outside air and of the gasoline in the carburetor passage for this fuel system.

In order to furnish a basis of comparison for the significance of the generalization just stated, Fig. 4 is presented. The diagram shows the effect of altitude on the boiling points of gasolines of several vapor-pressure values. The temperature–vapor-pressure relationships on which the curves in Fig. 4 are based are those of the paraffin hydrocarbons. The straight line at the left represents the temperature of the standard atmosphere as defined in National Advisory Committee for Aeronautics, Report No. 218, "Standard Atmosphere—Tables and Data," by Walter S. Diehl. The dotted line represents the author's estimate of the air temperatures above territory at high elevation on an exceptionally warm day. The shaded section represents the 20-deg. difference observed above. It is to be noted that the shaded portion is closest to the gasoline curves at ground conditions and recedes from the curves with increased altitude.

Where the gasoline curves cross the shaded section represents the conditions where difficulty with vapor lock would occur in the particular fuel system studied. Under the extreme conditions outlined, namely, high altitude and excessively high ground temperatures, fuels having vapor pressures of 10.5 lb. per sq. in. are on the border line of satisfactory operation. Even under these extreme conditions, fuels with vapor pressures of 8.5 to 9.0 lb. per sq. in. would appear to provide a suitable factor of safety.

Fig. 4 indicates that any fuel which will operate an airplane engine for several minutes on the ground will be satisfactory in that fuel system from a vapor-lock viewpoint under all other conditions. It does not follow that a fuel which performed satisfactorily on the ground in a gravity-feed system would necessarily be satisfactory under all conditions of flight from a suction-feed system.

Another interesting observation in the flight tests with the Travelair plane was the fact that the gasoline temperatures in the fuel system were at no time in the course of the experiments as high as the temperature of the gasoline in the fuel tank on the ground before the take-off. With the rapid effect of air temperatures on fuel-system temperatures previously noted, this would indicate that a condition of vapor lock would be practically impossible in this particular installation, inasmuch as the gasoline, if susceptible of vapor locking, would be "weathered" in the fuel tank to such an extent that it could not again reach the "bubble point." The effect would be similar to that observed by Bauer where he noted that vapor lock could not occur in automobiles

equipped with vacuum tanks provided the vacuum tank was the hottest part of the fuel system. It is, of course, possible to conceive conditions of flight and of loading with gasoline which would invalidate this observation.

It has been suggested that the use of natural gasoline results in high cylinder-head temperatures. Some preliminary experiments have been conducted to determine this fact. The indications so far are that for fuels of corresponding anti-knock rating and with the proper adjustment of air-fuel ratios, there is no noticeable difference in cylinder-head temperatures.

The test results just submitted are not presented as conclusive. They are limited to the particular fuel system studied and the flight conditions of the tests so far conducted. The data have been presented as illustrative of the type of work being conducted under the present research program of the Natural Gasoline Association of America.

5 Other Investigations

Prominent in this field has been the U. S. Bureau of Standards, which is doing excellent work in providing a better understanding of the fundamental factors involved in vapor pressure, vapor lock, A.S.T.M. distillation data, and engine performance by means of laboratory investigations. With the continually augmented supply of volatile gasolines, it is absolutely necessary for all interests concerned to consider gasolines from an entirely different viewpoint than formerly. The Bureau of Standards has made some valuable contributions to this end.

Although not directly on the subject of aviation gasoline, the investigations conducted during the past three years at the University of Michigan under the direction of Dr. G. G. Brown and sponsored by the Natural Gasoline Association of America have a very close relation to the subject. The research work referred to is a very comprehensive study of the effect of volatility on motor-fuel performance.

Many industrial laboratories and organizations have contributed to the general knowledge concerning fuel qualities. For the benefit of those interested in the subject, an incomplete bibliography is appended, with special emphasis on those articles which have a bearing on the use of aviation natural gasoline.

Although there are many advantages in the use of more volatile fuels, such as ease of starting, improved warming-up characteristics, superior accelerating ability, and higher anti-knock ratings, the introduction of these fuels is not without its entries on the other side of the ledger. The problem of vapor locking is a very definite liability, and a factor to be seriously considered. Due no doubt to its pronounced commercial success, there has been a marked tendency to attribute this relatively new factor in gasoline performance to the activity of the natural-gasoline industry. But it cannot be too strongly emphasized that aviation natural gasoline is but one of the several manifestations of an economic condition which clearly indicate the gasolines of the future will be decidedly more volatile than those of the past. The surveys of motor and aviation gasolines cited in this paper clearly demonstrate this trend. And as certain as is this trend, it is just as certain that the designers of gasoline-consuming units have not kept abreast of the times. Automobiles continue to appear on the market with fuel systems so located that it appears a deliberate attempt had been made to heat the gasoline to the utmost on its path to the carburetor needle. In the airplane industry, safety is an added inducement to keep the gasoline removed from zones of high temperature. Moreover, the airplane industry is young and without the burden of tradition, so that the application of sound engineering principles is not so difficult. With the trend toward more volatile fuels receiving so much emphasis at this time, the airplane industry can well afford to keep a watchful eye on the design of its fuel systems.

Bibliography of Papers Related to Use of Aviation Natural Gasoline as Aviation Fuel

1. G. G. Oberfell, T. W. Legatski, and Billy Parker, "Aviation Natural and Its Relation to Other Aviation Gasolines," U. S. Air Services, February, 1930.

2. R. C. Alden and Billy Parker, "Natural Gasoline as Aviation Fuel," *Oil and Gas Journal*, May 29, 1930; also Proc. N.G.A.A., May, 1930.

3. G. G. Brown, "The Volatility of Motor Fuels," Engineering Research Bulletin No. 14 of the University of Michigan, May, 1930.

4. G. G. Oberfell, "Volatility of Natural Gasoline," Proc. N.G.A.A., May, 1928.

5. R. E. Wilson, "The Significance of Tests for Motor Fuels," Detroit Regional Meeting, A.S.T.M., March, 1930; also *Oil and Gas Journal*, July 17 and 24, 1930.

6. G. G. Brown, "Motor Performance as Determined by Fuel Volatility," Atlantic City Meeting, A.S.T.M., June, 1930.

7. O. C. Bridgeman and Elizabeth Aldrich, "The Properties of Gasolines With Reference to Vapor Lock," Detroit Meeting of S.A.E., January, 1930.

8. O. C. Bridgeman, Elizabeth Aldrich, and H. S. White, "Vapor-Pressure Data on Motor Gasolines," Jl. S.A.E., vol. 24 (1929), p. 488.

9. W. C. Bauer, "Vapor Locking Tendencies of Automotive Fuel Systems," Jl. S.A.E., vol. 26 (1930), p. 736.

10. O. C. Bridgeman and E. W. Aldrich, "The Effect of Weathering on the Vapor-Locking Tendencies of Gasolines," Jl. S.A.E., vol. 27 (1930), p. 344.

11. C. S. Bruce, "Engine Acceleration," Jl. S.A.E., vol. 27 (1930), p. 275.

12. O. C. Bridgeman and C. S. Cragoe, "An Interpretation of the A.S.T.M. Distillation Curves of Motor Fuels," Proc. Am. Petroleum Inst., vol. 9(1927), no. 7, p. 54.

13. U. S. Bureau of Mines Semiannual Surveys of Motor Gasolines.

14. G. G. Oberfell, R. C. Alden, and H. Hepp, "Comparison of Vapor-Pressure Testing Methods," Proc. Am. Petroleum Inst., vol. 9 (1927), no. 7, p. 106.

15. G. G. Oberfell and R. C. Alden, "A Discussion of the Front-End Volatility of Commercial Motor Fuels," *National Petroleum News*, August 27, 1930.

Discussion

H. K. Cummings.[2] The Phillips Petroleum Company has made an important contribution to information regarding the commercial aviation gasolines available today by permitting Mr. Alden to give in Table 3 complete laboratory data on 126 samples representing 27 different brands of gasoline as sold at various airports between November, 1929, and July, 1930. It is hoped that arrangements can soon be made for a semiannual aviation gasoline survey to be undertaken by the Department of Commerce.

The anti-knock ratings reported in Table 3 and those shown graphically in Fig. 2, like all ratings expressed in terms of a particular reference gasoline, are difficult to compare with results obtained in other laboratories, even though the author furnishes the additional information that his reference fuel required 7.3 per cent aniline to equal the 1929 "Ethyl Standard" for motor gasoline. An approximate evaluation of this gasoline-aniline scale in terms of octane numbers, as recently recommended by the Detonation Sub-committee of the Cooperative Fuel Research Steering Committee, would be very helpful. The octane number of a fuel is defined as the percentage by volume of iso-octane (2-2-4 trimethyl pentane) which must be mixed with normal heptane to give a blend having the same tendency to knock as the fuel in question. The octane number of his 7.3 per cent aniline blend would be about 72, while the octane numbers of most straight-run Pennsylvania gasolines are between 40 and 50.

In connection with the author's conclusion 3 (*f*), Dr. Bridgeman's work shows that the Reid vapor pressure can be determined from the A.S.T.M. 10 per cent evaporated temperature, if allowance is made for the slope of the distillation curve at

[2] Bureau of Standards, Washington, D. C.

that point, provided the gasoline is propane-free or contains a known percentage of propane. As a corollary, the propane content can be calculated from the Reid vapor pressure and the distillation data.

It should be noted that the temperature data presented in Fig. 3 apply only to a single airplane. For a given initial gasoline temperature, the spread between air temperature and gasoline temperature at altitude varies widely in different types of airplane. From data already obtained on fuel-line temperatures in seven types of airplane during flight, it appears that the temperature of the gasoline in the main tank when the airplane leaves the ground is the most important single factor as regards vapor lock. The volatility of the gasoline and the design of the fuel system are of course the other factors. In some installations it may be desirable to provide means for cooling the gasoline as it leaves the fuel tank.

As pointed out by the author, weathering of the gasoline in the fuel tank affords some factor of safety. The amount of weathering depends on the rate of climb and the size of vent provided. Tests at the Bureau of Standards indicate that weathering is ordinarily negligible up to the vapor-locking temperature. If vapor lock does not occur when the gasoline first starts to boil, weathering will probably take care of it.

J. H. Doolittle.[3] This is an extremely well-written paper, and the author handles the subject in a masterly fashion. On a few points the writer either does not quite understand Mr. Alden or is not in complete agreement with him. He stated that the Federal specifications took cognizance of vapor pressure. This has been true in the past, but the new Army specifications will, it is believed, limit the Reed vapor pressure to 6½ lb. The military specifications are of course somewhat more strict than those required for commercial use, as military airplanes usually have the gasoline tanks located in the wings and employ a pump to deliver it to the carburetor. This pump sometimes imposes a negative pressure head on the gasoline in the line, especially when the fuel tanks are almost empty. In commercial work the gasoline tanks are generally located in the wings and gravity feed is utilized. Furthermore, in military work, especially with supercharged engines, the airplanes are sometimes obliged to operate at extremely high altitudes, so high that the temperature becomes constant while the pressure continues to decrease. This has the effect of steepening the dotted curve and will tend to bend it over toward the vapor pressure curves shown on the right-hand side of the sheet.

Although a maximum temperature difference between the gas-line temperature and the outside-air temperature was found to be 15 deg. in the Travelair tested, in our Travelair "mystery" plane, with its poor ventilation and N.A.C.A. cowling, this temperature difference was of the order of 30 deg. In this airplane in hot weather we experienced gas locks several times when testing out gasoline of moderately high vapor pressure. Frank Hawks in one of his first test flights in a smaller airplane had the motor stop, and the writer is quite sure that this was due to vapor lock. In both his plane and ours, ventilation systems were arranged to take in cold air at the leading edge of the wings and pass it around the gasoline system. In this way the temperature difference between fuel system and outside air was substantially reduced and no further difficulty with gas lock was experienced. Mr. Alden states that fuels with a vapor pressure of from 8½ to 9 lb. per sq. in. would appear to provide a suitable factor of safety. Considering Fig. 4, suppose it is assumed that Yuma, Ariz., is located approximately at sea level (200 ft. above). The temperature there this summer was 112 deg. Adding to this the 15-deg. difference found in their Travelair, we found the maximum vapor pressure to be 8.8 without any factor of safety, and if the vapor pressure in our own Travelair is considered to be 8.8 with a 30-deg. temperature difference, the maximum vapor pressure is 6.8. Even this does not take into consideration variations in barometric pressure at any given point. Added to this there is sometimes, up to altitudes of 4000 ft., a temperature inversion where the temperature at altitude is actually higher than the temperature at the ground. This phenomena is most common in the winter time, but has been observed even on very hot summer days.

[3] Aviation Department, Shell Petroleum Corporation, St. Louis, Mo.

Robert E. Wilson[4] and Daniel P. Barnard, 4th.[5] The author is to be congratulated for his most thorough survey of the composition-vapor pressure characteristics of aviation gasolines. To the best of information it is the most complete yet published, and should be of considerable assistance to the further development of these fuels.

It has been the writers' experience that fuels of rather high vapor pressures may be used under many conditions with no signs of vapor lock, and to this extent there is complete accord with the author. Unfortunately, however, the fact that such fuels may be used in the majority of instances does not give them a clean bill of health, as there always exists the possibility that some combination of circumstances may cause a fuel system to vapor-lock under the most embarrassing conditions. Most of the commercial transport planes now in use will handle a fuel of, say, 8 or 8.5 lb. Reid vapor pressure with entire success under normal conditions. Let the air temperature rise to 100 or 110 deg. fahr., however, as frequently occurs in summer in the southern half of the country, and vapor lock with such a fuel not only becomes possible, but in many planes it is probable. Furthermore, it appears to have been quite definitely established that current military equipment cannot safely utilize a fuel having more than 6.5 to 7 lb. Reid vapor pressure.

It is unfortunate that the author does not express his knock ratings in terms of iso-octane and normal heptane. Experience indicates, however, that natural gasolines of the highest practicable volatility do not possess knock ratings even near the 74 per cent iso-octane–26 per cent *n*-heptane which is evidently required for the engines built concurrently for transport and military use. It is worthy of note that a fuel such as natural gasoline having 90 per cent evaporated at, say, 230 deg. fahr. responds quite readily to the addition of tetra-ethyl lead much more readily than motor gasolines or even gasolines corresponding to D.A.G. specifications. Unfortunately, this method of improving the anti-knock value of natural gasoline cannot be generally used today because certain metals used in aircraft-engine construction are attacked by the products of combustion of ethylized gasoline. This difficulty has been entirely overcome in automobile engines and is being overcome in aviation engines, but for the present it does not seem safe to market it for general use with the wide variety of equipment at present employed in military, transport, and private flying.

C. M. Larson.[6] The facts that the author has presented show clearly how progress and engineering data make it possible to overcome objectionable stipulations. By this is meant that during the World War the Navy Department specification 8-G-1-B, Sept. 1, 1917, used by both the Army and Navy Air Services, prohibited the use of natural gasolines, as shown by the following extract:

"*Aeroplane Gasoline.* Quality: To be high-grade, refined,

[4,5] Standard Oil Company (Indiana), Chicago, Ill.

[6] Supervising Engineer, Sinclair Refining Company, New York, N. Y. Mem. A.S.M.E.

and free from water. . . . No natural-gas gasolines shall be accepted nor shall they be mixed with any gasoline submitted for acceptance."

However, the objection to the use of natural gasolines in part or in whole has been dropped from all aviation gasoline specifications in effect today.

The main trouble experienced from natural gasolines when used in aviation engines is that of "gas locks." The chemical constituents of natural gasolines have been isolated and their respective vapor pressures studied, as shown in Fig. 5. By

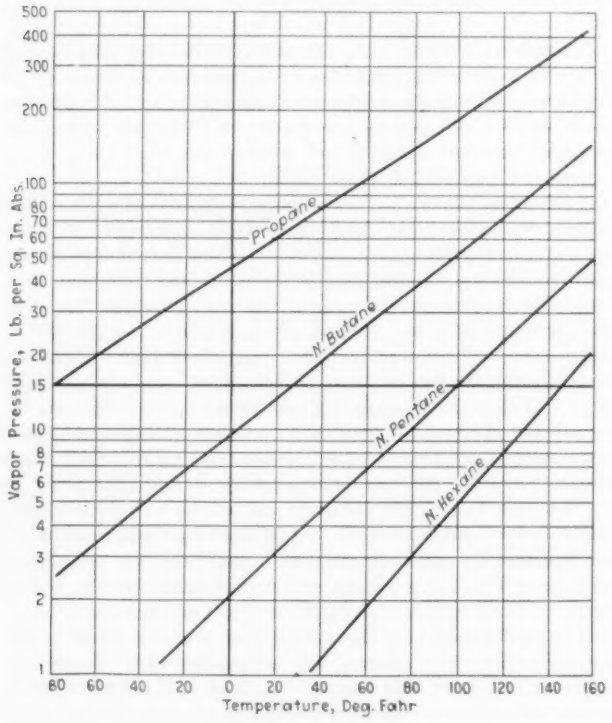


FIG. 5

knowing the constituents to remove to overcome vapor locks, and by their extraction, it is possible to control the vapor pressure of any gasoline at the desired 10 per cent over point, as plotted in Fig. 6.

The question of suitable fuel for private and commercial aircraft purposes has been a subject of wide discussion during recent months throughout the United States. From the fuel producer's standpoint, development of an improved normal fuel with a proportionate and yet reasonable increase in cost is necessary so that the engine manufacturers will be in a position to offer an improved product to the operator. Just how far and how fast this improvement in the Federal Specification Board's Domestic Aviation Gasoline Grade B (Department of Commerce Technical Paper 323B) can be made, depends on the economic distribution and availability of the product as well as the standardization of the aircraft-engine manufacturer's requirements.

The petroleum and aircraft engine industries will welcome the day when the working out through proper channels of

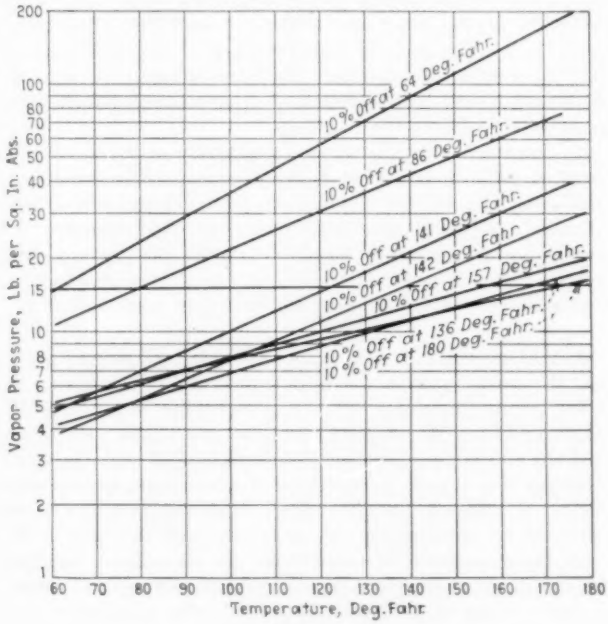


FIG. 6

specifications of two grades of aviation gasoline are arrived at—one aviation gasoline with a reasonably high anti-knock value, to be used for private flying and aerial service as well as by scheduled transport consumers, and perhaps one other, but not more than one, additional specification to cover aviation fuel for the Army and Navy and such scheduled transport lines as well as those consumers that may desire to use supercharged engines, all of whom find it necessary to go to compression ratios higher than 5.1 to 1 for air-cooled engines.

Until this is done, the oil companies c nnot afford to plan national distribution for a demand equivalent to only that used by 50,000 automobiles, as compared with the 26,600,000 automobiles operating over the same territory.

AER-53-7

Commercial Aircraft Radiophone Communication

By ROBERT H. FREEMAN,[1] OAKLAND, CALIF.

The need of dependable two-way aircraft communication has emphasized the service requirements as they are presented to the pilot. The equipment must be light and yet rugged, and as the essential parts are placed at a distance from the pilot, they must be simple of operation. The paper, without stressing the electrical-engineering side of the air-communication problem, covers the actual mechanical set-up. The mechanical and electrical difficulties encountered in shielding the ignition system and the complete bonding of the ship are discussed in detail. The placement of the component parts for ease of operation is taken up and the subsequent maintenance of the equipment is briefly outlined.



IN the last year the need of reliable two-way aircraft communication has been brought before the engineering world in a manner only too realistic and vivid. Every week situations arise which could have been foreseen and most probably avoided if there had been two-way radio-telephone communication between the aircraft and its home station or field. There have been presented many papers before different bodies of engineers detailing the great need for this service, and some of the papers have dealt with the theoretical and practical electrical-engineering side of the question, but so far little attention has been paid to the actual service problem as presented to the operator of the aircraft.

In the past engineers have sat at their desks and designed equipment to fulfil certain needs without first ascertaining the conditions under which this equipment would be used. Later nearly all this equipment has been changed to suit the operating conditions. This is even more true in the air-transportation game. As will be pointed out later, there is equipment now manufactured to accomplish certain tasks, which it will do in the laboratory, but which fails under the tests given it by the air-transport operator. He has schedules to maintain and life to protect, and cannot afford to take any undue risks in the way of poor equipment.

The requirements for apparatus to fill this general need of communication are many and varied. The equipment must be light in weight, small, compact, and rugged enough to withstand rough handling and continuous vibration. In the mail ships the essential parts of the equipment must be located at a distance from the pilot, and must be simple in operation as he is unskilled in the manipulation of radio equipment. The apparatus should be designed so as to permit ready removal, and substitution of duplicate equipment in the short space of time it takes to oil and gas a ship.

[1] Communications Engineer, Boeing Air Transport, Inc., Oakland Municipal Airport. Mr. Freeman was born May 22, 1903, in Chicago, Ill. He received the B.S. degree from the University of Colorado in 1925. He was with the Bell Telephone Laboratories in the Radio Transmitter Development Department from 1925 to 1929.

Presented at the Aeronautic Meeting sponsored by the Aeronautic Division and Pittsburgh Local Section of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS and the Engineering Society of Western Pennsylvania, Pittsburgh, Pa., March 12, 1930.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

PREPARATION OF THE SHIP

Shielding. Because of the low powers of the transmitters on the ground the receivers on the ship must be unusually sensitive to obtain reliable communication. The gain of the receivers must be of the order of 120 decibels. With this enormous gain the interference problem is a very serious one, and before reception may be had at a distance of greater than 10 to 15 miles using 400 watts of power on the ground, all sources of electrical interference on the ship must be eliminated.

The greatest offender is the ignition system. This must be completely covered with metal in order to cut the disturbance to a point where it no longer hinders the reception of the voice signals. Shielding equipment has been turned out by various manufacturers and most of this has been tried. In all cases the apparatus failed in one of the two following manners. Either it did not shield at the frequencies employed, which were 5660 and 3172 kilocycles, or if it did shield, it did not last more than a few hours when flown over the regular mail route from Oakland to Reno. The whole situation simmers down to the fact that the designers of the equipment did not have a first-hand knowledge of the service conditions encountered. From the theoretical viewpoint the equipment was fine, but failed miserably when applied to practice.

The air-transport operator was not in the game of designing and building shielding equipment, but was forced to it by the lack of cooperation received from the manufacturers of aircraft engines and accessories. Not being versed in the art, it was a long and tedious process, but there was evolved a harness which completely eliminates the interference from the ignition system when using a receiver having a gain of 120 decibels.

The problem divides itself into three heads: Shielding the spark plugs, shielding the leads from the plugs to the magnetos, and shielding the magnetos. Although these three were worked out simultaneously, they will be taken up in the order named.

The problem of shielding the spark plug was a hard one to attack, as it must be kept in mind that one is dealing with high voltages of the order of 15 to 18 kilovolts. Space is at a minimum where the plugs fit on the back side of the engine and the plugs must be kept fairly cool. A good many types of cans were tried in which the standard plug was placed, but these proved to be a failure, either being unserviceable or impairing the efficiency of the motor. At this time a manufacturer of aviation plugs put out a shielded plug that seemed to be the answer to the situation. It was very effective in shielding and seemed to be readily serviced. They were used on the test ship for many flights while the other parts of the system were being worked out. Then they were placed on the regular run, and the plugs failed regularly after about 40 to 50 hours of service. The idea of a can was revived, and many types were tried to obtain a device which would shield the plug, stand up mechanically under the abuse of the service crew, be light in weight, low in cost, and in no way affect the operation of the

engine. Many weeks of experimenting passed before the final model was obtained. The plug itself is only a slight change from the plug used by many air-transport operators. This change is in the design of the jamb nut which holds the plug together. The nut is run up a little over $^1/_2$ in., forming a tubular sleeve with a groove at its base. Of course this nut has to be made first and the plug built up on the nut. However, this is no drawback as the manufacturer can supply them in a short time. The cap which fits over the plug is made of nickel steel, and the prongs, which fit down over the plug sleeve, and clamp into the groove at the base of the nut, are tempered so that they hold the spring action which is imparted to them during manufacture. At the top of the cap there is a gooseneck which reverses the direction of the wire in a rather short arc and heads it back toward the direction of the harness. This gooseneck also forms a convenient handle for removing the cap from the plug. At the end of the gooseneck there is an enlargement which is split, and allows the harness to enter and then be clamped up, thus forming a good electrical shield and an oil-tight joint. Inside the cap there is placed an insulator which fits down over the center electrode of the plug inside the nut sleeving. This forms a long path for the electrical energy to jump across or leak through. In the top of the center of the insulator there is placed a small spring and screw which connect the top of the plug to the wire entering the cap. All this can be seen readily by referring to the illustration of the shielded spark plug. This plug

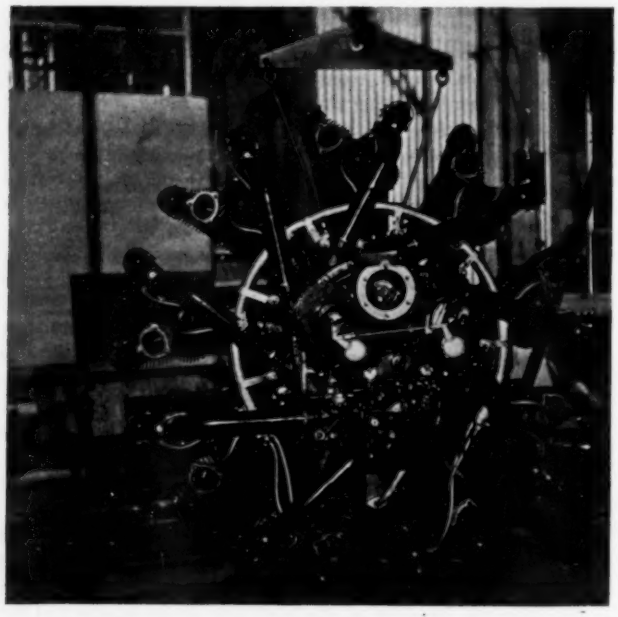


FIG. 1 OLD-TYPE SHIELDED HARNESS

is now in daily use on the lines of the Boeing System and other air-mail lines. The cost is nominal and the replacement of worn-out parts is practically the same as with the unshielded plug. The serviceability is even less complicated than with the open type of plug, for there is no need of tying the parts for safety, and upon the removal of the cap the plug can be removed from the cylinder as readily as with the regular type. As for the protection of the plug in stormy weather, there is no question as to its effectiveness. This plug is the only one to date being manufactured in quantity which is inexpensive, effectively shields the radiation of electrical energy, and will stand the "gaff" of repeated servicing by mechanics.

The shielding of the leads from the magnetos to the plugs while not so difficult was just as important. Belden braid or woven copper braid was first used to cover the wires and proved to be a very good electrical shield. However, after about 4 to 5 hr. in the air the ignition noise again became bothersome and at 10 hr. was sufficient to preclude any reception beyond about 15 to 20 miles. It developed that the cause was the oil soaking of the braid which insulated each strand from its neighbor enough to impair the effectiveness of the shielding. Many types of this woven braid were tried, but all failed after a few

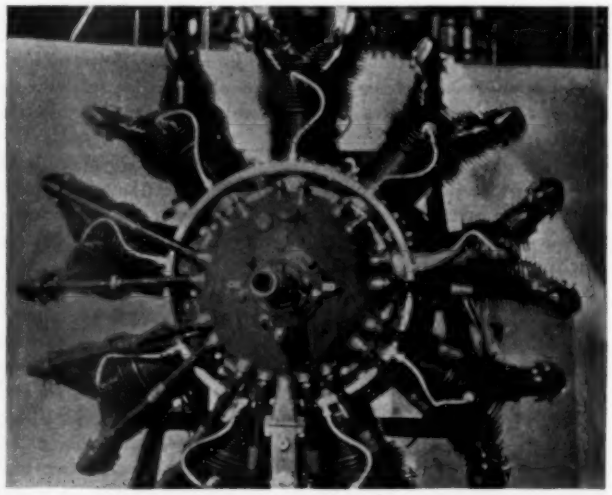


FIG. 2 NEW-TYPE HARNESS, FRONT VIEW

hours in the air or had received a few servicings by the ground crew.

The next step was the use of solid aluminum tubing which would stand a great deal of use as well as abuse. It consists of a main aluminum ring placed behind the cylinders. From this ring a small tube leads to each individual spark plug and a large "horn" carries the wires from the ring to the magnetos. To allow the servicing of the plugs and the magnetos it was necessary to use double braid from the ends of the tubes to the plugs and magnetos in order to get the desired flexibility. The braid was covered with tape and shellacked to prevent oil saturation. While this effectively shielded the disturbance it did not allow the desired ease of application of the harness or repeated servicings which are necessary in the air-transport business. The solid tubing did not stand up under the continual vibration and had to be replaced fairly often.

All sorts of flexible hose, tubing, and the like were tried, but none of them gave all the qualities desired, which were flexibility, shielding, and oil and mechanical protection to the ignition system. As the braid was a good shield and flexible carburetor hose was oil and mechanical protection, the two were combined, placing the braid inside the hose, and the long-looked-for result was achieved. The new harness still has the main aluminum ring, but with the small flexible leads running to the spark plugs, and also a large flexible tubing to the magnetos. The braid is used only where the tubing is flexible, thus saving weight and cost of shielded ignition wire with its accompanying corona losses and deterioration. This harness gave the necessary shielding, ease of application, would withstand innumerable servicings, and was inexpensive. The size of the tubing is such that if one wire becomes unusable it may be withdrawn and a new one pulled in. This is a big factor in the maintenance cost which was overlooked by several manufacturers, in that part of

their harness must be thrown away if one wire becomes damaged.

The shielding of the magnetos was a relatively simple matter. The one thing, however, that had to be kept in mind was the necessity of being able to remove all the magneto-distributor blocks without loosening or removing the magneto from the motor. This is easy to do if the motor is not equipped with an inertia starter, but as most ships now have starters of this type

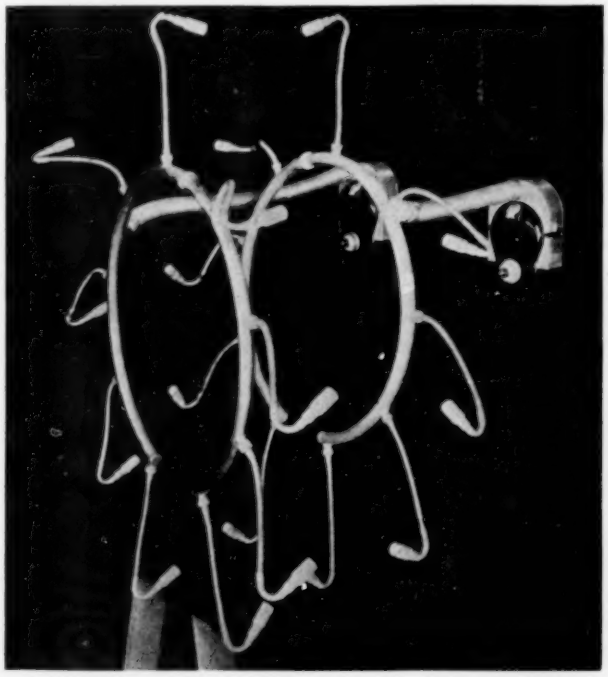


FIG. 3 SKELETON SET-UP SHOWING RELATIVE POSITION OF HARNESS AND MAGNETO AS MOUNTED ON A PRATT & WHITNEY MOTOR

some method had to be used to overcome this difficulty. The illustrations of the magneto shields are self-explanatory. The shield consists of two aluminum sheets bolted on to the magneto, and a band of spring bronze which covers the gap between the plates. In the front of the shield is a removable block with the outlet tube. This comes out when the blocks are removed to facilitate the operation and makes for a stronger shield. The booster and ground wires which have to be shielded make their exit at the rear of the shield through small tubes fastened in the rear plate. This allows the shielding on the wires to be readily attached to the magnetos. The spring band is held together with a toggle catch, making it very easy to open the shield for inspection and servicing. The cost of this shield is less than half that of any on the market and the weight is considerably reduced over the cast-aluminum type.

Thus a complete covering of metal for the ignition system is provided which suppresses the interference to a point where it is not audible in the receivers with their enormous gain. This covering does not interfere with the operation or maintenance of the ship and stands up under all the conditions encountered on the transcontinental and coastal lines of the company. It must be remembered that all this work was done using high frequencies, and does not necessarily apply to the intermediate frequencies used by the Department of Commerce for Radio-Range and Weather Services. However, a ship equipped for the high frequencies is better able to make use of the intermediate frequencies.

BONDING

Another source of great interference in the reception of the signals was the discharge of the static electricity in the different parts of the ship. It was not enough that the ship was bonded according to the Department of Commerce regulations. Every place that one piece of metal could rub on another was a source of interference. All joints had to be bypassed with a pigtail of copper braid soldered to both of the moving parts. The standard turnbuckle was replaced with a new type. This is essentially the standard type, but instead of using the brass safety wire a strip of bronze is wired and soldered on to the eye of the turnbuckle and run down past the hole in the barrel. The hole is slightly enlarged, and a specially made bolt provided with a half-moon-shaped head which holds the bronze strips from turning is inserted and fastened with a nut and lock washer. A piece of pigtail braid is soldered on to the eye, and passed across the joint to the wire or rod and is soldered there, thus making a continuous low-resistance circuit for all static charges.

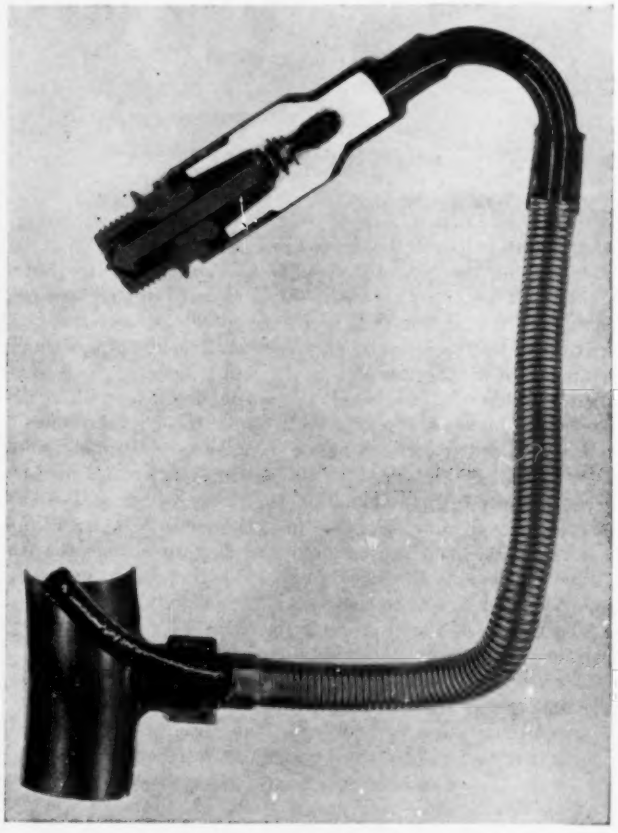


FIG. 4 NEW-TYPE HARNESS AND SHIELDED PLUG, WITH WHITE INSULATOR

All the wires in the ship are either shielded with the copper braid or else run in conduit. Wherever possible the conduit method is used as it permits the replacing of wires with the minimum of labor and expense. When the ship is equipped with a high-frequency transmitter, it is imperative that all the wires are completely covered with a grounded shield as otherwise they will pick up the energy and either burn out or at least absorb enough of the already too small energy being radiated from the antenna system.

Ship's Radio Equipment

The electrical qualifications for the radiophone equipment on the ship are easily enumerated, but not quite so readily arrived at. First, the transmitter must have as great an output, with the power input on the high-voltage side not exceeding 400 watts. Second, it must have an extremely stable frequency. Third, the percentage of modulation must be above 80. When these are combined with light weight, ruggedness, ease of operation, and compactness, one gets an idea as to what the problem is.

For phone communication the best antenna power available from the limited input is in the neighborhood of 50 watts. This

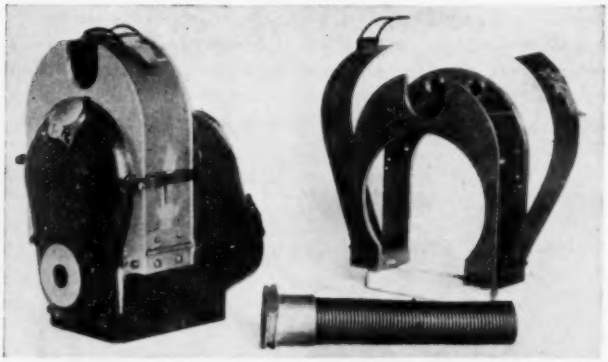


Fig. 5 Detail View of Magneto Shield

is sufficient to cover the same distance from the air that the 400 watts on the ground is capable of. To hold the frequency stable with the vibration of the ship and under such a high percentage of modulation it is necessary to employ quartz-crystal control of the oscillator. This crystal is held at a constant temperature by the use of a heater and a mercury thermostat. The crystal works at one-half the output frequency and a frequency-doubler tube is employed to obtain the desired frequency and also to act as a sort of buffer stage between the modulating tube and the oscillator. This is necessary since to obtain such a high percentage of modulation the plate voltage of the output tube has to vary from zero to twice normal value, and this will reflect back on the oscillator, tending to change the frequency. The use of this system holds the frequency as constant as is necessary to overcome distortion from shifting carrier.

The weight of the transmitter is about 32 lb., and the apparatus occupies a relatively small amount of space.

The receivers are of two types: The high-frequency one for the reception of the two-way ground transmitters, and for two-way communication with other ships; and the low-frequency one for the reception of the Department of Commerce weather stations and radio-range equipment. These receivers are extremely sensitive, as has been pointed out, having an overall gain of better than a million. They weigh 13 lb. apiece, and are remotely controlled by the use of a high-speed tachometer shaft so designed that there is very little or no backlash in the tuning of the receivers. The volume is controlled by the pilot, the units being situated just above the throttle, as can be seen by reference to the illustration. The two volume controls are at the right side and the two tuning controls at the left side. In the lower right-hand corner, the main off-and-on switch may be seen. After the equipment has been placed in the standby position the pilot merely presses a button on his stick and talks into the microphone, and to listen he removes his finger from the button. Thus the equipment is made as simple of operation as is possible so that the pilots will make ready use of the equipment and not be overtaxed trying to remember "how this new-fangled thing works."

The extent to which this simplicity of operation has been pushed can be seen from the final models of the radio helmet. The microphone is held in front of the pilot so that he has his hands free to manipulate the controls of the ship but arranged so that it can be pushed up or down out of the way. The receivers are of a special light-weight construction, weighing less than an ounce. Attached to the receivers is a molded ear-piece shutting out external noises, and putting the signal into the inner ear where it does the most good. Thus the acoustical problem has been somewhat solved. The power supply for the transmitter and receivers was given a great deal of thought, and only after trying out all the available types was the present one chosen. It consists of two dynamotors, one for the transmitter, and the other for the receivers. The transmitting dynamotor derives its power from the ship's storage battery and delivers 1050 volts at a current drain of 400 milliamperes. The smaller machine also derives its power from the storage battery, but delivers only about 200 volts. This supplies the "B" and "C" power for the two receivers.

The reason for choosing this equipment is that it is always available for use whether the ship is in the air on the ground with a dead motor. The engine-driven generator cannot be used when motor failure occurs, and the wind-driven type cannot be used to any advantage when the ship is on the ground, especially if the motor is out of commission.

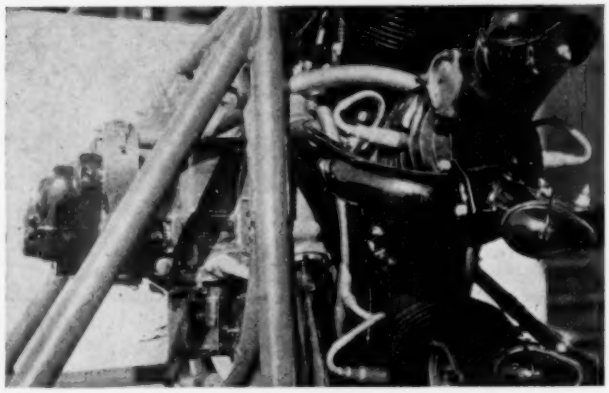


Fig. 6 Rear Ring Magneto and Shields

The two dynamotors with their control equipment and filter systems are mounted on a frame which can be removed from the ship and another inserted in about two minutes' time. The connections are made with a multi-terminal plug and jack box. All of the equipment which has a possibility of failing in service is arranged for ready removal and replacement by tested units.

Antenna System

In the past practically all radio work done on ships used the trailing-wire type of antenna. This is not satisfactory from a good many angles, as there is considerable head resistance, the lead "fish" dropping off every once in a while, and having been known to cause loss of life, and in flying the air mail from Chicago to San Francisco, many times it is necessary to fly at an altitude which precludes all possibility of any hanging wires.

The mast type of antenna is used for the reception of signals, and also for transmission on the daytime frequency which is 5660 kilocycles. When using the 6-ft. dural mast for the night frequency of 3172 kilocycles, the effective radiation is so reduced that the proper coverage is not obtained. To overcome this

difficulty two wires are run from the front-wing spar on either side of the gas tank to the top of the mast and then back to the vertical fin. With this antenna, transmission can be effected when the ship is on the ground without the pilot's running out a wire to the nearest fence post and retuning his transmitter. The wires also tend to steady the mast.

The transmitter on the ground end of the circuit is electrically the same as the aircraft transmitter with the addition of a power amplifier which boosts the power from 50 to 400 watts. It has the same crystal control and the units are interchangeable, so a smaller number of spares are needed. The power is derived from a double-voltage three-phase 220-volt rectifier. This supplies the power stage with 2500 volts, and the rest of the transmitter with 1000 volts. It makes use of the newly developed hot-cathode mercury-vapor tube for rectifying.

The transmitter and rectifier are housed in separate frames, but are placed on the operating table alongside each other. This makes a very compact set-up and has proved entirely satisfactory. The receiver is of the same type as the high-frequency receiver used in the ship. The remote tuning mechanism is removed and a knob is substituted for the ground work. Thus a smaller supply of spares is needed to take care of emergencies.

All of the stations are equipped with the teletype or telephone printer, which is a source of never-ending radio interference. After working with the installing companies for many months trying to get them to clear up the trouble it was found necessary to tackle this problem ourselves. As a result all of the machines in use at our stations are now, or soon will be, equipped

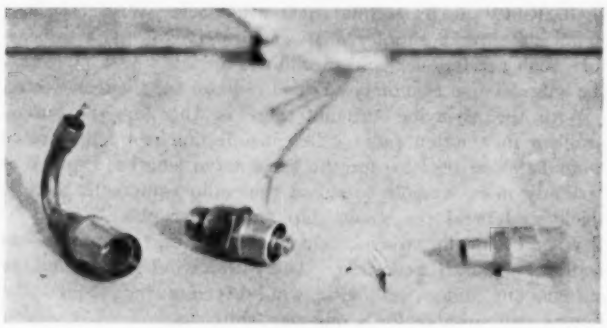


FIG. 7 SHIELDED SPARK PLUG, GOOSENECK SPRING, AND INSULATOR

with a rotating squirrel-cage type of induction motor. This removes the cause of the radio interference as the usual-type motor has five sparking contacts. The maintenance of the printer has been reduced and the reliability has not been impaired; in some cases it has been reported that the speed is more constant than with the governor type of motor.

SERVICE PROBLEMS

Each pilot fills out a card at the end of his run, giving the record of the performance of the ship and the radio equipment. If there is anything wrong with the radio equipment it is removed and thoroughly tested until the cause of the failure is found. Before a ship is allowed to be taken out on the line the radio mechanic gives the entire ship's radio equipment a test. Besides that he tests the condition of the battery and the generator. All of the readings are marked down on a form provided for that purpose, and this record is submitted to the radio man in charge. If the readings vary from the allowable limits it is the duty of the radio mechanic to find what the trouble is and fix it. No ship can go out until it has his check of approval.

If a ship comes in to one of the intermediate fields with something wrong in the radio equipment it is the duty of the operator in charge there to replace the defective apparatus with the spare and send the ship on its way. He then sends the apparatus to the nearest repair depot which is equipped to handle the repairing and testing of the apparatus. In this way there are no makeshift repairs which will lead to further grief, but a standard job is done which will stand up.

CONCLUSION

The results obtained from this system of two-way radiophone communication have more than justified the time and expense. It is a means whereby the lives of the pilots and passen-

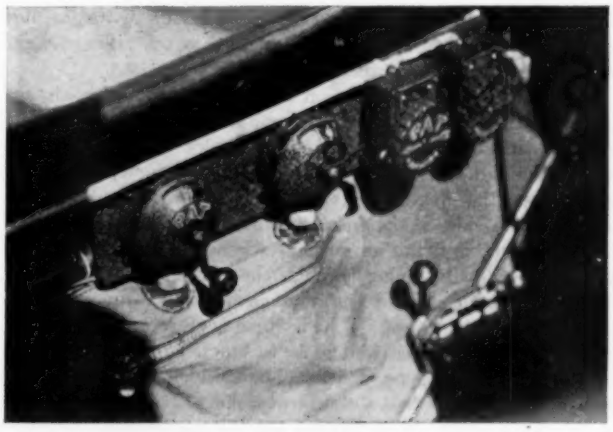


FIG. 8 REMOTE TUNING AND VOLUME CONTROLS AT LEFT SIDE OF COCKPIT ON MAIL PLANE

gers may be safeguarded and the loss of expensive equipment kept at a minimum. It is necessary to shield the ignition completely, including the spark plugs and all wires in the ship, and very thoroughly bond the ship if any reliable communication is to be hoped for.

The work as described is only the beginning of many interesting and varied problems of air-transport communication which will present themselves in the near future, but an idea is given of the problems confronting the operator of commercial planes in the installation and maintenance of two-way radiophone equipment.

Discussion

WESLEY L. SMITH.[2] The whole problem of shielding the ignition circuits is a very complicated one and it will probably be eliminated entirely as the Diesel aircraft engine comes into general use, as it now seems possible that it will. There will, however, still remain a low-voltage or combined low- and high-voltage generator whose output will need to be filtered to eliminate the high-frequency harmonics, and at the same time the electrical wiring and the battery will need to be shielded to absorb the last vestige of these high-frequency harmonics that may have got past the filters of the generator.

The all-metal airplane will eliminate most of the bonding that is now required. The control wires and all moving parts will be the only ones that need treatment in that type of plane. Radio reception in the all-metal plane is also much better than in any other type because the metal parts of the plane serve as the counterpoise or ground, and increasing the capacity of this counterpoise will automatically increase the signal strength

[2] Superintendent, Eastern Division, National Air Transport, Inc., Cleveland, Ohio.

received. This has already been demonstrated by practical flight tests in daily service operations.

High-frequency or short-wave radio communication has not proved as reliable in the eastern half of the United States as it has in the Far West, there being considerable skip distance and blind-spot effects. Because of the results that are obtained between San Francisco and Reno, they cannot be taken as conclusive for general use in this country.

The use of two separate receivers for high-frequency and intermediate-frequency reception is not now necessary since at least two manufacturers are building receivers with two systems of coils, oscillators, etc., so that the pilot, by throwing a switch, may receive in either part of the radio spectrum. This development will result in a considerable saving of weight, not only in the receivers themselves, but also in the added weight that is necessary for power supply.

Practical every-day operation for over a year with all airplanes of the National Air Transport, Inc., equipped with radio receivers has demonstrated that it is vitally necessary to have all of the radio control including tuning, volume, on-and-off switch, and the plug-in for the head phones located in one unit on the right side of the fuselage where it is easily accessible. The pilot who is flying an airplane where he really needs radio aid will usually be flying through the clouds, and then he has no time to hunt around the cockpit for the various controls that he may need to use. If he is flying in clear weather and has a lot of time to do these things, he has no need for radio.

The "Phonette" type of head phone, which fits inside of the pilot's ears, requires that a cast be made for that particular pilot's ears. As a result, these phones are not interchangeable, and were not considered for our daily operations on that account. The ordinary type of head phone that was commercially available was decidedly too heavy and too inefficient for aircraft use. However, a very light type of head phone is manufactured by the Dictograph Company for use with its dictographs. By changing the number of windings on these to get the proper impedance a very excellent type of phone results. After trying a great many methods it was finally found that these could best be mounted in pockets on the outside of the pilot's regular helmet and located directly over his ears. A small hole is then cut in the helmet opposite the opening to the ears and the results are very satisfactory. It has, however, been found very necessary to line these head-phone pockets to prevent the phones from freezing and becoming inoperative in sub-zero weather.

The matter of power supply for radio transmitters and receivers seems to be very much an open question. Batteries are heavy and a sufficient number cannot be carried for transmission purposes. A dynamotor, which is in reality a motor generator, is very inefficient because the current that is used must first be generated at low voltage and translated from electrical to mechanical energy and back again at a higher voltage in the dynamotor. There is an extra loss of efficiency in going through the dynamotor which can and does add both weight and power to the equipment that is needed. A wind-driven generator cannot be used, because its driving propeller would become unbalanced with a coating of ice in winter and thereby tear the generator loose from its mounting. The double-voltage generator which converts mechanical energy directly into the two voltages needed for filament and plate current seems to be the proper answer. A clutch and a third winding on the armature will permit its being used as a dynamotor on forced landings for power supply when the motor is dead. It is true that there are still some mechanical difficulties with the double-voltage generator, but it is felt that these can be eliminated. It is the lightest combination available that will provide the necessary power for the radio transmitter.

The radio interference caused by the motors of the teletype machines has been very satisfactorily eliminated by the American Telephone and Telegraph Company. This did cause some concern, but the proper combination of chokes and condensers properly to filter out the interference that was caused has now been found. This has all been accomplished without changing any of the motors.

It is a regular part of our radio procedure to have the radio men check the radio receivers with the motor running full on just before the plane is dispatched. A test that is made without the motor running has been found to be of little value since the motor introduces not only electrical interference, but also motor noise and vibration. The vibration may cause troubles in the radio receivers that interfere with reception and that are not present when the motor is not running. In addition to this, it has been found very necessary to check the pilot's head phones before each trip is started to be sure that these are functioning properly. Some of the trouble where receivers seemed to be weak turned out to be due to poor head phones.

C. Francis Jenkins.[3] The author's cleverness in working out the usual ignition-shielding solution of the airplane radio problem deserves commendation, but why locate the antenna in the radiation field of the engine-ignition system? There is a much better place for it, where shielding of engine and bonding of plane are unnecessary. Behind the fire-wall of the plane is an electric, interference-free shadow. An antenna located in this shadow does not, therefore, pick up any ignition trouble. And in the wake of the plane the antenna can be as long as desired, held taut by a miniature wind-sock. With this long signal collector, a simple 25,000-gain receiver is more satisfactory than a 6-ft. mast and a million-gain set. And, of course, the long antenna also makes a good radiator for the transmitter.

With the aft-flying antenna there is, therefore, no special problem in aviation radio. The installation can follow usual ground-station practice for the same wave lengths. And it is certainly more sensible to shield the radio equipment than to shield and bond the whole ship and power plant.

With a Zeppelin twisted, neutralized lead-in, the antenna can be flown as far aft as desired. And with a wind-sock to pull the antenna out, and an automatic, wind-driven rewind, the installation is still suitable for a one-man ship.

The writer flies a four-place cabin plane with a J-5 engine, 225 hp. The engine is not shielded, nor is the plane bonded in any way. With this plane to work with, the facts herein stated have been verified in actual practice.

Frankly, as a radio engineer, the writer would have felt chagrinned to have been obliged to ask the engine manufacturer to help work out his radio problems. Fortunately he did not need to do so.

Better than all else, this construction does not require the nuisance of a shielding which confessedly jeopardizes the lives of passengers by ignition short-circuits which involve forced landings. Ignition harness and plane bonding is absolutely not required in aviation communication.

Author's Closure

Since the paper was written, numerous improvements have been made on the equipment, mainly in the type of harness used on the ignition shielding. The present type of harness now consists of a double horseshoe or distributor ring, one located in front of the cylinders and the other behind, thus doing away with the necessity of running the leads for the front spark plugs over the end of the cylinders.

Instead of using the Belden braid covered with flexible conduit,

[3] Jenkins Laboratories, Washington, D. C.

a new type of flexible conduit has been developed which in itself is a very good shield. This conduit is also completely oil and water-tight. The fittings, instead of being clamp fittings as before, are now the screw type so that the whole harness is completely weather- and oil-proof. The pictures will show this very clearly.

Since using this new type of harness, the service time has been decreased appreciably. After original installation, it is not necessary to service this type of harness until it is removed from the motor at time of overhaul. The condition of the wire as removed from the harness, after a 300-hour period, is such that it can be used for at least another 300 hours. If it is necessary to rewire one of these harnesses, a man that is acquainted with the method can readily do this in two hours' time. This is a decided improvement over the former type, where it took one man eight hours to rewire the harness. There are many other advantages to the new type which are self-evident.

There have been small improvements made on the rest of the equipment, but nothing of great note. A few of these are shown in the pictures, however.

AER-53-8

Plywood in the Aircraft Industry

By JAMES R. FITZPATRICK,[1] CHICAGO, ILL.

Although plywood is used for nearly all parts of airplanes, its most general use is in wing construction. It is used for wing covering, leading edges, spar webs, gusset plates, cover plates, nose pieces, and tail pieces. The author in this paper tells of joining methods, results from use of plywood in shear, and takes up typical examples of its use in airplane construction.



IN WOOD, nature has given to mankind a light, strong, and stiff material of construction. These characteristics, together with workability, beauty, durability, and abundance of supply have made it a favorite with artizans ever since the dawn of civilization.

Various fabrication methods of wood have been developed, both for artistic and structural purposes. These methods of fabrication have been characterized in general by an overabundance of strength of the individual wood members and by comparatively weak jointing of the several members. With the advent of the present-day development of light-weight transportation mediums, notably in the automotive, electric-railway, ship, and aircraft industries, more balanced constructions have been needed and found. The weights of individual members have been reduced, and the strength of the ties between members increased.

As examples of this change may be mentioned, (1) the use of molded-plywood panels for the roofs of buses and street cars, (2) the use of plywood panels for stateroom partitions on ships, and (3) the use of diagonal plywood panels for the webs of aircraft spars.

Laminated wood and plywood were originally developed to increase the artistic appeal of furniture. Thin veneers of figured woods were glued to ordinary woods, and often successive slices were matched together to give an attractive grain pattern. This art has been well developed and is much used today. The glues used for this purpose, however, were satisfactory only because the finished product was usually well protected from extreme atmospheric changes. They did not possess good weathering qualities.

[1] Vice-President and Secretary, Haskelite Manufacturing Corp. Mr. Fitzpatrick received his education at Cornell University and Rensselaer Polytechnic Institute, graduating from the latter school with a degree of Civil Engineer. He was construction engineer with an automobile company in Detroit, later vice-president and general manager of the Grand Rapids Hydraulic Company. During the World War, he was factory manager of the Haskelite Manufacturing Corporation, which turned out aircraft plywood for the United States, England, and France. Mr. Fitzpatrick is a section Vice-President of the Aeronautical Chamber of Commerce, is chairman of the Accessory-Material Section, and is a member of the Show Committee and Membership Committee. He has been on the aviation committees of the Illinois Manufacturers Association, Chicago Association of Commerce, and the Chicago Rotary Club. He has just completed a term as chairman of the Manufacturers Section of the Automobile Body Builders Association, and was recently made a member of the executive committee of the American Electric Railway Association. A former trustee of the Rensselaer Polytechnic Institute, he is also a member of the Society of Automotive Engineers, American Society of Civil Engineers, and the Aeronautical Chamber of Commerce.

Presented at a meeting of the Aeronautic Division with the Chicago Section, Chicago, Ill., August 25, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Plywood as an Engineering Material

Plywood is, of course, no stronger than the glue which binds its veneers, and it was not until the introduction of water-resistant glues that plywood could be considered an engineering material. It is understood that an engineering material has definite physical properties, such as weight, strength, elasticity, and hardness. It will have properties which can be relied on to such an extent that structural and other parts can be designed from this material so that their strength and other properties will be appropriate to the loads which the parts are expected to carry or to the service which they are expected to give. For example, shingles are not used in design to carry particular loads; they are not structural parts and cannot be considered an engineering material. On the other hand, the rafters that support the roof are made sometimes of one size and sometimes of another, depending upon the loads that must be carried and upon the manner in which they are to be supported. Rafters are structural parts and they may be considered engineering materials.

Wood is a non-homogeneous material. Laboratory experiments show that the strength of wood along the grain averages about seventeen times as great as the strength across the grain. This fairly characterizes wood in its ordinary dry condition just as it is found anywhere. The purpose of plywood is to meet this deficiency by cross-banding, which results in a redistribution of the material. Thus in a plywood panel with a large number of plies the mechanical properties in the direction parallel to the face grain are almost the same as those at right angles, and the greater the number of plies, the more nearly homogeneous is the finished product.

A glue bond has been developed with blood-albumin glue which gives a shear-test value, an average taken from thousands of tests, of 370 lb. per sq. in. in dry shear and 240 lb. per sq. in. in wet shear (after the test specimens have been soaked for 48 hr.). With an adequate glue joint and selected veneers it is possible to predetermine the physical characteristics of any given construction of plywood. Engineers have found it admirably suited to carry an appreciable load in any direction. A few of plywood uses in transportation fields were mentioned before, and in every case it has replaced heavier and less adequate constructions.

In the aircraft industry, although plywood is found in practically all parts of a plane, its most common applications are in wing construction. It is used for wing covering, leading edges, spar webs, gusset plates, cover plates, nose pieces, and tail pieces.

Joining Methods

Fig. 1 shows a joint in a wing covering at a rib. It illustrates the use of plywood for gusset plates and wing covering. A simple lap joint is used for transmission of tensile stresses by means of shear. If a joint of this type is used, and it is desirable to make the joint as strong as the plywood, it is necessary to have enough gluing surface in the overlap to make the ultimate strength in shear equal to the ultimate strength of the plywood in tension. Ordinarily a joint of this type would be backed up by other structural members rigid enough to prevent

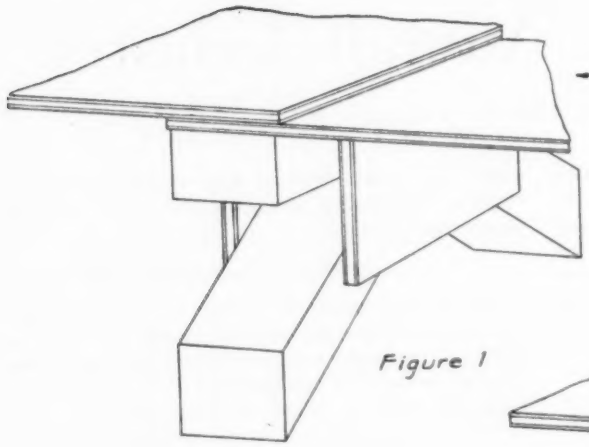


Figure 1

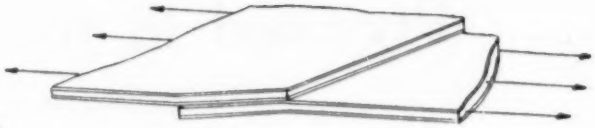


Figure 1-a

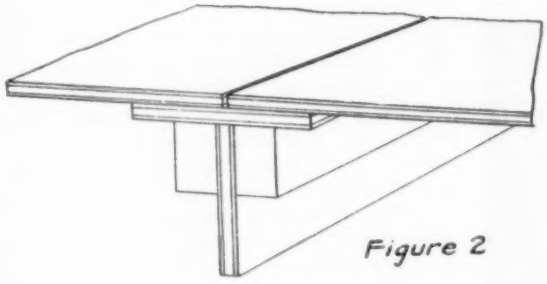


Figure 2

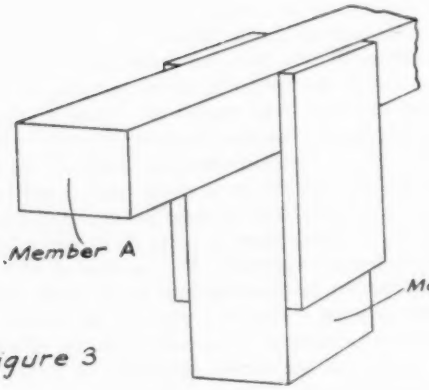


Figure 3

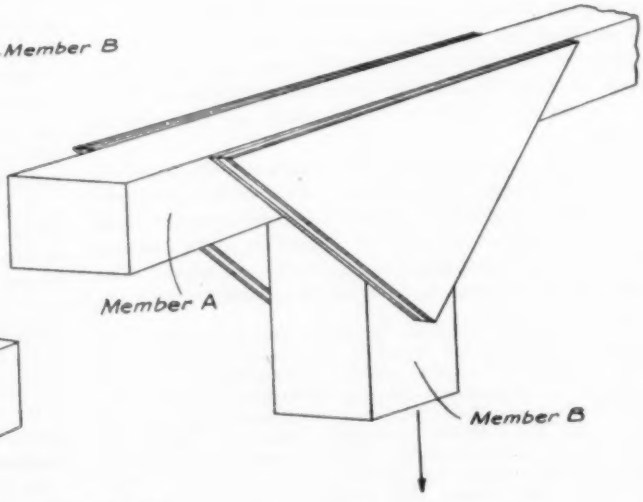


Figure 4

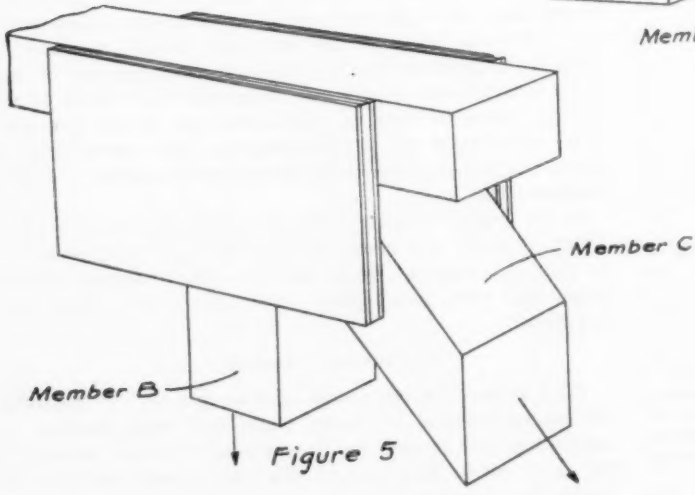


Figure 5

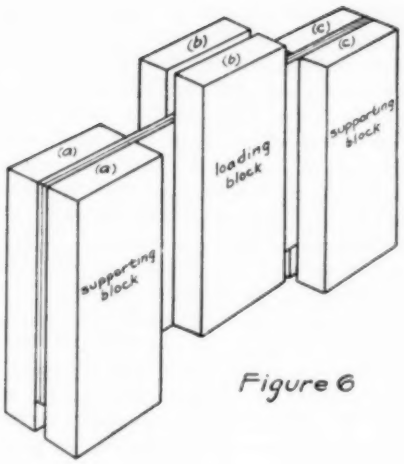


Figure 6

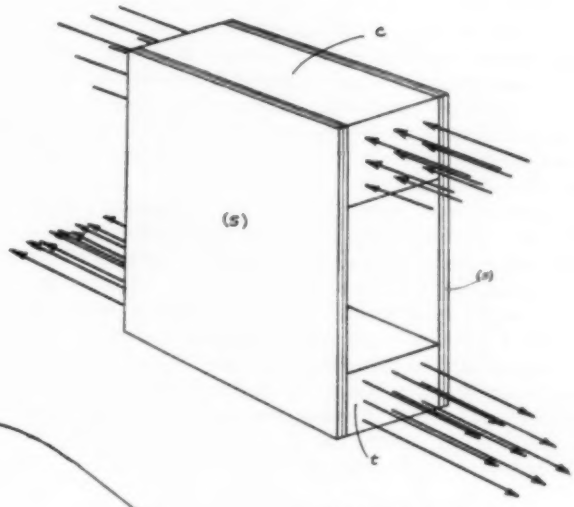


Figure 7

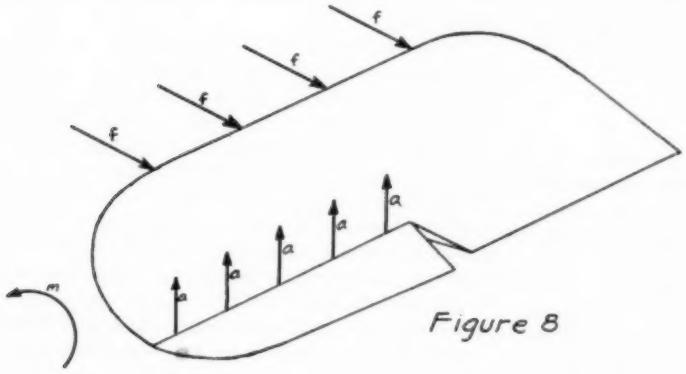


Figure 8

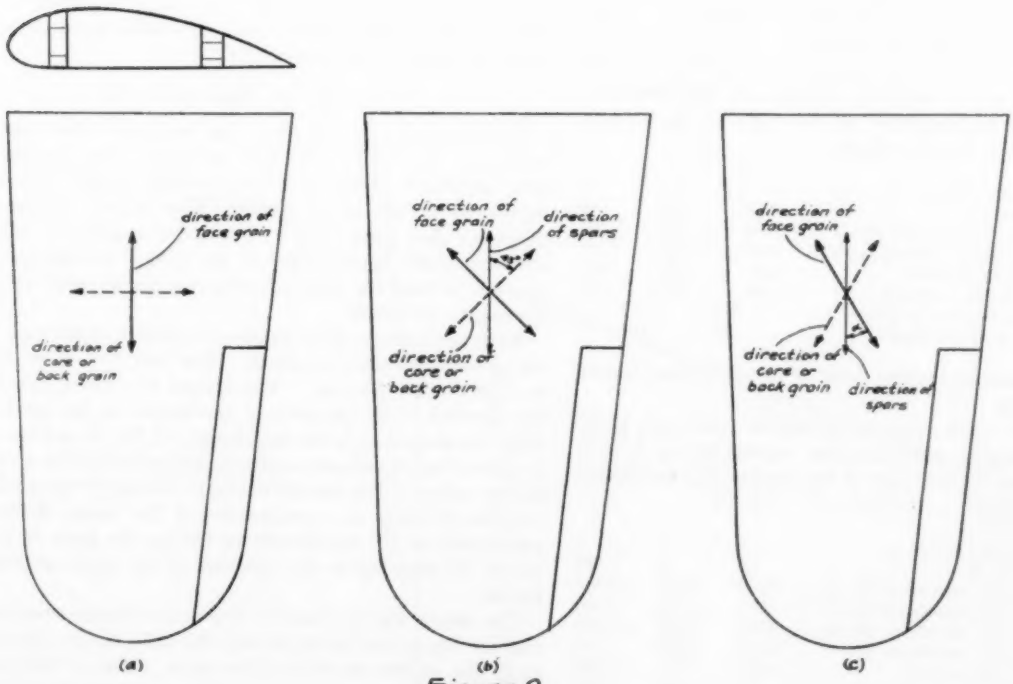


Figure 9

the bending shown in Fig. 1*a*. When such bending takes place, it increases tensile stresses on one side of the plywood near the joint.

Fig. 2 shows plywood used as a wing covering, a cover plate, and web of a rib section. It shows another method of transmitting tensile stresses from one sheet of plywood to another. The use of the cover plate decreases the bending moment and is an addition to the flange of the rib. Although this joint has the structural advantage of mitigating any transmission of bending moments into internal structural members, it is slightly heavier than the joint illustrated in Fig. 1.

Fig. 4 shows plywood gussets joining two members of a wing rib. This is a typical simple application of plywood carrying load from one structural member to another. For simplicity's sake, let it be assumed that member *B* is in tension. The gussets should be large enough so that they can transmit whatever load member *B* is to take either in compression or tension.

If it is attempted to make gusset plates from thin strips of wood, the construction would be similar to that represented in Fig. 3. It apparently has insufficient gluing surface between member *A* and the gusset plates, but the use of a wider plate would be inadvisable because of the inability of wood to carry shear.

Testing Plywood in Shear

Reference has been made to the ability of plywood to carry shear. To test this assumption, the following experiments were performed: Plywood panels were glued to wooden blocks, as shown in Fig. 6. The panels were 6 in. wide, and either 4, 6, or 8 in. long. Supporting blocks *aa* and *cc* were $^1/_2$ in. longer than the panel, and were glued to each end of the panel. Loading blocks *bb* were of similar length to *aa* and were glued to the middle of the panel. In testing, the specimen stood on the platform of the testing machine, the lower ends of the supporting blocks acting as legs, and loads were applied to the upper ends of the loading block *bb*. The clear distance between the loading blocks and the supporting blocks was $^3/_4$ in. The upper parts of *aa* and *cc* were connected by 2-in. strips of $^1/_8$-in. plywood glued to their faces, but not touching the blocks *bb*. These bracing strips are not shown in the sketch.

The results with $^1/_8$-in., two-ply mahogany plywood glued with blood-albumin glue in the ordinary manner, i.e., with the grain running parallel or perpendicular to the edges of the loading and supporting blocks, were as follows:

Height of plywood panel tested, in	4	6	8
Length in single shear, in	8	12	16
Maximum load, pounds, specimen No. 1	1240	1700	2750
Maximum load, pounds, specimen No. 2	1450	2140	2210
Average maximum load, pounds	1345	1920	2480
Strength, lb. per in. in single shear	168	160	155
Load increase for 2 in. increase in height	575	560	
Strength, lb. per in. in single shear	144	140	

The foregoing table shows the result of six specimens, two in each of three heights.

The results with $^1/_8$-in., two-ply mahogany plywood, glued with blood-albumin glue, with the grains running 90 deg. to each other and at 45 deg. to the edges of the loading and supporting blocks, were as follows:

Height of plywood panel tested, in	4	6	8
Length in single shear, in	8	12	16
Maximum load, pounds, specimen No. 1	3250	5075	6535
Maximum load, pounds, specimen No. 2	3410	5515	5000
Maximum load, pounds, specimen No. 3	3085	4360	7250
Maximum load, pounds, specimen No. 4	2240	4435	7810
Average maximum load, pounds	2996	4846	6649
Strength, lb. per in. in single shear	374	404	416
Load increase for 2 in. increase in height	1850	1803	
Strength, lb. per in. in single shear	462	451	

The preceding table shows the result of 12 specimens, four in each of three heights. The object of using three heights was to obtain a value for shearing strength which would be largely independent of the form of the test specimen.

In the lower part of each table, two estimates of the shearing strength are obtained by comparison of maximum average loads by different heights of panels. The final results for shearing strength obtained from these tests may be recorded as about 140 lb. per in. in single shear when the grain of the plies was parallel to the edges of the loading blocks, and about 450 lb. per in. in single shear when the grain of the plies made an angle of 45 deg. to the edges of the supporting blocks.

It is believed that this value is low as an average if it is to be used for a basis of designs for box beams, in which all edges of the panel are well glued. Each test specimen has four free edges; the strength near these edges should be smaller than elsewhere. This variable has been partly eliminated by taking differences as mentioned in the foregoing paragraph. In a test specimen, however, the stress is likely to be largest at the top and bottom.

The deflections in the two experiments were also taken, and they seemed to indicate that the deflection due to shear in the case of a diagonal web grain is not more than half the deflection in the case of ordinary plywood. It is interesting to note that $^1/_8$-in. solid mahogany would have roughly 100 lb. of shearing strength parallel to the grain. The strength of plywood, especially diagonal plywood, is used to advantage in the box-type spar as illustrated in Fig. 7. The tension and compression in such a spar is mainly taken care of by means of the spruce longerons *t* and *c*, while the shear is taken care of by means of the two-ply diagonal plywood *s*. Constructions of this type have been found light, stiff, and strong. Although diagonal plywood is particularly appropriate for spar webs because of its strength and stiffness for carrying loads in shear, it is not always found advisable to use it with a uniform thickness from end to end of a panel. It is evident that the shear is not uniform in some wing beams, particularly those of a monoplane, and some manufacturers meet this problem by increasing the strength of the web in various parts. This is done by adding extra ply or plies where the extra thickness is wanted.

Plywood for Wing Covering

Plywood for wing covering was originally developed as a substitute for fabric. At first extremely thin constructions were developed, some with plies as thin as 0.01 in. It was considered merely as a covering with greater strength and toughness than fabric, and no attempt was made to utilize its strength in the construction of the wing. At present $^3/_{64}$-in. plywood is used for wing covering and its physical properties are used to advantage.

There are four groups of stresses to which plywood wing covering is simultaneously subjected. The first is a stress induced by direct wind pressure. The second is tension, considering the plywood as an extension of the flanges of the spars. The third is a shear due to the drag forces *f* of Fig. 8, and the fourth is a shear due to moment *m*, developed by forces *a* as a result of aileron action. The tensile strength developed by considering the wing covering as a continuation of the flanges of the spars would indicate the importance of having the grain of at least one of the plies run in the direction of the spars, as shown in Fig. 9-*a*.

The shears due to drag and due to the aileron action seem to indicate the advantage of having the grain of the plywood run at 45 deg. to the direction of the spars. This is shown in Fig. 9-*b*. A combination effect might be produced by having the angle of Fig. 9-*c* less than 45 deg.

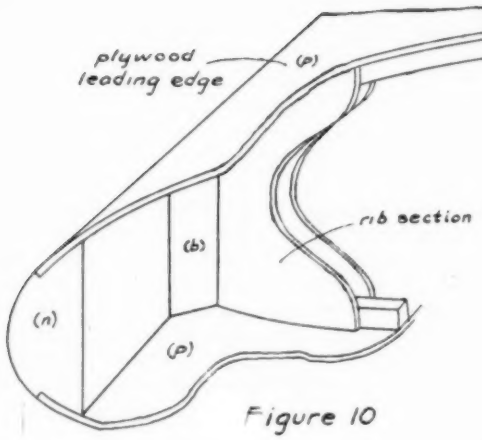


Figure 10

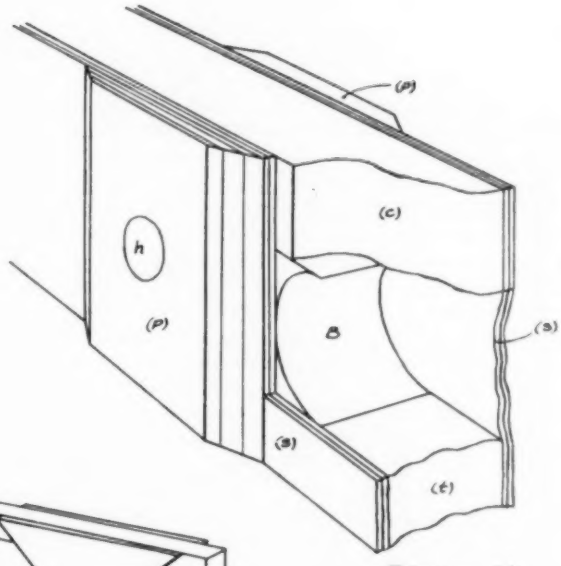


Figure 11

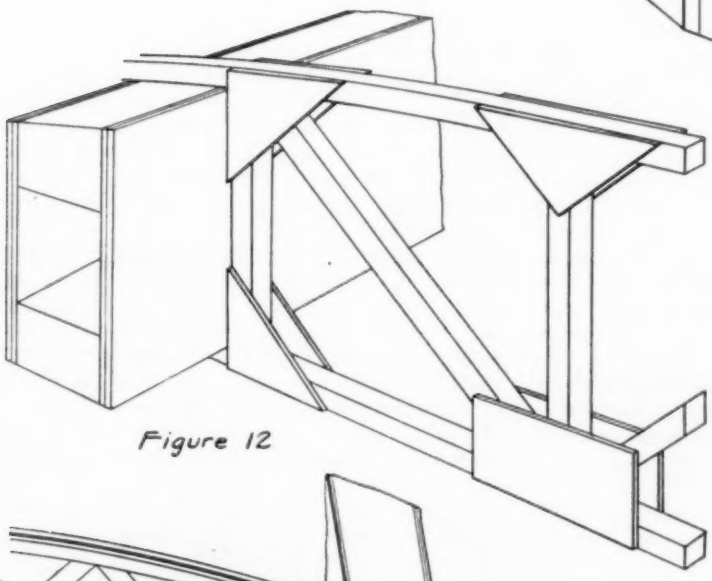


Figure 12

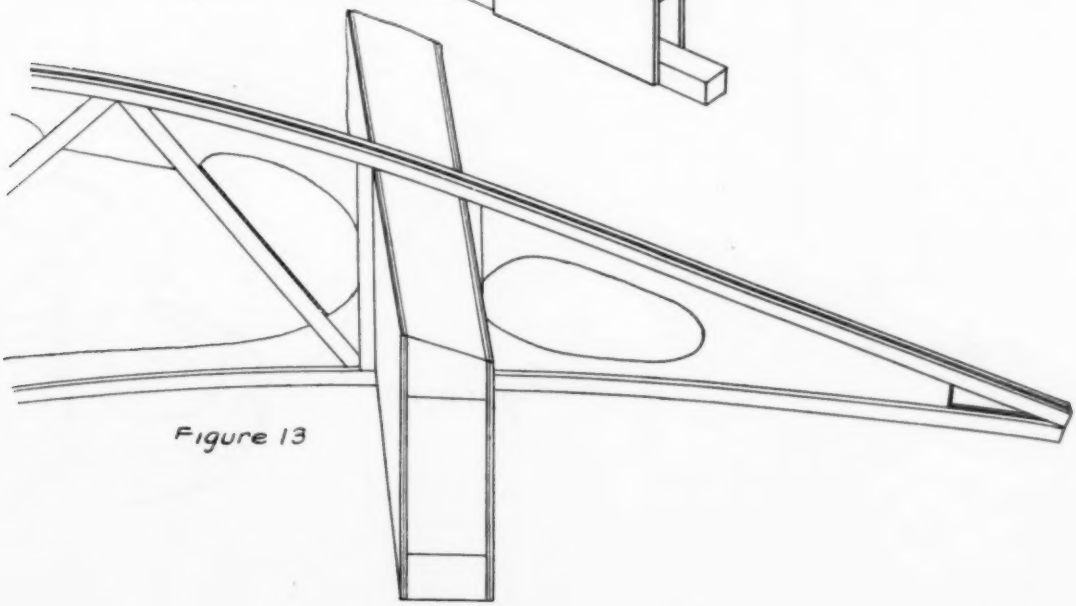


Figure 13

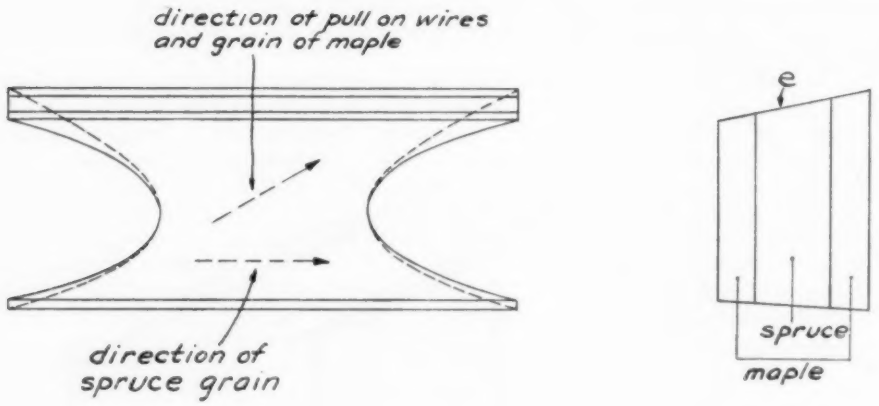


Figure 14

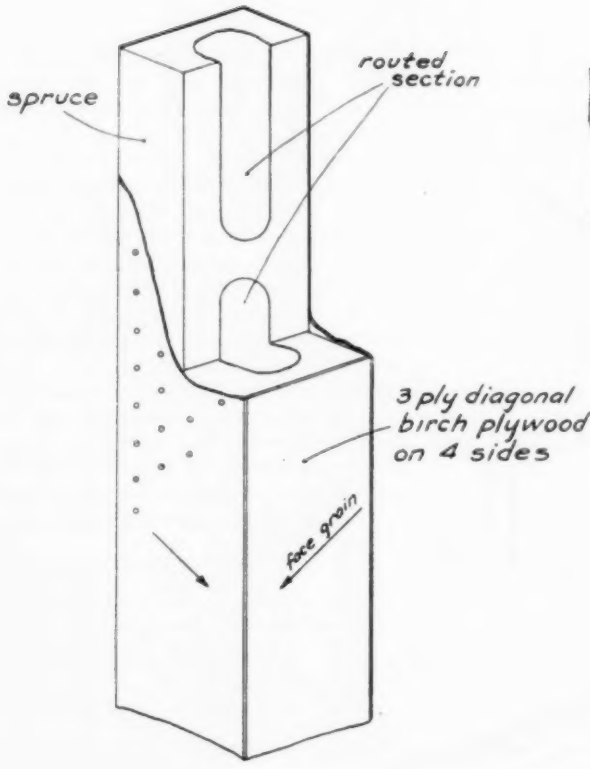
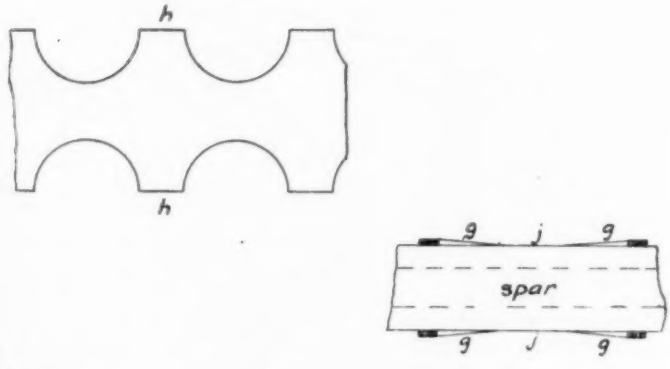


Figure 15



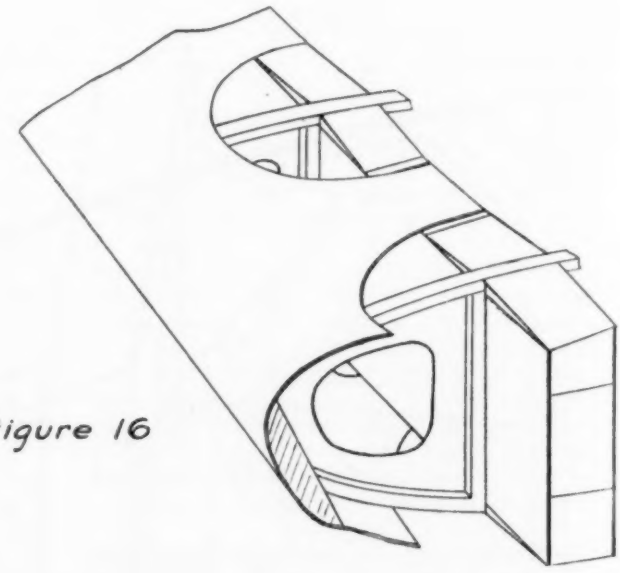


Figure 16

Typical Examples of Plywood in Wing Construction

Fig. 10 shows a typical wing-nose construction, consisting of a solid nose piece *n* tied to the ribs by means of blocks *b*, and covered at least to the front spar by a plywood leading-edge covering *p*. Such constructions have been considered capable of resisting impact of solid particles which may be in the air, and similar constructions have been found successful by a large number of the airplane manufacturers.

At the beginning of the paper, the importance of plywood as a medium of transmitting loads from one member to another was emphasized. Fig. 11 shows a very good example of such a use of plywood; *t* and *c* show the flanges of a spar, while *s* shows the diagonal plywood webs of the spar; *b* is a spacer block, and *p* is a plywood reinforcing member. Through the entire beam, hole *h* is drilled for a bolt fastening. Plywood reinforcing members *pp* are used to prevent splitting of spruce members adjacent to them. They transmit the load from the bolt to a much wider loading surface.

Fig. 12 shows a wing-rib construction and its attachment to a spar. Plywood is used for gusset plates in the rib and for webs in the spar. Fig. 13 shows another wing-rib construction and its attachment to a spar based on the same principles illustrated in Fig. 12, but different in appearance. It illustrates another method of utilizing plywood as a shear-carrying member of a rib.

Fig. 14 illustrates a spacer block used between the upper and lower flanges of a spar. It is designed to carry a load from the spar to other structural members. This particular type is used at the point at which the strut and lift wires are applied. It is a built-up three-ply member. The face plies are of maple with the grain running in the direction of the pull of the lift wires. The core is spruce with its grain running along the axis of the spar. This connection member shows the importance of large gluing surfaces. The load is transmitted from the flanges of a spar to the spruce core of this spacing block over the large surface of *e* and from the spruce core to the maple faces again over a large surface. The loads are taken through bolts, mainly from the maple of this block, to another structural member. The reason for this is clear when it is considered that even though the bolts run through the entire block, the load carried along the grain of the maple will exert much larger stresses on the contact surface of the bolt and the maple than on the contact surface of the bolt and the spruce whose grain is not in the direction of the load.

Fig. 15 shows a section of a rudder beam. It must carry shear and bending moments applied from all directions. It consists of a spruce beam with a rectangular cross-section, partially routed out in the center and covered on the four surfaces by three-ply diagonal birch plywood. The spruce carries the tension and compression of the bending moment while the diagonal plywood carries the shear. The resulting beam is extremely strong and light.

Where fabric is found for wing covering, plywood is usually employed for a leading edge. An interesting example of the manner in which plywood is cut for this use is shown in Fig. 16. The protruding parts *h* are glued to point *j* of the spar between the small filler strips *g*. These are glued to the spar in order to taper the fabric gradually from rib to spar. The result is a nice, smooth airfoil section. Plywood leading edges present a smooth contour and permanently retain their shape under any weather conditions. The face grain is parallel to the edge, and the cores are made slightly thicker than those used in ordinary construction in order to give the desired stiffness. Leading edges are given a desired curvature by boiling the plywood in water for a suitable length of time and then bending it wet around special forms. These forms have a system of clamps for holding the panel in place. When the panel dries on the form it has acquired the desired shape and will not change under ordinary conditions.

In this paper only a few of the uses and possibilities of plywood are touched on. If an attempt were made to list all of its applications, such a list would contain at least 50 different uses. Plywood is found in some of the earliest planes. In the World War it received a decided impetus along with other aeronautical constructions. Since that time it has been used in increased quantities from year to year in spite of inroads of other materials. With the advent of the all-metal ship, it was thought by many that other constructions were doomed, but plywood has maintained its place. Plywood wings are used with some all-metal fuselages and even in all-metal ships it is used in combination with metal for interior paneling and flooring. Plywood with metal glued to its surface is stronger than sheet metal of the same weight and in addition it provides insulation against both heat and sound. Although plywood has been mentioned in this paper principally as a structural material, it is widely used in transport ships for decorative and insulating purposes. These additional uses are mentioned merely to indicate plywood's wide scope. Its importance in the aircraft industry is due to the fact that it is an engineering material whose physical properties are known and can be depended upon to perform the tasks for which it is intended.

The Layout and Design of Modern Airplane Plants

By G. W. PLAISTED,[1] SEATTLE, WASH.

Plants for the manufacture of airplanes are in their design subject to the same general principles as are applied to all factory buildings. They must meet the requirements of straight-line production, flexibility, ample daylight, analysis of material handling, and the special needs of individual departments. Changes in methods of manufacture or changes in conditions should not affect a building so laid out originally as to permit of a flexible straight-line production process nor affect the maximum return on the investment. The author exhibits the general plant layouts and flow diagrams of four existing airplane plants and a typical cross-section of an economical factory building that might be converted to airplane manufacturing.

THE fundamental principles applied to the design of all factory buildings are obviously also applicable to the airplane manufacturing industry. The essential requirements that must be satisfied in the design of all manufacturing buildings are as follows:

(1) Straight-line production
(2) Flexibility
(3) Ample daylight
(4) Analysis of material handling
(5) Special needs of individual departments.

The objective of the design of an airplane plant or any other modern manufacturing plant is, briefly, to provide a straight-line production process so flexible that the buildings can be used satisfactorily in spite of change of conditions and methods of manufacturing, at the same time providing housing for the industry which will result in ultimate maximum return on the investment.

The factors involved in the design of an airplane factory may seem to the layman so numerous that it would be practically impossible to attain the ideal, and yet it has been proved in many industries, including aircraft manufacturing, that the ideal can be closely approached if the fundamentals are carefully analyzed.

The Curtiss plant, designed and constructed by the Austin Company in 1917 for the manufacture of airplanes for use in the war, was so flexible that not only could it readily be used for the manufacture of airplanes today, but the design is such that it could be used successfully for many other industries.

Building "A" of this plant was composed of nine aisles approximately 100 ft. wide and 700 ft. long. The side high aisles in Building "A" could either be equipped with crane runways or with a mezzanine over either part or over the entire area. These side aisles could be used for assembly or, in fact, almost any part of airplane manufacturing. Furthermore, these aisles would accommodate the assembly, crating, storage, and shipping of ships in crates on flatcars.

This plant stands out rather conspicuously for the way it has anticipated manufacturing requirements of even today. The reason for this is that this plant layout was so flexible that all kinds of airplane manufacturing in either wood or metal could be carried on in Building "B" and all kinds of assembly, crating, storage, or shipping could be done in Building "A," and, conversely, Building "A" would make an ideal building for ware-

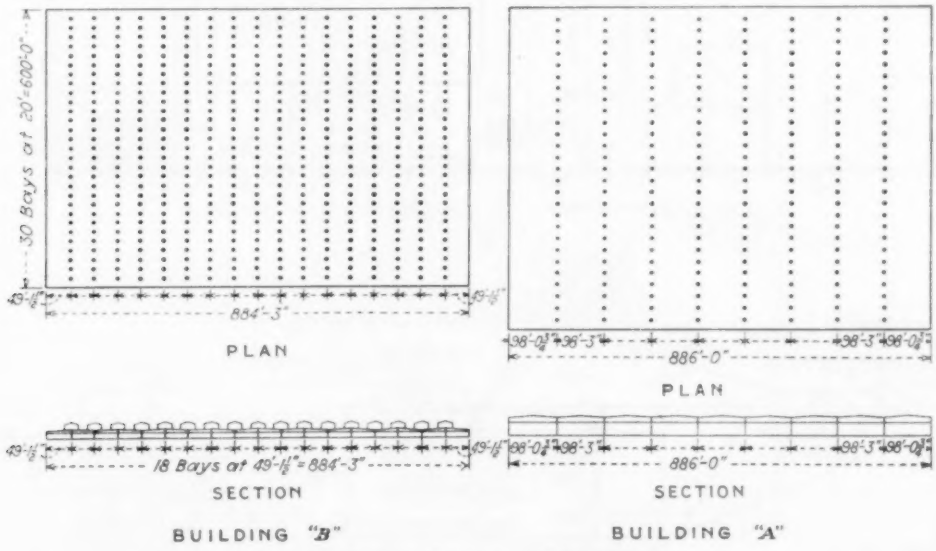


FIG. 1 CURTISS BUILDINGS "A" AND "B"

[1] The Austin Company.

Contributed by the Aeronautics Division and presented at the National Aeronautics Meeting of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, Seattle, Wash., May 28 and 29, 1930.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

housing, for receiving, storing, and assembling ships for local fly-away delivery. (See Fig. 1.)

The aircraft factory designed and constructed for the Navy Department at League Island Navy Yard at Philadelphia in 1917 was similarly flexible and satisfies the requirements of the manufacturing of airplanes of all kinds today, with minimum change during the period since it was constructed, almost as well as it did when originally occupied.

The problem of the airplane manufacturer during the past 15 years has been the problem which has existed and confronted every new industry. It has been impossible for the industry as a whole to foresee accurately the opportunities for profit ahead of it, and it has been extremely hazardous to attempt to forecast the volume of business that would be available even twelve months ahead.

Companies like the Boeing Airplane Company, leaders in the industry today, were started with a group of three or four aviation enthusiasts who were willing to take a sufficient gamble on the future to invest their capital while the industry was in its infancy.

At the beginning of this period the problems of housing the manufacturing utilities of the industry were minor and were little considered, as the objective was the design and the construction of the airplanes themselves. Government orders during the war and since that time to a considerable extent furnished sufficient volume in the industry, so that the number of ships produced annually in the manufacturing plants of the country required more attention to production costs than before.

It is true that the demands of commercial aviation have increased remarkably during the last five-year period, and with

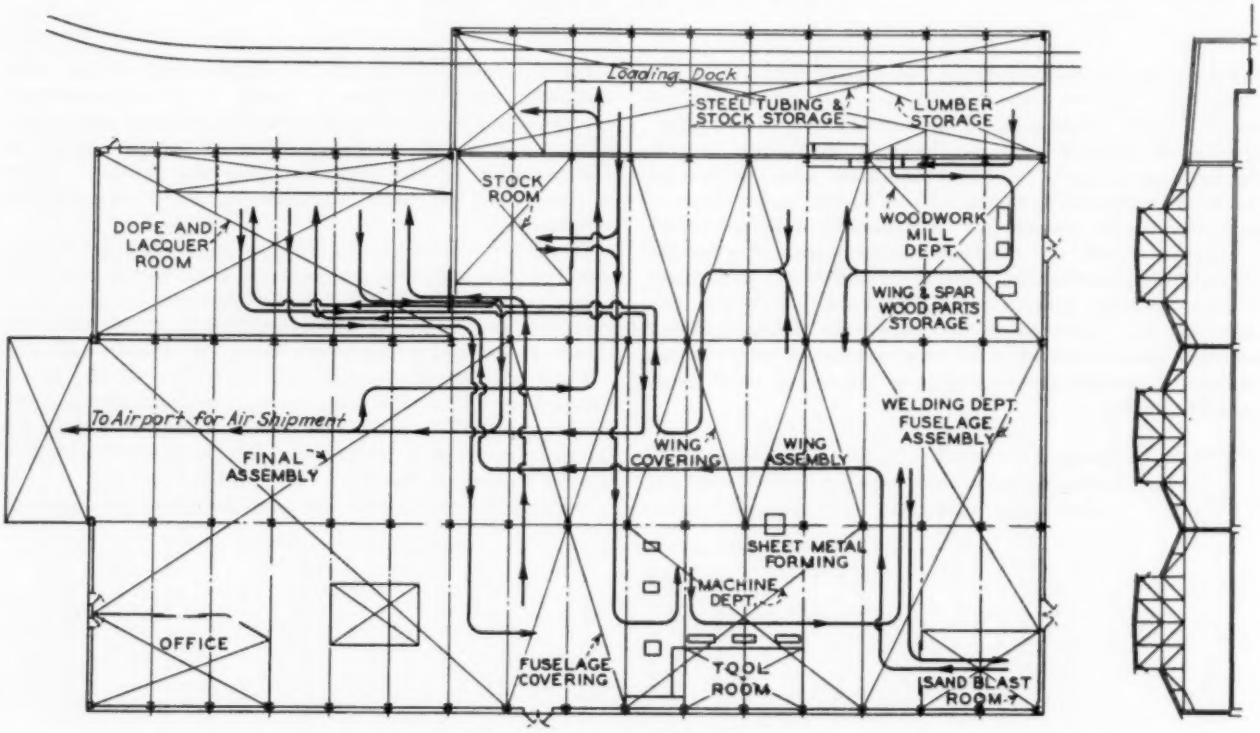


FIG. 2 TYPICAL PLANT "A" FOR 1500 AIRPLANES ANNUALLY

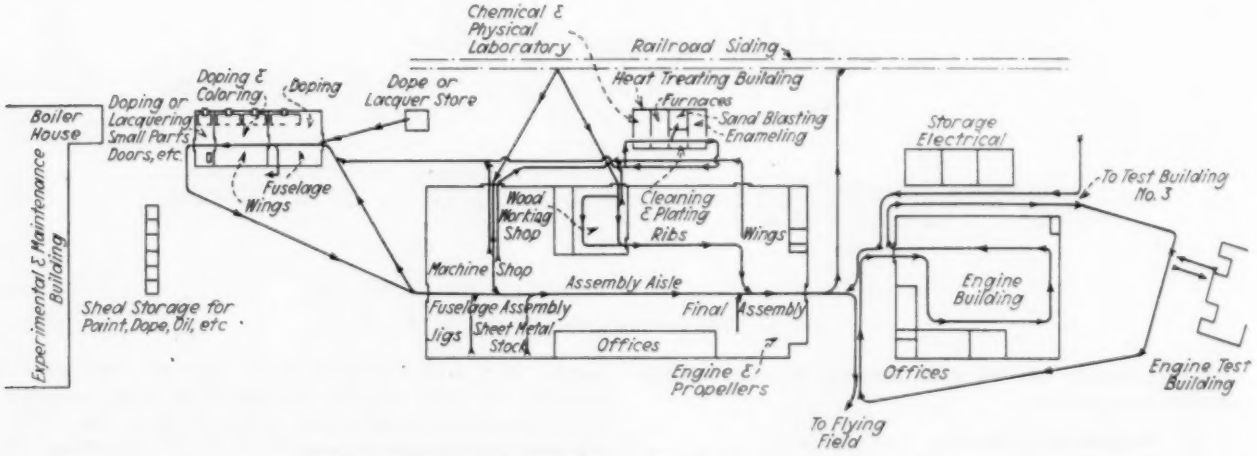


FIG. 3 TYPICAL PLANT "B" FOR 2200 AIRPLANES ANNUALLY

this condition very naturally came the problem of increased competition, and this condition in turn tended to fix the selling price ever closer to the cost. Mass production, in the real sense of the word, as it is applied to the automobile, however, has never been reached in the airplane industry, and there has been a considerable tendency on the part of airplane factories to neglect to a large extent the important factors which tend to lower production costs.

This condition may be more readily seen by an analysis of the plant layouts of a few of the manufacturers of commercial planes.

Plant "A" is designed for a maximum production of approximately 1500 commercial airplanes annually, with about 300 employees. It will be noted from the flow sheet that the main plant building is practically square, resulting in a complicated routing system as the material moves through the plant toward assembly, involving considerable back-tracking. The flow sheet indicates quite clearly that the fundamental condition of straight-line production was not satisfied in this plant. At the same time it must be recognized that this square building results in minimum building costs on account of the small wall area per square foot of floor space. (See Fig. 2.)

Plant "B" is designed to produce 2200 commercial planes annually. This company manufactures engines and cabin planes at this plant, also a considerable number of custom-built planes. The layout of this plant is an example of the other extreme, with too many buildings for specific purposes and with the obvious disadvantages resulting from handling between buildings, as well as a layout which is not as flexible as when the industry is housed in one large building. Moreover, this type of layout is wasteful of land, and the large wall area per square foot of building area obtained results in increased capital expenditure for a given amount of manufacturing space. This flow sheet clearly indicates again that the conditions of straight-line production have not been satisfied. (See Fig. 3.)

Plant "C" is designed to produce approximately 1100 commercial airplanes annually. It is composed of several buildings in parallel with closed passageways between the buildings. The system of manufacturing is departmental, as is indicated by the flow sheet. The amount of back-tracking is comparatively small. The flexibility provided in this layout is obviously better than in Plant "B," though it is not as great as in Plant "A," and the production process in general is the most logical of the three layouts presented. For a given type of building the investment is large on account of the large wall area per square foot of floor area, and it is safe to say that the ground could have been more economically used and more provision made for expansion by housing the industry in one building or in two buildings connected with loading docks. (See Fig. 4.)

Plant "D" is designed to produce approximately 300 commercial monoplanes annually. The flow sheet indicates practically straight-line production and the housing of this small industry essentially in one building, and from a layout standpoint this small airplane plant satisfies more of the conditions than any of the other layouts shown. The principles of straight-line production indicated clearly in the layout of plant "D" can be applied successfully and satisfactorily to airplane factories whose business requires much larger production. (See Fig. 5.)

It must be realized, however, that the restrictions placed on engineers designing airplane factories today involve the use in most cases of existing facilities, and it is seldom practical in the design of additional buildings to increase the production of existing plants to satisfy all of the conditions of this ideal layout. As

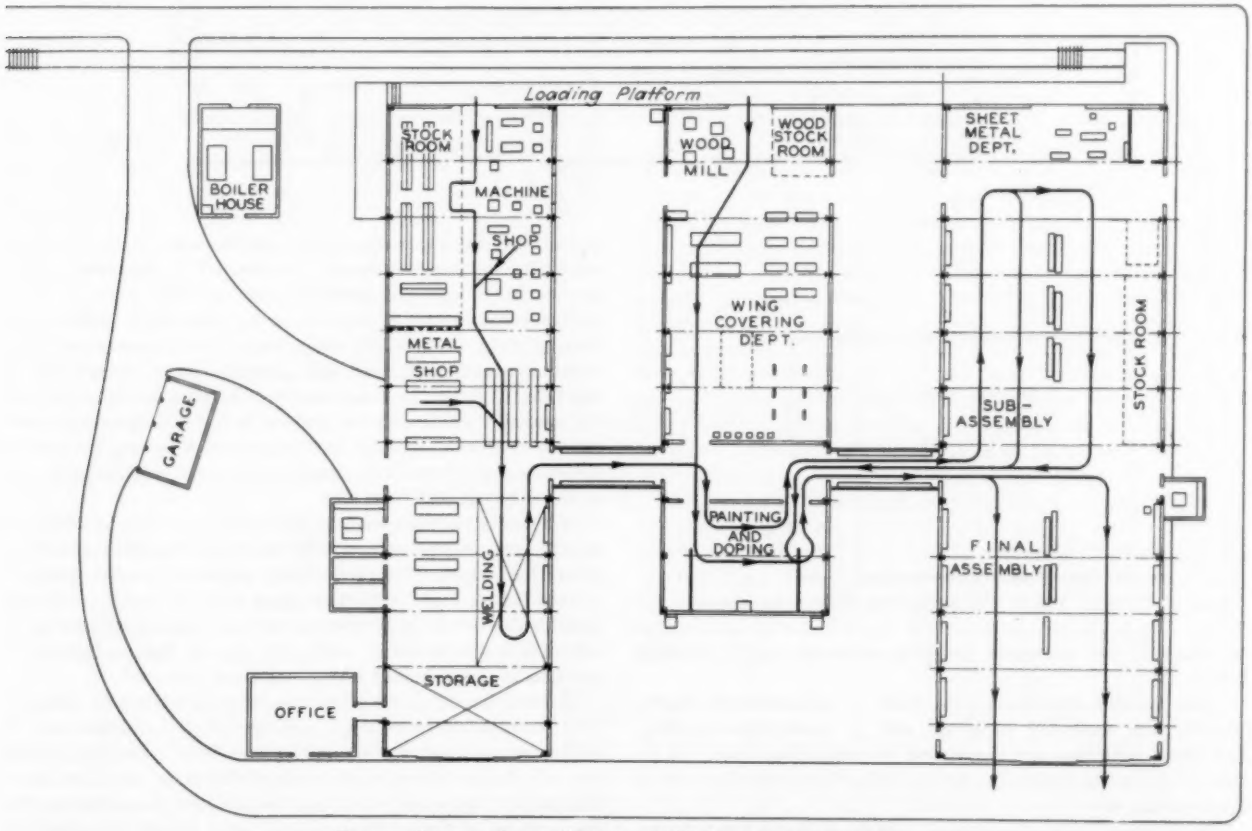


FIG. 4 TYPICAL PLANT "C" FOR 1100 AIRPLANES ANNUALLY

the industry grows, however, it is true that there will be new plants built on vacant property where all factors of the ideal layout can be considered and attained, and while it is impossible to foresee at this time whether the manufacture of airplanes will even approximate the production conditions of the automobile, it is interesting to refer to one of the most modern automobile plants and consider it for a moment as an airplane plant.

The 1000-cars-a-day plant of the Pontiac Division of General Motors is an outstanding example of close coordination of straight-line operations with material-handling equipment. This is said to be the largest factory constructed in the United States during the year of 1926. There are 35 acres of floor space. Lost motions are avoided, aisles are wide, and assembly lines in various

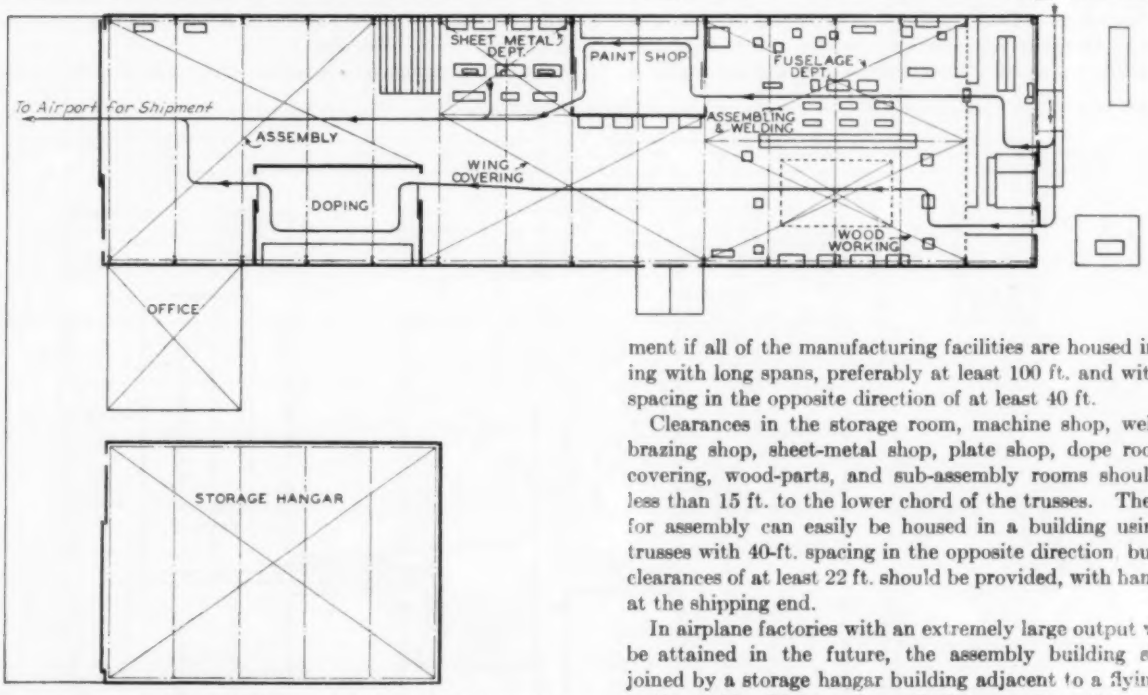


FIG. 5 TYPICAL PLANT "D" FOR 300 AIRPLANES ANNUALLY

departments are arranged to give workmen ample working space.

Three parallel structures were built—a motor-manufacturing building, an assembly building, and a car-storage building. The three buildings are connected by unloading docks 75 ft. wide for incoming material. A third unloading dock is located on the receiving side.

The diagram showing the layout clearly indicates the arrangement adopted, and it is not beyond conception to imagine this plant being converted to the mass production of the airplane instead of the automobile, in so far at least as the preliminary layout is concerned. The primary conditions of straight-line production, flexibility, ample daylight, material handling by conveyors, and special requirements of individual departments have been carefully studied and satisfied, and the plant has actually attained a production in excess of the 1000 cars per day for which it was designed originally.

Perhaps the most outstanding factor of this plant is that everything is under one roof from the time that the freight cars loaded with parts and other materials reach the unloading docks until the finished automobiles leave the shipping department.

The principles of the layout of airplane factories have, it is believed, been established, but even though an excellent layout is adopted the design of the buildings is equally important to assure economy of investment and a maximum return on the investment. The first principle to be applied to the design of the buildings themselves is flexibility.

No industry is changing so rapidly today as is airplane manufacturing. The receiving end of the plant and the storage of most materials may be provided at minimum expense in buildings of 40-ft. or 50-ft. span with column spacing in the opposite direction 20 ft. apart. The machine shop and welding and brazing may well be housed in structures using spans from 30 to 50 ft. and with column spacing 20 ft. apart in the opposite direction, but the plant will be more flexible and will represent a better investment if all of the manufacturing facilities are housed in a building with long spans, preferably at least 100 ft. and with column spacing in the opposite direction of at least 40 ft.

Clearances in the storage room, machine shop, welding and brazing shop, sheet-metal shop, plate shop, dope room, wing-covering, wood-parts, and sub-assembly rooms should not be less than 15 ft. to the lower chord of the trusses. The facilities for assembly can easily be housed in a building using 100-ft. trusses with 40-ft. spacing in the opposite direction but vertical clearances of at least 22 ft. should be provided, with hangar doors at the shipping end.

In airplane factories with an extremely large output which will be attained in the future, the assembly building should be joined by a storage hangar building adjacent to a flying field or testing field. Light and ventilation may be satisfactorily provided in all buildings by the use of continuous steel sash in the sidewalls and in monitors which will provide light to the interior portions of the buildings where long spans are used.

Material handling, fire protection, insurance costs, as compared with the required investment, and light studies and space studies in the various departments must be considered. Heating, plumbing, and locker rooms must be given the same consideration in this industry as in any other, and the problems in connection with the location of the washrooms and toilet rooms will materially affect the production cost of any plant.

It may be the simplest thing to do to provide the same heat in all departments; but the design of the heating system, while it must be made flexible, as must the system of illumination and power wiring, must take into consideration the special demands of the various departments, for it must be quite possible, for example, to maintain a temperature of 90 deg. in the dope room for satisfactory results, while a temperature of 60 deg. in cold weather in the machine shop will prove entirely satisfactory, providing proper ventilation is supplied.

The standardization of modern factory buildings makes it possible to provide the building structure itself, including the foundations, walls, and roof, at a minimum expense, satisfying the conditions of daylight and ventilation at the same time.

In conclusion, it is evident that the straight line to profits in manufacturing today is found in straight-line production methods. Many concerns have moved ahead of their industry by taking full advantage of this fact, while others have waited until forced to do so by competitive conditions. With the fundamentals of proper design and layout of airplane plants satisfied, the handicap of selling price approaching cost, as is the tendency today, is minimized. All signs indicate that the industrial concern which comes to the end of this year without a completely modernized plant will be seriously handicapped in doing a profitable business. Soon this urgent problem may mean complete new plants, perhaps in entirely different locations.

Discussion

GARDNER W. CARR.[2] Without attempting to criticize the paper, it is believed that certain points can be profitably amplified or explained. To do this a survey of this company's problem will be enlightening. Four years ago the Boeing Airplane Company was manufacturing an airplane the structure of which consisted of a welded tubular steel body with wings manufactured of mahogany plywood and spruce, covered with fabric. The following year the same type of wing construction was used by the Boeing Airplane Company, but the body structure had been changed, supplementing a considerable number of welded steel tubes with bolted dural members. The following year found a reactionary change in the body construction, although the wings remained the same. The forward section of the body was of welded alloy-steel tube construction, but the rear section of the body was made up of bolted rectangular dural tubes. The fourth year, that is, 1930, finds a still farther and more radical departure from previous practices. The wings are now fabricated from dural sheet, the ribs, spars, and covering being metal. The body, although it still retains the steel forward section, is also metal covered, the covering being applied over a skeleton of bolted dural tubes. Quite a change has taken place in the shop organization. In 1927 there was a large amount of electric welding and work in steel tubing, with the wood shop manufacturing spars and ribs and the wing room manned by cabinet makers who could handle the assembly of wood structures. The year 1928 increased the load in the machine shop, where the dural tubes and terminal blocks had to be made up, while the load in the welding shop decreased. The year 1929 found a still greater load in the machine shop, with the welding problem almost eliminated and replaced by a new type of work, that of assembling bolted dural structures. This year, 1930, brings up a problem involving almost nothing in the welding shop, a normal load in the machine shop, and a tremendous burden in the sheet-metal department. A plant not laid out to give this flexibility would have rendered the manufacturing problem extremely uneconomical if not prohibitive.

Another phase to be considered is that of size, and particularly over-all height or clearance under the roof trusses. As long as small airplanes are being built, little overhead clearance is required. In this plant, however, airplanes have been manufactured ranging in weight from 2500 to 27,500 lb. and in wing spread from 27 to 120 ft. The latest design has wings of such size that they could no longer be assembled in a horizontal position, and vertical jigs were constructed. Although the assembly department in any plant will require head room, the probable size of units to be produced in other departments must be given consideration.

From an observation of the industry and the changes which have taken place in the last ten years, the writer believes that a plant laid out in the shape of a letter E or with wings leading into a main building offers the greatest advantage. There are two problems connected with the manufacture of airplanes which require isolation of equipment—that is, electroplating or anodic process equipment, and doping or enameling. Either electroplating or anodic process involves a problem of ventilation and heat control not required in other operations. The doping or enameling processes require not only heat control and ventilation, but special protection against fire hazard. The problem of isolating these functions seems easiest to solve by housing them in a wing offset from the main building. The experience of the Boeing Airplane Company indicates that such a unit can be placed between other departments to advantage, thereby simplifying the flow lines and eliminating backhaul. In construction this wing should not be radically different from the other wings of the plant, as these requirements seem to be changing quite rapidly, and particularly the doping operation may be eliminated in the near future.

The foregoing comments are offered not as a criticism but rather to emphasize the author's point that no plant should be laid out that will not be flexible and capable of handling the varying types of construction and new manufacturing problems which are constantly appearing in the particular branch of engineering where progress is exceedingly rapid and radical changes occur almost over night.

[2] Boeing Airplane Company.

AER-53-10

Stability and Controllability in Free Flight

By PAUL E. HOVGARD,[1] BRISTOL, PA.

Making the airplane pleasant to fly is discussed by the author in this paper. Strength, durability, and performance of airplanes are determined quite accurately by analyses and tests, but the ease with which a particular plane can be maneuvered is often left to chance. Stability at cruising speeds and balance with the throttle closed and open are essential for ease of control. The author tells of various methods that can be used to test flying characteristics and make certain that airplanes will perform as desired.



IT IS NOW becoming possible to choose an airplane in much the same manner as one chooses an automobile, that is, by picking out from a directory the type and size that suit the needs, scratching off a few because they are too slow, and of the remaining, buying the one that is the most pleasant to operate. If one finds the operation of the airplane not to his taste, he will know from automobile experience that it is useless to ask the manufacturer to modify something to eliminate the queer feeling he gets coming out of a turn, but will step into the next hangar and buy a competitive airplane almost identical to the first except that he likes the way it flies better.

In one case three pilots from different parts of the country and strangers to each other informed the author that the proper way to make a turn in a certain popular cabin monoplane was to start by pushing the rudder hard in the direction of the desired turn, and when the airplane had reached a sufficient degree of bank, relieving the pressure on the rudder. The plane continued to turn without further attention. As soon as it was headed in the desired direction the top rudder was to be pushed till the airplane straightened out. The ailerons were to be used only for rough weather take-offs and landings. Such haphazard design may be tolerated when airplanes are hard to purchase, but with the strong competition in selling now in vogue, such an airplane has little chance.

While the strength of an airplane can be determined by stress analysis or static tests, the durability by service tests, and the performance by flight instruments and a stop-watch, the "flyability" is too often left to get along the best it can, and when finished is expressed in such vague terms as to have no meaning to the designer. In some cases the test pilot is reluctant to find fault with the airplane for fear others will think it was just getting the best of him, and on that, of course, he cannot afford to take a chance. Making an airplane pleasant to fly requires close cooperation and intimate understanding between the test-flight and design sections. The stability and controllability forms used by the Army and Navy serve well the purpose intended, but are of little help to the designer in this respect. The Department of Commerce instructions are very indefinite and in some cases even incorrect, as an airplane would have to be spirally stable to pass either their directional or lateral stability tests.

Determination of longitudinal stability is the most straightforward test of all; yet even there mistakes are made especially in large airplanes. The stabilizer must first be set to trim the airplane at the given engine speed, and the air speed for that engine speed recorded. The throttle and stabilizer setting remain constant throughout the test. The airplane is then nosed over to increase the air speed 5 m.p.h. and held in the required position until the air speed is constant. If upon releasing the control column the nose returns toward its original position, the airplane is stable. The test is repeated for an air speed 5 m.p.h. below the cruising speed.

Quantitative results may be obtained by measuring the forces on the control column required to hold the airplane constant at given air speeds and the results plotted or by finding the center-of-gravity location at which the airplane is neutrally stable. If a revision of any kind is made on the airplane, one or both of these methods should be used to determine the effect of the change on the stability. In case the tail surface is changed the test should be made with both the old center-of-gravity location and the new, as quite often the increase in weight of an enlarged tail plane will move the center of gravity far enough to the rear so that the stability at or close to cruising speeds will remain unchanged, though usually the stable moments will be increased at very high and very low speeds.

If the airplane is found to be statically stable by the above tests, its dynamic stability is next determined. To do this the airplane should be dived or climbed and the control column released. If the oscillations decrease and dampen out, the airplane is dynamically stable. There are very few airplanes that are statically stable that are not also stable dynamically.

Besides the stability at cruising speeds the power "on" and "off" balance should be determined. This can best be given in terms of air speeds with throttle closed and wide open with the stabilizer set for cruising and no pressure on the control column. Ordinarily, the greater the stability, the higher the power "off" air speed, a rather unpleasant condition. It is possible to maintain a constant air speed for all throttle setting regardless of the vertical position of the engine except for rapid movements of the throttle. Such an airplane will invariably be longitudinally stable.

The amount of longitudinal control necessary varies with the purpose of the airplane. All airplanes should have enough elevator to raise the tail on the take-off by the time 50 per cent of the stalling speed is reached and with the stabilizer set for cruising.

Small airplanes should have enough elevator control to get the tail fully down when landing without changing the stabilizer setting, though this feature is not necessary on large airplanes because the pilot expects to take more time for the approach preparatory to landing. It is not necessary that a large airplane be able to make a three-point landing when only partly loaded even by the use of the stabilizer as long as the landing speed in this condition is not greater than the landing speed with full load.

The lateral stability of an airplane is determined by its ability to bank itself for a turn and to level itself from a turn without the use of the ailerons, the turn being made with rudder alone. The rudder action for this test should be quite slow, and a certain amount of slip during the maneuvers must be expected.

---

[1] Research Engineer, Keystone Aircraft Corporation.

Presented at a meeting of the Aeronautic Division with the Chicago Section, Chicago, Ill., August 25, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

The amount of lateral stability an airplane should have can be set by no definite rule and must be determined by compromise. Too much lateral stability is much more unpleasant than too little. Too much means dangerous take-off and landing characteristics in gusty weather or cross wind and, worse than that, causes the airplane to be very rough and sickening in bumpy air. Too little lateral stability means mainly that the pilot must be a little more careful with the aileron control when maneuvering. There should be enough so that the pilot can follow a straight course by using the rudder alone.

The lateral control must serve two functions; overcome inherent lateral stability and provide sufficient maneuverability. If an airplane can be flown on its side at speeds not over 10 m.p.h. above the stalling speed in this position, it has ample aileron control for the first condition. The aileron control for sufficient maneuverability cannot be determined definitely, but is a matter of judgment.

The directional stability is the most difficult of all to determine accurately, and it is possibly fortunate that very few airplanes are directionally unstable. For the test a bank-and-turn indicator is very helpful. The rudder should be either held rigidly in the center or entirely released and the maneuvers made with aileron alone. The pilot must bear in mind that there will be a yaw due to the ailerons which must be accounted for in the test.

The airplane should first be banked in one direction by the ailerons. This should be done very slowly and for this start it is better to counteract the yaw of the ailerons by applying the rudder in the direction of the turn. The ailerons should then be applied in the other direction until the airplane is approximately level and the turn indicator shows zero turn. Ailerons are then either entirely released or held rigidily in neutral. If the airplane turns into the direction of the slip it is directionally stable.

It can be seen from this that the test is made with wings horizontal and the airplane side slipping either to the right or left but not turning. The test should be made with various degrees of slip as shown by the ball of the bank indicator and should be made both to the right and to the left.

The degree of directional stability can best be given by the position of the center of gravity on the mean chord at which the airplane is directionally stable with the rudder held in neutral and directionally unstable with the rudder free. Difference in stability, rudder held or free, with a balanced rudder is surprisingly small so this position can be definitely located within about 1 per cent of the mean chord. Ordinarily this position is 10 or 20 per cent to the rear of the full-load center-of-gravity position.

The rudder area required depends for the most part on the degree of directional stability except in cases of airplanes without brakes where the determining factor is the ability to taxi on the ground. The rudder area and directional stability should be so proportioned that the airplane can be flown on its side at low speeds with full top rudder and without stalling in this position. In the case of multi-motor airplanes the rudder should be powerful enough and well enough balanced so that the airplane can be flown straight with one engine idling and the other engines wide open without overexertion on the part of the pilot.

For spiral stability an airplane must have small directional stability as compared to the lateral stability and whether or not the airplane should be spirally stable is sometimes questionable because of the small amount of directional stability or the excessive dihedral required to obtain it.

To test for spiral stability the airplane is put into a normal steady turn and the controls released. If the airplane recovers from the turn it is spirally stable and if not is spirally unstable. Tests should be made both to the right and to the left, and either the vertical fin should be adjusted or the spring attached to the rudder to make the airplane act the same in right and left turns. In multi-motor airplanes the outboard-engine tachometer should be carefully checked to be sure the engines are synchronized before making the test.

For normal flying the spiral stability is closely tied in with the yaw due to the ailerons. Because of the small directional stability, pilots invariably over-rudder an airplane that is spirally stable. If aileron yaw is high this rudder movement is necessary for normal flying and the pilot is at ease. It is advantageous at times to use unbalanced rudders on airplanes that are spirally stable to reduce the tendency to overcontrol.

One of the most important considerations in making an airplane pleasant to fly is the coordination of the three control surfaces. A normal flying airplane will be defined as one which requires an approximately equal pressure on the rim of the wheel or top of the stick and the rudder to start and recover from turns, the pressures being applied or released at approximately the same time. In spirally stable airplanes a very fine degree of relationship must exist between directional stability, lateral stability, and yaw due to ailerons without making the airplane fly abnormally. Spiral stability therefore should not be required in an airplane, but of course excessive instability is not desired. Even though an airplane were spirally stable the action would be so slow that the pilot would not care to wait for it.

It is impossible to fly airplanes above a given size without balance of some type on the control surfaces. The overhang balance gives satisfactory results for elevator and rudder at small angles of attack of the surfaces, but has very little effect when the surfaces are moved through more than about 10 deg. It is possible to design a partially hidden balance that will give excellent results over the entire control movement. Frize type balance for ailerons with hinge line at about 18 per cent or 20 per cent of the chord gives excellent results and is generally acceptable.

It is quite possible that at some time in the near future the method of controlling airplanes will be entirely changed and one of the controls eliminated. Until this time comes, however, it behooves the manufacturers using the present system to so refine it that the best results possible will be obtained.

Discussion

HALL L. HIBBARD.[2] I believe that the paper is very much to the point. The modern airplane can have stability and controllability not thought possible several years ago. Aircraft properly proportioned and designed are in existence which have good stability and controllability, but there are still a larger number which do not have these desirable features; especially is this true of stability. It has been the writer's experience that many pilots are unable to discover accurately whether an airplane is stable or unstable. It is for this reason that the author's "Methods for Determining Stability and Controllability" should be studied and tried by pilots. Many of the war-time pilots became accustomed to the airplanes used during the war. As a rule these craft, although quite controllable, usually lacked stability. Numbers of these pilots, however have become accustomed to this unstability, and when buying a modern plane pay little attention to this important item. Not long ago attention was called to a pilot who had been flying an airplane which was laterally unstable and which had to be "flown" all the time. He was asked to try out the stability of a particular airplane which happened into the field and which he had never flown before. This airplane was known to have poor or no lateral stability. When the pilot returned from the test flight he was questioned about the lateral stability, and much to the surprise of all he reported good lateral stability.

[2] Project Engineer, Stearman Aircraft Co.

The answer is that he been accustomed to poor lateral stability and was unfamiliar with the method of testing airplanes for this quality. It is for such pilots that the paper is recommended.

H. RAWDON.[3] Control and stability in general cannot be expressed quantitatively, and one is usually compelled to accept the qualitative or descriptive statements of test pilots. At least there is very little in the way of generally accepted units of measurement.

It is exceedingly difficult to find a test pilot who has a sufficiently clear idea of what he is to look for and to give an interpretation and presentation of opinions of his observations. This fact imposes difficulties on the design staff, since under the circumstances the pilot will probably change his mind later on, on his first impressions, or since it is difficult to get two test pilots with a common opinion. The difficulty could be helped if engineers in general would take greater interest in flying themselves.

The Keystone Company is fortunate in having a man who is both a capable pilot and engineer.

Regardless of size, it is always desirable to have an airplane that can be landed three points under any condition of load within the range for which the airplane was designed. There is no practical reason why this should not be possible to accomplish.

Spiral stability is generally of the least value. However, excessive spiral instability is not desirable by any means. A case is known in which the directional stability was deemed insufficient and a small amount of fin area was added. The resulting directional stability was satisfactory, but the airplane was objectionably unstable in a spiral without using a prohibitive dihedral in the wings. This was a case for compromise.

A measure of control which seems desirable to obtain is the ability of the airplane to make a turn at a speed something below that of stalling speed without the aileron action causing the wing to drop. The lateral control required here would be greater than in the case cited by the author.

A. A. GASSNER.[4] The paper covers the subject thoroughly. One misses, however, the mentioning of two criterions of the stability of an airplane:

1 A ship should be longitudinally balanced in such a way that, with stabilizer and throttle setting at cruising speed, closing of throttle would bring the ship in a glide and opening of throttle would bring the ship in a climb.

2 The behavior of the ship in a stall. If small-sized planes slide off one wing when brought in a stall and go in a spiral, which is sometimes mistaken for the beginning of a stall, or if they actually go into a spin, the forces required to stop this movement will not be excessive and recovery is possible. It is, however, of imperative necessity to test all large-sized planes for their behavior with engines partially or totally throttled. A ship which behaves well in a stall, i.e., goes into a straight dive when all controls are released, can be termed as safe playing.

WILLIAM H. MILLER.[5] The author is to be commended for inviting attention to a series of very simple tests by means of which airplanes may be compared as regards stability and controllability. The mere definitions of these quantities are too often misunderstood or confused by both pilot and engineer.

From a strictly academic viewpoint the author has not drawn a sharp enough distinction between stability with free controls and with locked controls; that is, "hands off" and "hands on." However, in actual practice, the former quantity is generally the easier to determine by such tests as the author describes, and, except in the case of spiral stability, it is usually the criterion of the latter.

Although the author criticizes the standard flight-test forms of the Army, Navy, and Department of Commerce as being of little help to the airplane designer, one must venture to say that his subject article may be criticized in the same respect. For any case of instability or deficiency in control the changes in the airplane necessary to correct the condition should have been mentioned. As an example, dynamical instability with free controls may be caused by any one of the following items:

(*a*) Center of gravity too far aft
(*b*) Center of gravity too far above thrust line
(*c*) Moment of inertia in pitch too great
(*d*) Too much area of horizontal tailplane in slipstream
(*e*) Elevator area too large a proportion of total tailplane area (where the control surface is unbalanced).

Generally, items (*d*) and (*e*) are the easier to change after flight tests indicate longitudinal stability. Simply increasing the aspect ratio of the horizontal tailplane, keeping the total area constant, and increasing the area out of the slipstream will usually greatly improve longitudinal stability.

Occasionally, in military aircraft, a rearrangement of the useful load, in order to reduce the moment of inertia in pitch, will remedy an unstable condition caused primarily by item (*b*).

It can be shown theoretically that the difference in elevator angle to trim, for the same airspeed, "power on" and "power off," is a function of the built-in wing incidence. The effect is theoretically small except, for example, in the case of a tractor biplane seaplane, with wings set 3 or 4 deg. to the thrust line for quick take-off. The relation of wing incidence to difference in trim for the aforementioned conditions ought to be given considerable attention by the designer. In this connection he must of course take account principally of the relative positions of propeller and wing.

An airplane which is stable statically in pitch or yaw with fixed controls, but unstable with free controls, can be made stable sometimes in either case by the addition of a small amount of balance to the control surface in order to reduce the hinge moment.

It is most important of all, in so far as yaw is concerned, to provide sufficient rudder moment to counteract the yawing moment with fully deflected ailerons at the stall with power off. Incidentally, this is a quite difficult condition to obtain (with conventional ailerons), and at the same time maintain spiral stability at low speed, unless the airplane has a large amount of sweepback.

From the standpoint of preliminary design, all satisfactory stability calculations should take account of slipstream downwash and velocity. Many designers do not realize that these effects are so pronounced in some tractor airplanes that they account for a difference of from 15 to 35 per cent in effective tail area required to give equal degrees of stability with free elevator, for "power on" and "power off" conditions.

G. G. BUDWIG.[6] The author states that a particular airplane, the name of which is not mentioned, is guilty of being haphazardly designed because it can normally be turned with the rudder only and without applying the aileron control at all. In view of the fact that such characteristics are very difficult

[3] Chief Engineer, Travel Air Company, Wichita, Kan.

[4] Chief Engineer, Folker Aircraft Corporation of America, Hasbrouck Heights, N. J.

[5] Assistant Chief Engineer, Berliner-Joyce Aircraft Corporation, Baltimore, Md. Mem. A.S.M.E.

[6] Director of Air Regulation, Department of Commerce, Washington, D. C.

to predict, he is probably correct in his assumption that the turning characteristics were the result of accident rather than design. However, it appears that this particular feature was rather a happy accident than otherwise because it would tend to make the airplane in question much more pleasant to fly. The reason for the author's condemnation, therefore, is not entirely apparent.

When the author accuses the Department of Commerce instructions of being "very indefinite, and in some cases even incorrect," it is assumed that he is referring to Inspection Service Memorandum No. 1, which was published and circulated October, 1929. This memorandum is fairly lengthy and was written for the sole purpose of acquainting the manufacturers in as much detail as possible with the routine procedure followed by inspectors when they test an airplane for conformity with the Air Commerce Regulations. The instructions contained in the memorandum do not in any way guarantee that an airplane which meets those requirements will be the best possible airplane of the particular type, but for the purpose for which the memorandum was written it is believed that it is both definite and accurate. If there are parts of the memorandum which are not clear, we would be very glad to have any one point them out specifically, so that we might clarify them.

It is stated by the author than an airplane would have to be spirally stable to pass either our directional or lateral stability tests. This is not correct, because during the directional stability tests the wings are held level by means of the ailerons, and during the lateral stability tests the airplane is held level by means of the rudder. No part of either of these maneuvers has a remote resemblance to spiral flight, and airplanes which successfully pass these tests are not necessarily spirally stable. In fact, an airplane with sufficient stability will usually dive out of a turn and assume a level flight attitude if the controls are released.

When the author states that a faster power-off glide results from longitudinal stability, it is difficult to follow his reasoning. A longitudinally stable airplane has a negative lift on the stabilizer, which builds up rapidly as the speed increases. Of course, as the down load on the tail increases, the nose of the airplane comes up and it flattens out, at the same time decreasing its speed. It is therefore not only apparent but true that the greater the longitudinal stability, the quicker a dive will be checked. A stable airplane therefore, cannot build up an excessive diving speed. An unstable airplane, on the contrary, will frequently get its nose clear under in a dive and will fall at terminal velocity if allowed to do so.

An excessive amount of lateral stability is unpleasant in several ways, particularly in a cross-wind, as the author says. However, it is believed that an insufficient amount of stability in this direction is even more unpleasant and more dangerous when one is called upon to fly in rough, gusty air. Fighting with an airplane to keep it on an even keel is far from being the ideal in flying. The writer agrees with the author that the coordination of the three controls is a very important consideration which is too frequently neglected.

JOHN C. NULSEN.[7] There is no question that the author is correct in his contention that airplanes should be so designed as to be easy and comfortable to fly. The necessity of this feature will of course grow in importance from the manufacturer's standpoint as competition increases. Nothing is more disagreeable from the pilot's standpoint than to get into a ship and find on maneuvers that it is light on the ailerons and heavy on the flippers, or vice versa. Such conditions are due simply to lack of attention to details.

[7] General Manager, Ryan Aircraft Corp., Anglum, Mo.

Stability about the various axes is a subject that should receive very careful consideration, and a conclusion should not be jumped at. It can without question be overdone. As the author points out, too much lateral stability is hard on the passenger prone to air sickness; likewise locating the center of gravity too far forward in an attempt to make the longitudinal stability extremely positive is detrimental to the full efficiency of the ship.

There is no good reason that the writer can see to try to balance the ship so that it will recover rapidly from a power-off dive after having been balanced at cruising speed, if it means that in normal flight sufficient negative setting must be carried on the stabilizer to cause a sacrifice of maximum performance.

Maneuverability is also a safety factor in tight places, and a careful balance between this feature and stability must be arrived at. This brings up a question that is, or has been, before every manufacturer; namely, the attitude of the individual engineering inspector of the Department of Commerce. As the author brings out, the department's instructions relative to the foregoing points are indefinite. The department unquestionably deserves the appreciation and thanks of all in the aviation industry for the splendid work that has already been done toward making aviation safe. However, the writer feels that the department could help matters still more by impressing upon the individual inspectors the necessity of approaching the subject of stability very conservatively. Because a sufficient amount cures, it does not follow that more will cure faster.

AUTHOR'S CLOSURE

It is easily possible in some types of airplanes to so design it that it will balance at the same air speed for any throttle setting. In other designs this is quite difficult, and for rapid movements of the throttle is impossible without going through a series of oscillations before reaching steady flight. It is of course desirable. This was discussed in the paragraph on power on and off balance which stated, "This can best be given in terms of air speed, throttle closed and wide open, with stabilizer set for cruising and no pressure on the control column." Mr. Gassner's other comment on the behavior of the ship in a stall is well taken. Attention is invited to the fact that an airplane goes into a straight dive when all controls are released only when stalled in straight flight. Since, except during demonstrations, airplanes are most often stalled while turning, this feature is not as important as it may at first appear.

The author did not intend to make a sharp distinction between stability with free controls and locked controls, because, since no airplanes are equipped with devices to lock controls, why should one be interested in the stability in this condition. If the pilot attempts to hold controls in neutral for stability, it is not only an unnatural procedure, but he has one chance of being right and an infinite number of chances of being wrong.

It was the purpose in writing the paper to assist the test pilot and designer in understanding each other and not in assisting the designer to correct faults found by the test pilot; the latter is purely a design problem in which the test pilot is usually of little assistance. The remainder of Mr. Miller's discussion, therefore, being beside the point, will not be answered.

In order to answer Mr. Budwig's discussion it will be necessary to repeat and expound some of the material given in the original article and to comment upon Inspection Service Memorandum No. 1.

In explaining the normal manner in which a certain airplane was to be flown, the author stated that the rudder must be pushed hard in one direction; by hard was meant to put both feet on the rudder pedal to try to bend it. It was impossible to push the pedal more than 2 inches in either direction without waiting for

response; while the airplane is making a turn it is impossible with both feet to even get the rudder pedals into neutral, much less the rudder. This point was brought out, not as a measure of lateral stability, but to show a lack of coordination in control forces. The aileron control could be moved to any position with one finger only, with absolutely no exertion.

Mr. Budwig stated in his discussion that "During the directional stability test the wings are held level by means of the ailerons and during the lateral stability test the airplane is held level by means of the rudder." Had this been stated in Memorandum No. 1, the author's statement that "Department of Commerce instructions are very indefinite and in some cases even incorrect," would not have been made so broad. As requested by Mr. Budwig, the author will point out specifically parts of the memorandum that are indefinite or incorrect.

Their instructions to determine longitudinal stability read as follows: "Whether or not an airplane is longitudinally stable at any given load or power condition can be determined by balancing the airplane with the stabilizer for level flight and then nose the airplane over approximately 15 deg. and release the control column. The airplane on being pointed down should increase its speed and the nose begin to come up." This is a very good test for dynamic longitudinal stability, but will not assist the designer in determining statical longitudinal stability, although this memorandum does not distinguish the two. According to these instructions, the test is started while the airplane is rotating about the axis of pitch, is at the moment 15 deg. to the horizontal, and the air speed is slightly above cruising speed, but actually unknown, and the angle of incidence is unknown. How far the nose continues to go down will depend upon the moment of inertia about the axis of pitch and the stable righting moment. The moment of inertia being so much larger, it is impossible to tell whether the aerodynamic moment is positive or negative.

The test for lateral stability reads as follows: "Lateral stability is determined by setting the controls for normal flight and then by purposely dropping either wing. If either wing tends to return to horizontal with the controls released, the airplane can be considered laterally stable. If, however, either wing stays wherever you put it, the airplane is neutrally stable. If the condition becomes worse and either wing keeps coming up or down, the airplane is laterally unstable. All airplanes must be at least neutrally stable laterally." Contrary to Mr. Budwig's statement, lateral stability test is made with controls released, if made according to instructions. If the top rudder were used to bring the airplane level, the test would be correct.

The next paragraph in the memorandum, that on directional stability, shows a similar fault.

Mr. Budwig's statement that "In fact, an airplane with sufficient stability would usually dive out of a turn and assume level flight attitude as the controls are released," is not entirely correct, since, if an airplane is going to recover from a turn, it will lose speed instead of diving before assuming level flight.

Mr. Budwig has somewhat confused the author's statements on stability and power on and power off balance. The "negative lift on the stabilizer, which builds up rapidly as the speed increases," as quoted from Mr. Budwig's discussion, may be built up from the slipstream as well as from increased speed.

It is quite common for pilots to say an airplane is stable when they mean its flight is stable. The more stable an airplane is, particularly in roll and yaw, the more unstable will be its flight. The necessity to fight controls in rough, gusty air is due to excess of directional and lateral stability and not to an insufficient amount.

AER-53-11

Opportunities for Aviation Training

By JOSEPH S. MARRIOTT,[1] WASHINGTON, D. C.

In this paper the author endeavors to set forth the possible methods of obtaining adequate instruction and training to enter the aviation industry either as a pilot or in a ground capacity. The article is written from the practical viewpoint of training facilities available in the industry itself rather than opportunities presented by strictly educational institutions such as secondary schools and universities. The author also points out some of the advantages and disadvantages that are apparent in the existing means of obtaining aviation training.



THOSE who have kept in close contact with aviation in the past three years realize that it has emerged from a babe in arms, passed through the creeping stage, and is beginning to take an important place in national industrial development. The continued expansion and development of this new industry can be accepted without question. Along with this progress and expansion is found a variety of opportunities likewise increasing for those who are adequately trained and qualified to find employment and fill responsible positions. While it is possible for some of these positions to be filled by men from other allied industries, a majority of the opportunities are such that specialized training or a background of experience is required. Since men who have had any previous experience are practically all connected with the industry in some capacity, the operators are faced with the necessity of calling on those who have been trained, since the expansion of the past few years has made the need for additional trained personnel evident. Therefore, an important part of the industry itself is the training facilities which are available for preparing men and women to take their places in the responsible positions of the aeronautical field.

Since it is the purpose of this paper to discuss the opportunities for training which exist as a part of the industry itself, only passing mention will be made of the facilities provided by strictly educational institutions such as secondary schools and universities. It is, however, worth while to mention that probably one of the best ways of preparing for a life work in aviation is by the route of an aeronautical engineering course. Courses of this nature are offered by a large number of our leading universities, considerable impetus having been given to the institution of such courses by the Daniel Guggenheim Foundation. In addition to the university courses the report of the survey made by the Daniel Guggenheim Fund Committee on Elementary and Secondary Aeronautical Education shows that an ever-increasing number of secondary schools are teaching aeronautics in some phase. True enough, a number of these courses are of general information value only, but a small group of polytechnic and trade schools are offering courses of such a nature that they could be classed as vocational education in preparation for ground positions.

For a first rough classification of opportunities existing in aviation the types of positions may be divided into: (1) Operations requiring flying ability and experience, and (2) operations of a ground nature. In the first group are commercial pilots operating aircraft on scheduled mail runs; passenger and express airlines; aerial taxi service; passenger hopping and miscellaneous operations; governmental pilots functioning as inspectors for the purpose of examining and testing pilots for license; aircraft salesmen demonstrating planes for sale; instructors in flight-training schools; test pilots testing the products of aircraft factories; specialized pilots operating some specific industrial mission such as aerial photography, mapping, crop dusting, aerial advertising, etc.

The second group includes the engine mechanic, airplane mechanic, maintenance superintendent, ground-school instructor, airport manager, traffic agent, airport engineer, aeronautical engineer, research engineer, airways mechanicians, airways keepers, weather observers, radio operators, parachute riggers, instrument repair men, airport lighting specialists, and a host of others requiring some technical or specialized training.

In addition to the above two groups, a third group might be added which would include the personnel engaged in the business phases which closely interlock with the activities of the ground and flight operation personnel. In this phase are minor executives and managers of such activities as aerial photography and surveying, aircraft sales and distribution, aerial advertising and promotion, airport development and management, accessories specialists, insurance experts, factory and office executives. It is evident that this third group is being supplied to the industry mostly from other allied industries and the business world. Therefore, no particular study has been made of the opportunities available for training in these fields because of the fact that the average individual entering such a field will have had previous business, sales, or engineering experience fitting him for the position involved, and he will find it entirely possible on his own initiative to acquire the general aeronautical information necessary for the executive duties involved in that type of position.

It will be seen readily from the preceding classification that the different positions in the field of aeronautics demand various types of training and experience. However, education in certain basic lines will lay the foundation upon which the individual

---

[1] Chief, Inspection Service, Aeronautics Branch, Department of Commerce. Mr. Marriott graduated from Leland Stanford, Jr., University in June, 1917, and was commissioned First Lieutenant, Aviation Section, Signal Corps, in December, 1917. He was stationed at Park Field during the war, and at the conclusion had passed through the stages of dual instructor, flight commander, officer in charge of second solo, and officer in charge of flying. Entered commercial aviation as manager of the Memphis Aerial Company and two years later was made head of the science department of the Tamalpais Union High School, Mill Valley, Cal. He remained there five years, during the last two of which he was vice-principal in addition to his other duties.

Mr. Marriott joined the Aeronautics Branch, Department of Commerce, in 1928. He was an inspector in and later supervisor of the New York district. In 1929 he went to Washington to take charge of the school inspection and regulation work as an assistant chief of the inspection section, and was advanced to Chief, Inspection Service, in October of that year.

Contributed by the Committee on Education and Training for the Industries and presented at the Annual Meeting, New York N. Y., Dec. 1 to 5, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

may acquire specialized information leading to other and higher positions provided the person possesses the proper qualifications and necessary basic training. The desirability of adequate preparation cannot be overemphasized.

Opportunities for Flight Training

In view of the fact that the entire first group requires ability and experience as a pilot, investigation should be made of the various avenues through which a person desiring flying training as a preparation for aeronautical work may acquire this instruction and training. The different means of acquiring flying training may be subdivided into: (1) Military schools, (2) commercial flying schools, (3) individual ownership and purchase of instruction, and (4) flying clubs.

Military Schools

The army, navy, and marine corps operate primary and advanced flying training schools for the purpose of providing officers and flying personnel for the particular branch of the service involved. The fact remains that these services do not require the permanent employment of all the graduates of their schools and, although each branch of the service requires that the graduates of their schools undergo a period of active duty after instruction is completed, a large number of the men so trained do eventually leave the service and enter commercial aviation. In view of the fact that the navy, marine corps, and army select the applicants for these schools with a view to their fitness as officer personnel as well as their fitness to become pilots, the educational requirements are somewhat stricter than are required in commercial aviation. These requirements are such for the navy and marine corps that only applicants with the equivalent of four years of college training stand an appreciable chance of being accepted. The army will accept applicants who have satisfactorily completed at least two years of college or who are able to pass an examination which is equivalent thereto. For all branches of the service, there is a minimum and a maximum age requirement and strict physical qualifications. Providing an applicant is able to secure an appointment to one of these military schools, there is, of course, no question regarding the quality of the training which he will obtain. However, it is only natural that this training will have the emphasis of the military phase of aviation, and the graduate of such school, while he will have had adequate flying training, will, in the beginning, be somewhat handicapped in commercial aviation due to his lack of knowledge and familiarity concerning the problems relative to commercial operation.

Commercial Flying Schools

It is believed that the commercial schools which have been issued approved-school certificates by the Department of Commerce provide the most complete and adequate training available in the industry itself for preparation as a pilot. The air-commerce act of 1926 was amended February 28, 1929, by the passage of the Bingham bill authorizing the Secretary of Commerce to inspect and rate civilian schools giving instruction in flying when such schools applied for approval.

Under the provisions of this amendment school regulations were formulated and inspection and rating of commercial schools were under way in June of that year. At the present time 43 schools are operating under approved-school certificates issued by the Department of Commerce. The results obtained by these approved schools have been thoroughly gratifying. The improved methods of instruction are decidedly apparent in the greater capability of the graduates whom they are presenting as applicants for Department of Commerce pilot's license. This can best be shown by the figures obtained from the Department of Commerce records regarding the examination of pilots for license. A comparison of the number of student permits issued, which are not over a year old, with the number of the holders of such student permits who have applied for higher grades of license indicates that 60.6 per cent of those issued student permits continued their instruction to the point where they applied for some grade of pilot's license. It is believed that this can be accepted as a fair figure for comparison because of the fact that the department's study shows that less than 5 per cent of those who hold a student permit for over a year ever apply for a higher grade of license. In comparison, or rather in contrast with this figure of 60.6 per cent, it is found that 91.6 per cent of the students enrolled in approved schools actually complete the course and obtain a higher grade of license. The remaining 8.4 per cent includes not only those who discontinued their instruction voluntarily or were dropped from enrolment by the school, but also those who completed their course, but were subsequently disapproved for license. This comparison shows conclusively that the methods of instruction as well as the attendant conditions at an approved school decidedly decrease the factors which influence an individual to discontinue his training before completion.

Further proof of the value of approved-school instruction methods is obtained by comparing the figures of all applicants for pilot's license. For the first half of the year 1930, Department of Commerce figures disclose that 20 per cent of all applicants for pilot's license were disapproved. During the same period, the figures of approved-school operation show that only 7.34 per cent of the applicants for license from these schools failed to pass the test for license, which means that 32.66 per cent of those applying from other than approved schools failed to obtain license. Here again, the percentage is greatly in favor of the approved-school instruction. This comparison is even more favorable to the approved school when the fact is taken into consideration that a larger number of the approved-school graduates are applying for the higher grades of license, that is, industrial, limited commercial, and transport, than is the case of applicants on the outside. By this is meant that a large number of those receiving instruction from other than approved schools acquire only enough instruction to obtain a private grade of license, the test for which is less difficult than the examination for the advanced grades. The present number of approved schools and their locational distribution throughout the United States make it possible for any one desiring to learn to fly to enroll at such a school with a minimum of inconvenience and travel expense. As an explanation of the reason for the results which it has been possible for the approved schools to obtain, it should be borne in mind that approved schools are required to give the student a complete course in ground instruction in addition to the flying training. Also, the student is required to have instruction on a number of different types of aircraft, which increases his capability and broadens his experience prior to the time when he actually enters the industry. The thorough instruction which the approved school is required to give the flying student has naturally increased the operating cost of the school. Certain advantages accrue to the approved school such as advertising the fact that they have governmental approval and the counting of a portion of the dual instruction flying time as solo flying experience. However, the approved schools are required to meet the serious competition of the unapproved school which can operate without providing a ground school or a variety of equipment.

As stated in the foregoing section, it is not compulsory for a school to obtain a Department of Commerce approved-school certificate in order to operate. The matter of applying for a

rating is entirely optional with the operator. Because of the necessity of meeting certain definite requirements such as providing a field of adequate size, several different types of aircraft, and maintaining a ground school as well as a flying school, a number of operators have continued to conduct their schools on an unapproved basis. The quality of instruction being offered by such schools may be of a high standing. However, there is no assurance that such is the case, and a careful investigation of the facilities, equipment, personnel, and reputation of the institution should be made before a person decides to enroll. After enrolment, the student will find that, under Department of Commerce regulations, only his solo-flying experience counts toward eligibility to apply for a pilot's license. Consequently, in all probability, he will receive only a limited amount of instruction in accordance with whatever restrictions the contract with the school calls for, and his solo-flying experience must be built upon his own initiative without adequate check time and advanced instruction. In other words, any check time or advanced instruction will have to be paid for in addition to the solo time, which very often results in the advanced instruction not being acquired at all. The persons involved, therefore, usually apply for license without adequate instruction and with the possibility of having acquired faulty flying habits which, even though they may not be dangerous, will be expensive to eradicate and will certainly affect the pilot's competency and his ability to obtain a worthwhile position in the industry. He also runs the chance of being in the 32.66 per cent group with those who are disapproved for license on the first examination and who, on that account, are required to wait 90 days before being eligible to reapply. This causes considerable embarrassment, loss of time, and expense for additional instruction. While it is possible for a student flier with the proper aptitude for assimilating flying instruction to obtain his training and experience in an unapproved school, he may, in the long run, find the course more expensive than the approved course would have been, which course carries with it a practical guarantee of being able to obtain the pilot's license.

Individual Ownership

When an individual is so financially situated that he can purchase his own airplane and hire a pilot to teach him to fly, or can reverse the process, that is, hire a pilot to give him instruction and, at the completion of the course, purchase an airplane to build up his flying experience, he is in a position to acquire thoroughly satisfactory training if he is sure of the competency of his instructor. The trend of aviation appears to be such that in the future a number of sportsmen pilots and individuals who purchase airplanes for their own private use will receive their preliminary training leading to a private pilot's license from the factory or sales agency from which they purchase the airplane. Teaching the purchaser to fly will be a part of the purchase price of the airplane. If the individual who desires to fly only for his own pleasure or to use an airplane as a means of his own personal transportation learns to fly by this method, the result will be that the established schools, both approved and unapproved, will be training centers for professional pilots only. In this case, the tendency would be for the schools to lengthen their courses beyond the present requirements for transport grade of license in order that they might provide specialized phases of instruction preparing pilots for air mail, passenger, and airline operation and other types of pilots' positions. In fact, one or two of the approved schools have already offered advanced courses of this nature. One school in particular offers a 500-hr. course for pilots for definite preparation for scheduled airline operation. This is in contrast to the customary 200-hr. course for transport license, which is the highest one the average approved school is now giving. The group of individuals who are able to purchase airplanes and employ pilots to give them private instruction is, at the present time, relatively small. Also, a relatively small number of this group are interested in entering commercial aviation in a flying capacity. This group is, therefore, not a particularly important factor in the nation's aviation training program.

Flying Clubs

The flying club offers an excellent opportunity for a person to build up his flying experience and keep his hand in after having acquired his preliminary instruction. Particularly is this the case if a person is not able to purchase a plane as an individual or is unable to obtain a position requiring flying duty immediately after obtaining his primary instruction. The flying club also offers an opportunity for an individual to obtain his elementary flying instruction if conditions are favorable and the club is organized properly with a membership including men who have sufficient experience and qualifications to serve as instructors, or providing the club is in such financial position that it can employ the services of a competent instructor. However, it should be pointed out that the limited amount of equipment and the usual lack of facilities for maintaining and servicing the club airplane usually make the matter of learning to fly by this method a long drawn out process. Also, when the flying instruction is so spread out, the individual finds difficulty in maintaining progress and, in most instances, takes a greater number of hours in the air to acquire satisfactory skill in manipulating the airplane. This may cause discouragement to the student flier and influence him to give up the experiment before he has reached the place where he can obtain a license and become a producing part of the flying fraternity.

The foregoing four methods, as outlined, now constitute the principal opportunities for flying training existent in the country today. The selection of the method of training should be made with the advantages and disadvantages of all the four available methods in mind, particularly as they apply to the individual himself.

While there has been some talk in the past six months to the effect that there is now an overproduction of pilots in the industry, a careful analysis of the situation discloses that a very large percentage of the pilots now out of employment are those whose capabilities and experience as pilots do not meet the standards set by the industry itself or they are lacking in certain other qualifications which render them unsuitable for a given phase of work. Unquestionably, with the present healthy growth of commercial aviation operations such as the extension of scheduled passenger-carrying airlines, the pilot who has been adequately trained and who has the proper qualifications will be able to find a position commensurate with his capabilities. In this connection, it might be well to add that the industry does not look upon the graduate of a school, even though that school be approved, as a comp'etely trained product. Ability to fly aircraft under varying conditions of weather and other emergencies can only be acquired over a period of time and cannot be crammed into the school course, regardless of how intensive the course may be. In addition, any specialized type of work, such as the variety of type of positions specified in group one, will require additional preparation if the individual does not have knowledge of same prior to taking the pilot's training. At the present time, very little has been done toward providing these specialized courses in commercial aviation schools. Some courses are available for aircraft salesmen, aerial photographers, and flying instructors, but the demand for such instruction has not been great enough to warrant the institution of many courses covering these fields.

Opportunities for Ground Training

Opportunities for training of a ground nature in connection with aircraft operation as they now exist are: (1) Trade and technical schools, (2) practical experience as apprentices in the industry, and (3) combination of aeronautical courses and practical experience.

A representative group of the trade and technical schools is offering courses in preparation for ground positions in aviation. However, the nature and the value of the instruction are as divergent as the number of schools. This is due to the fact that there are no established standards for any of the courses of this type, which means that the kind and quality of instruction depends entirely upon the initiative, capability, and viewpoint of the instructors and operators of these schools. A greater portion of the trade and technical schools attempt to prepare a person for a position in the mechanical phase of aviation ground operation. This school preparation, if adequate in any sense, prepares an individual to take a position as an apprentice mechanic working on the maintenance and repair of aircraft under the supervision of licensed personnel. After having acquired sufficient practical experience, the apprentice mechanic is eligible to apply for a Department of Commerce mechanic's license, of which there are two types, the airplane mechanic's license and the engine mechanic's license. In addition to the trade and technical schools, quite a number of the commercial schools, especially the approved ones, are offering courses for mechanics. There is no provision at the present time for the rating of such courses by the Department of Commerce. Consequently, little can be said about the quality of the instruction being offered or the uniformity of instruction as given by the different schools.

The army air corps conducts schools for mechanics for its enlisted personnel. This army course is a combination of theoretical and practical training. Together with the subsequent experience obtained during the three-year enlistment period, it produces thoroughly competent mechanics provided they have the necessary aptitude and qualifications to begin with. The army air corps also conducts at Chanute Field a school for parachute riggers. To date, the graduates of this school have furnished practically all of the army as well as commercial parachute packers. In view of the recent Department of Commerce parachute regulations which require a parachute packer to hold a Department of Commerce parachute rigger's license if he is to pack parachutes used in commercial aviation, this type of position will become increasingly important in civilian aeronautics.

New York University has done some valuable work in developing a course for ground-school instructors. This activity has been conducted under the supervision of the Daniel Guggenheim Committee on Secondary Education. So far as is known, this is the only school offering a course preparing ground instructors for positions in approved schools.

Practical Experience as Apprentices in the Industry

By far the greatest number of men serving in a ground capacity in connection with the operation of aircraft have obtained the position which they now hold as a result of previous practical experience. These men, particularly the airplane and engine mechanics, served with the air service during the war or with the barnstorming pilot since the war and have developed their knowledge of airplane maintenance and repair from the "University of Hard Knocks." The work of an aviation mechanic is rapidly developing into a trade of its own. The men without previous experience or with a mechanical experience from other fields are, in many cases, entering aviation groundwork by the apprenticeship method. If the individual is fortunate enough to obtain a position where he can, by applying himself, learn the various phases of work in connection with the mechanical repair of aircraft, it is possible for him to acquire sufficient knowledge to enable him to obtain a mechanic's license, and with the proper qualifications, to progress to a responsible position supervising this type of work. In many instances, however, the apprentice finds himself working under a mechanic who obtained his knowledge and experience in the early days of the industry and who, although thoroughly capable himself, is all too often unwilling or unable to impart his knowledge to the apprentice. Factories are hiring from time to time inexperienced employees and training them on some portion of the aircraft construction work. These men are not required to hold a license because they are not working on the maintenance and repair of aircraft, but rather on the construction of a component part under the supervision of competent engineers. Such men, while they have an opportunity to work into factory production, will have some difficulty in getting into aviation operation because their training has been entirely limited to one phase of airplane construction. As the factory production increases and as commercial operation expands, increased opportunities for men to work as apprentices and learn their trade will become available.

Combination of Aeronautical Courses and Practical Experience

The aviation ground man who obtains his training solely by practical experience will be limited in the advancement which he can make in the industry. This is true, because he will have learned to do certain things, but will not have learned the why and the wherefore. He will have no theoretical or technical background, and will have to rely entirely upon his experience of having done a job before or having seen it done in order to be able to do it himself. Therefore, probably the best method of training for aviation ground positions is a combination of some aeronautical course which will provide the theoretical instruction regarding the position which the person desires to fill combined with the practical experience in work of this nature. Whether this aeronautical course is taken before or during the time when practical experience is being acquired is not necessarily essential, although if it can be taken in conjunction, the theoretical training will be more easily assimilated by the student. If the person is to follow this method, it will probably be necessary for him to attend night courses or take correspondence courses in order to enable him to work and obtain his practical experience during the day. The selection of a ground course should be made only after a careful investigation of the reputation of the school and the facilities available. For courses of a mechanical nature, the amount of modern demonstration equipment provided for laboratory or maintenance and repair instruction is an important factor. The value of a correspondence course is dependent, to a great extent, upon the application of the person taking such a course and his ability to apply practically, in his daily work, the principles which he is studying. Earlier in this article, it was stated that there was a lack of uniformity in ground courses and that no standards existed for relative comparison of such courses. There has been a standard set up by one governmental agency for courses which are designed for the instruction of airplane and engine mechanics. The Federal Board of Vocational Education has issued a bulletin setting forth the phases of instruction essential for each of these two courses. The bulletin gives the duties of the airplane and the engine mechanic, and includes in the course outline a complete set of typical jobs for practical instruction. It is believed that the combination of theoretical and practical experience is the soundest means of preparation for any ground aviation position.

The necessity for some theoretical and technical instruction is conclusively indicated in the high percentage of failures by applicants for mechanic's license who apply with only a background of practical experience.

New Ground Positions Developing

The provision for the licensing of parachute riggers, which has already been mentioned, is an example of the new types of specialized positions which are developing in the ground operation of aviation. Another example is the necessity for competent instrument repairmen on operating airlines. Still another is the development of radio in connection with aircraft and the necessity for radio operators to handle such equipment on the airplane.

The radio division of the Department of Commerce has established a new grade of radio operator's license known as commercial third class which is specifically designed to take care of men operating radio-telephone or radio-telegraph installations on aircraft. This license or a radio operator's license of a higher grade will entitle the holder to serve as a licensed airman in connection with licensed aircraft.

In closing, the fact should be pointed out that the acquisition of proper training for filling a definite ground position in connection with aviation depends, to a large extent, upon the ability of the individual to make the most of the facilities which can be obtained. His success or failure will be governed by his initiative and persistence in following through to a conclusion the course of instruction which he maps out for himself. In addition to the theoretical training, the practical experience is equally as important. The selection of the place to obtain this initial experience should be made with consideration given to the opportunities which are available for progress rather than the amount of salary which is involved. While it may appear that the opportunities for aviation training for ground positions are limited, it is, nevertheless, possible for any one with the proper qualifications to equip himself adequately for a successful future in this work. Information relative to the names of schools offering aeronautical courses, either for ground or flight instruction, can be obtained from Department of Commerce bulletins. The department has recently prepared a new bulletin on the entire subject of aviation training which will be of value to any one interested in acquiring such training.

Discussion

T. A. Sims.[2] After reading this paper there are several thoughts which may or may not be of any value. The writer knows little about commercial aviation beyond the average military pilot's knowledge of the commercial game gleaned from several years' experience in flying around the country visiting civilian fields as well as military. However, the writer formerly was stationed at the Air Corps Advanced Flying School at Kelly Field, Texas, as a flying instructor and is familiar with the present system of flying training in use by the Army Air Corps. It is this system which he would like to recommend to the industry.

An enormous expansion of commercial aviation in this country is being held up simply because the public is still skeptical, and rightly so, about the safety of flying. Although the large air lines have a very low accident rate, the general public forms its opinions of aviation in its entirety, and every accident exerts its disquieting influence on the whole industry. So it behooves the whole industry to take drastic steps to demand safety in flight; to convince even the insurance companies by cold statistics that it is safe to fly. Now every one knows that approximately 75 per cent of the accidents are caused by errors in pilotage. So let us begin at the pilot factory and nip these errors in the bud.

The standards of commercial schools at present are far below what they should be for the best interest of commercial aeronautics in that they have no tendency to eliminate the inferior student. Any man who pays his tuition fee for a 200-hour flying course, for instance, can complete the entire course regardless of his progress in flying ability, which may or may not be satisfactory. The schools should be honest enough to eliminate the student when he has reached his ultimate limit of learning, beyond which further instruction is useless. To graduate such a student is a crime against the aviation industry, for this man is handicapped from the beginning, and lacking the normal safety factor he stands more chance of becoming a casualty.

Imagine the condition at West Point if we had no method whereby the deficient cadet could be eliminated. About fifty plebes per year would just be wasting their time when carried to more advanced classes, because the subject matter would be so far over their heads that they would be completely lost. This is the condition that exists in the commercial flying schools today. Instead of turning out a smaller number of safe, dependable pilots, they have entered mass production of a larger number of pilots of the questionable variety. Compare this with the Government flying schools, which perhaps swing the pendulum the other way and graduate about 25 per cent of a class, eliminating any one the instant he fails to show satisfactory progress and becomes dangerous to himself and others. It is only fair to the student that he be eliminated when he reaches this stage. Unfortunately, some of these eliminated students will not believe that they are incapable of further flying, and will enter the commercial game. Handicapped as they are by the "lack of inherent flying ability," they will eventually become a casualty. They should be forever prevented from touching the controls of an airplane in flight.

Now the writer realizes that there is at present a vast difference between commercial and military flying. As he understands it, the military pilot must be able to use his airplane as a military weapon; that is, the actual technique of flying to him is secondary to the proper use of the military weapons at his disposal. He must possess "inherent flying ability" or, in plain language, the ability to operate the airplane automatically by his subconscious mind while his main mental effort is directed toward his enemy, his machine guns, bombs, camera, radio, his tactics and strategy, and countless other tasks. On the other hand, it would seem at first glance that the commercial pilot needs very little of this "inherent flying ability" because he can devote most of his mental effort to the actual technique of operating the plane. However, it is this "inherent flying ability" that the writer believes should be a requisite of the commercial flier as well as the military. What may happen to the pilot without it? He meets an emergency sooner or later wherein he is required to look for a landing field or make his choice between several courses of action, or in other words, to use his mind for something other than concentration on the technique of operation. He does one or two things, both wrong; he either neglects the technique and allows the plane to fly itself, which, together with his natural tenseness on the controls, will cause the plane to fall off into a spin, or he neglects the headwork or judgment end of his forced landing, and the chances are that he crashes. The possession of "inherent flying ability" is an enormous safety factor, and the industry will do well to follow the lead of the military schools and demand it in their pilots.

Now some people might question this and bring out the fact that the military accident rate is as high as the commercial. This is true, and the reason for this lies in the type of flying that the military services are required to perform and which to be of any tactical value whatsoever are far more difficult and danger-

[2] Second Lieutenant, Air Corps, U. S. A., West Point, N. Y.

ous than most commercial missions. Were the military services required to reduce their accident rate to the minimum, it would be necessary to dispense with formation flying, for instance, but at the expense of the effectiveness of the Air Corps as a military arm.

Another thing, the writer thinks the 200-hour requirement of the Department of Commerce is really no fitting criterion for a transport pilot's license. A student's knowledge depends not upon the amount of time spent in study, but upon what he gets out of the study, on the quality of time in the air rather than quantity. Quoting from the paper that "Graduation from a 200-hour course in an approved school carries with it a practical guarantee of a license," the writer has seen some schools where the students were merely counting flying hours in building up to that coveted 200; they were doing the same thing day after day (acrobatics) without much thought toward improvement of flying ability. The writer has seen one school where the bulk of the students were airplane factory workers, who would save out their $5 from their weekly pay roll and spend it for 15 minutes of flying instruction, a slow and tedious course, without any hope of sequence or any instruction in a phase of flying that took more than 15 minutes to complete. The knowledge of either of these two types cannot be compared to that gained by a student who completes his course of supervised instruction within a year's time, who has flown in various types of airplanes, made cross-country flights, or perhaps flown as co-pilot on transport lines. The writer thinks that a stiff flight check by the Department of Commerce covering everything an embryo transport pilot should know is a far better criterion of flying ability, and too much should not be taken for granted upon presentation of 200-hour credentials.

The third opportunity for aviation training, that of individual ownership, is relatively small due to the large financial outlay it takes at present to purchase, maintain, and operate aircraft. However, with the advent of standardization and mass production this cost will diminish in the future, and this group will become more important. To insure safety it will be necessary to rule this group with better regulations than are in existence today. This group is controlled mainly by the various state laws governing flying, just as automobile licenses are controlled today. These laws lack uniformity; in fact, some states have no laws, and it is possible in these states for a man to purchase an airplane, get in, and fly it away with little or no flying experience, when the same man would be arrested for driving his car without a license. It appears that the only solution rests in more stringent and uniform state regulations. So much for pilot training.

Obviously, the graduate of any school is not a finished product. Under present conditions the only way a pilot can learn anything more is through the "school of hard knocks," his own experience over a period of time. The pilot has several real forced landings, and finally learns that he can "walk away from them" if he flies at a safe altitude, so that he has plenty of time to think and plenty of room in which to maneuver. Finally, flying at a safe altitude becomes a habit. However, these lessons by personal experience are sometimes fatal, if the pilot makes the wrong decision when he gets into some predicament that he has never experienced. In fact, the first year after graduation is the most dangerous one of the pilot's career. The various statistics on aircraft accidents emanating from the Chief of the Air Corps and the Department of Commerce are extremely valuable to a person who understands statistics and who studies them thoroughly. By these means we learn from the experience of others. However, it is the writer's opinion that there is a considerable amount of information on aircraft accidents that is going to waste because it cannot be expressed by statistics and graphs. By this is meant all the various details of an accident, the conditions under which it happened, what was done by the pilot, what happened, and a statement by an expert on what the pilot might have done to avoid the accident. Such information never gets into print and is consequently lost. The Navy gets considerable information on landing accidents by having a moving-picture camera trained on the approaching airplane, and taking the photograph when an accident looks possible. Obviously this is impractical for the mass of accidents, but the essential information could be reproduced by drawings and pictures. This information might be broadened to include personal experiences of reputable men. To be specific, suppose a pilot in a land had a forced landing, with the choice of landing either in the woods or in a river. Certainly the great collection of data on aircraft accidents could give some indication of the best line of action in this case. Also certainly some experienced pilot could be prevailed upon to explain the technique of landing a land plane on water. In the military services a common mistake of the inexperienced pilot having motor trouble while flying in formation is to pull up and out of the formation in a natural reaction to gain more altitude. He flies directly across the path of the succeeding elements in the formation, and sometimes a collision results. This happens several times each year. If these collisions were represented pictorially and repeatedly published to the services every time such an accident happens, it would soon teach every one to dive out of a formation instead of pulling up. Some Government agency should be prevailed upon to publish this detailed information as an addendum to the statistics.

Perhaps quoting a few statistics will tend to verify some of these contentions. In the paper, "Opportunities for Aviation Training," it is noticed that 7.34 per cent of the applicants for license from accredited schools and 32.66 per cent of the applicants from other than approved schools fail to obtain their license, whereas the writer knows not a single case of a graduate of the Army flying schools failing to obtain a transport pilot's license.

To summarize, it has been attempted to show that:

1 The standards of commercial schools at present are far below what they should be for the best intent of commercial aeronautics
2 The 200-hour requirement of the Department of Commerce is really no fitting requirement for a transport pilot's license
3 Better state laws are needed to cover cases not controlled by Federal Interstate Commerce regulations
4 Some Government agency ought to publish and distribute detailed information of an educational nature about airplane crashes and the technique of flying.

The foregoing are the writer's personal views.

J. S. A. Johnson.[3] The author has not only described in a comprehensive manner the present opportunities for training in aviation, but also has outlined fully the methods of broadening these opportunities and creating others.

Of the two kinds of training, flight and ground, the former will appeal most strongly to the more adventurous natures. Accordingly, with the gradual increase of aerial transportation an increasing number of regulatory measures must be introduced for safeguarding not only pilots and passengers, but also the general public. As small a number as possible of such means of regulation should be of the "thou shalt not" type, but all of them should be designed to contribute toward the safety, comfort, and convenience of both fliers and those upon the earth. In

[3] Professor of Applied Mechanics and Experimental Engineering, Virginia Polytechnic Institute, Blacksburg, Va. Mem. A.S.M.E.

congested areas aerial traffic regulations therefore will doubtless open up an ever-increasing number of opportunities for employment, along with which will proceed increased opportunities for training in both flying and ground work. The aerial traffic officer of the future must possess not only great courage and skill, but also tact and general resourcefulness. The installation and operation of aerial signs and signals such as beacon lights and possibly buoys, together with effective weather forecasting and indicating equipment, will provide opportunities for numbers of well-trained and skilful pilots and other aerial employees.

Especially significant are the author's statements regarding the necessity of proper general qualifications and the thoroughness of basic training and preparation of the pilot and the flight mechanic. In many cases those who know least concerning the principles and mechanism of the automobile become the most skilful chauffeurs after a few lessons in manipulation and steering. In case of failure to function, the car can at least be stopped until aid is secured. Not so with the airplane, however, when failure immediately to locate and correct trouble may result in disaster. It would appear, therefore, that in time all agencies giving instruction in flying must be chartered, at least by the respective state governments, or come within the classification of the approved schools.

With the increase of aerial transportation owners of airplanes and other aircraft will be required, doubtless, to secure accident insurance for the benefit of passengers and employees, and this would involve a corps of aeronautical experts in various capacities, thereby offering additional opportunities for those trained in both flight and ground schools.

In connection with flight schools, especially those under governmental control, it is probably that consideration should be given to the establishment of Reserve Officers' Training Corps units in aeronautics at certain technical schools, together with aerial summer camps for R.O.T.C., possibly at least one of each in every state, depending upon general conditions, such as population, climate, and topography.

For training in ground work the opportunities would appear to be limitless because of the need of development of materials, fuels, mechanisms, and methods of construction. A broad field in design and research is accordingly opening up for any one with technical training and research ability. As in any industry, there have been many, and there will be many more, impractical suggestions and "inventions," and it will become the duty of those with technical training to pass upon them. For such investigators, especially as the author suggests for those who intend making aviation their life work, the best preparation would be the completion of a course in aeronautical engineering at a standard technical college.

R. Burdette Dale.[4] The author has confined his paper principally to the opportunities which exist within the aviation industry for those who are connected with operation and maintenance. He has given but little consideration to those who are engaged in the planning and design of the airplane structure itself. He did mention the aeronautical engineer whom he has classed in the group whose operations are of a ground nature.

A survey of the industry would probably disclose that not all those employed in the engineering department are aeronautical engineers. The industry rightly makes use of many young men with a basic engineering background who have not had the advantages of a specialized course in aeronautical engineering. This is a customary procedure in the general engineering field. For example, not all technical men in the engineering department of an organization engaged in the manufacture of steam turbines are turbine engineers, but many of them are apprentices or cadet engineers with a mechanical engineering background. These young men deserve special training to enable them to take their proper place in the industry and to fit them to carry on the responsibilities of the future.

The aviation industry is at the forefront of this forward-looking movement. It has observed that those industries which have not been active in recruiting and training promising young men are today suffering for their neglect. This is particularly true in the case of planning activities. Every industry must look forward to the future if it intends to maintain its position with the highest efficiency.

At least one organization in the New York district provides special out-of-hour training for their engineering apprentices. Advancement in the organization is predicated on technical examinations which are held every six months. Their engineering designers and draftsmen are encouraged to attend evening classes in airplane design offered in technical institutions in the city. In selecting the general run of designers and draftsmen, preference is given to the young man with a sound, fundamental engineering training rather than to graduates of aeronautical engineering courses. The industry itself assumes the responsibility of providing the required specialized training either within its own organization or through other agencies.

For the past three years, Pratt Institute has offered an evening course in airplane design. This has been very successful. Most of the students are actually employed in the industry, though a number have prepared themselves to enter the industry by taking the course. The enrolment is limited to twenty members, and these are carefully selected for their suitability and qualifications. Many are college graduates from a general course in engineering or from a specialized course in aeronautical engineering, but membership is not restricted to college graduates. The instructor is a practical and experienced airplane-design engineer who is employed in the industry. The objective is to give intensive and practical but highly technical training which will be of immediate and useful application in the industry.

Roland H. Spaulding.[5] The author has given a splendid paper on the opportunities for aviation training to be found in the industry itself. It may be of interest to know that approximately 500 public schools are giving some form of aeronautical instruction. In about 50 of these the instruction is on the vocational level. The Daniel Guggenheim Fund Committee on Elementary and Secondary Aeronautical Education, with headquarters at New York University, has been aiding the public schools in this instruction for the last two years by its research work and its facilities for the training of teachers, both for ground schools and for public schools. One may obtain the publications of this committee gratis.

The Educational Committee of the Aeronautical Chamber of Commerce of America is also sponsoring the development of educational opportunities in aviation. Last year, in cooperation with the Daniel Guggenheim Fund Committee, it held the First National Conference on Aeronautical Education in St. Louis. The next one will be held in April in Detroit, coincident with the International Aircraft Exposition.

Author's Closure

I wish to take exception to Lieutenant Sims' statement that the standards of commercial schools, at present, are far below what they should be for the best interests of commercial aeronautics in that they have no tendency to eliminate the inferior students. I would call attention to the fact that the approved schools, in order to hold their certificate, are required to maintain

[4] Department of Machine Design, Pratt Institute, Brooklyn, N. Y. Mem. A.S.M.E.

[5] Professor, New York University, New York, N. Y.

a standard that will guarantee that nine out of ten of their graduates qualify for license. This requirement has resulted in the approved schools eliminating inferior students from their enrolment prior to graduation. In fact, most of the schools have adopted the policy of eliminating such students as soon as their unfitness for flight duty becomes apparent. Statistics for the first year of approved-school operation definitely show that the standards of instruction in commercial aviation have been greatly improved.

The Secretary of Commerce is charged with the duty not only of regulating commercial aviation but also of fostering and promoting its development. The approved-school set-up, therefore, was made as nearly ideal as possible without being impractical from a commercial operating standpoint. As a matter of fact, it was made so nearly ideal that, although it improved conditions so far as student training is concerned, the approved schools have found it difficult to maintain their required standards and, at the same time, make a profit.

It is believed that the transport graduate of an approved commercial school is a thoroughly competent pilot. Furthermore, in contradiction to the opinion of the discussor, it is believed that the commercial pilot must possess inherent flying ability and does. He is called upon to fly a wider variety of weights and types of aircraft than the military pilot, under all sorts of weather conditions, and, in the case of air transport pilots, must be able to maintain two-way radio communication with the ground while in flight. Along this same line, Army Air Corps officials freely admit that commercial aviation has done more to develop instrument flying than the military branches.

The author agrees with the critic that too large a proportion of aircraft accidents are caused by errors in pilotage. Therefore, anything that can improve pilot training by having the embryo pilot gain his experience under a variety of operating conditions and thus better qualify him to fit into commercial operation, would be of great value to the aeronautical industry.

Lieutenant Sims points out that the two-hundred-hour requirement of the Department of Commerce is not a fitting requirement for a transport pilot's license. While it is agreed that a certain number of hours does not indicate a pilot's ability, it might be pointed out that the two hundred hours constitute a minimum requirement and all applicants for license are required to undergo a flight test to determine their actual ability. It also might be worth while mentioning that the graduates of military schools have little more than the two hundred hours' total solo flying time upon their graduation.

The author is heartily in accord with the critic's statement that better state laws are needed to cover causes not taken care of by Federal Interstate Commerce regulations. He also is in accord with the suggestion that more publicity of an educational nature concerning airplane crashes and the technique of flying would be of value.

Mr. Dale has pointed out one of the methods whereby the industry may recruit men with a technical background and assist them in obtaining additional technical training needed for the duties involved in connection with a position in the aeronautical industry. His discussion bears out the suggestion that practical experience, supplemented by theoretical instruction, is sound means of advancement.

Prof. J. S. A. Johnson's suggestions for the further development of opportunities in aviation, particularly as they apply to the allied ground opportunities, are excellent. Undoubtedly, as the various phases of the aeronautical industry become better organized, their development will require the services of a large number of men trained in specialized fields.



FIG. 1 AIR VIEW OF LOS ANGELES METROPOLITAN AIRPORT—THE DARK CROSS IS THE SODDED RUNWAY

Management and Operation of a Large Airport

BY WALDO D. WATERMAN,[1] LOS ANGELES, CALIF.

The policies and methods outlined indicate how an airport may be managed so as to take care of all interests. The author covers relations of the airport management with transport companies operating large fleets, with air-service operators having fleets of taxi and charter planes, with schools, with dealers and distributors, with manufacturers, with private owners of airplanes, with individual owners of single airplanes or small fleets for business purposes, and with the general public. Figures are given showing how a successful business has been built.

THE management and operation of an airport presents many interesting problems both serious and difficult. Perhaps the most difficult of these is the maintenance of the proper relationship between the management and those located on the airport.

In an analysis of the situation each individual class of operators must be considered as the problem of each is often greatly different from that of the others. For such an analysis the various operators should be grouped so that each case can be studied individually:

1 Transport companies operating large fleets of planes
2 Air-service operators operating fleets of taxi and charter planes
3 Schools
4 Dealers and distributors
5 Manufacturers
6 Private owners
7 Individual owners of aircraft operating one airplane or small fleets for business purposes, such as oil companies, accessory manufacturers, etc.

Speaking for all classes there are a few fundamental rules which it is believed are necessary in the successful operation of any com-

[1] General Manager, Los Angeles Metropolitan Airport.

Presented at the Meeting of the Aeronautics Division, Seattle, Wash., May, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.



FIG. 2 HANGARS NUMBERS 6, 2, AND 1 AT THE LOS ANGELES AIRPORT



FIG. 3 AIRPLANES IN FRONT OF ADMINISTRATION TOWER AT LOS ANGELES AIRPORT



FIG. 4 HANGARS NUMBERS 1 AND 2 USED FOR TRANSIENT BUSINESS AND BY DEALERS AND DISTRIBUTORS

mercial airport upon which diversified operations are taking place. The first of these is that all commercial activities must of necessity be given every chance to operate at a financial profit or it is certain that a time will eventually come when they cannot pay their bills and the airport will be the financial loser. With this in view it is highly advisable for the airport management to exert considerable effort in publicizing the field and the operators thereon in as many ways as the particular case makes advisable.

It is the policy of many airport organizations to engage in commercial aviation activities to as great an extent as is financially profitable. Many such airports reserve for their own organization passenger-carrying and charter rights, repair facilities, and the marketing of supplies, gasoline, and oil, etc. Although engaging in such activities is unquestionably in many cases lucrative, it is not a function of the airport operator. It is certain that many plane operators would be drawn toward such an airport if they were permitted to participate in such activities. If airplane operators conduct such businesses in competition with the airport operator, the situation would be very undesirable, although such cases are occasionally found. In an ideal set-up the airport management should be in such a position that it cannot favor any particular operator over another, whether he be an airline, a school, a dealer, or service station. Under such circumstances a spirit of good business and clean competition can be built up among the operators, which is the basis upon which true modern American enterprise is developed.

Rules and regulations, in addition to Department of Commerce rules, must be promulgated; but only such rules as are absolutely necessary to the safety of aircraft operation and those that will prevent an injustice to other operators should be in force. The enforcement of airport rules is a difficult problem, as every airport manager experiences minor infractions which under the particular circumstances render no hardship or hazard to any one, yet seriously impair the morale of the pilots. Such penalties as grounding or otherwise punishing such offenders can very quickly cause considerable friction between the pilots and the airport management. A new idea is now being used on the Los Angeles Metropolitan Airport, which has so far solved this rather difficult problem. A "kangaroo court" has been established, having its judge, prosecuting attorney, and clerk selected by the operators appointed by their own group. At periodic meetings the offenders are given a fair trial and, if found guilty, are fined in small amounts. The proceeds of the fines thus derived are used for development or social purposes participated in by all of the operators. This is being found to make the worst offenders an excellent policing body. The system has thus far proved very effective, and any hard feelings that would otherwise be felt toward the management are eliminated.

With such a system there is still nothing to prevent the management from taking any steps which it may see fit along disciplinary lines, but naturally such authority would be unwisely used unless the offense was a serious one and the kangaroo court could not adequately take care of the matter.

It has been found that periodical meetings between the operators and the management are of great benefit. At such meetings many things can be explained and the reasons thereof enlarged upon by the management, thereby preventing cause for dissension. Many constructive ideas can be obtained at such meetings from the operators, and many petty grievances such as exist on all flying fields can be aired and usually amicably settled.

One of the most essential things to make it possible to maintain the proper relationship between the airport management and the operators is to have the original fundamental set-up between the management and the operators of such a nature that certain relationships can be established and maintained. Without this proper foundation there is always room for discord and strife. There may be many set-ups that can be used to advantage, but there are certain things which, if included in the original set-up, make the airport management's future job much less complex. These are briefly outlined as follows:



Fig. 5 Ruckstell Corporation Building Where Service and Repair Work Is Done

The granting of an exclusive privilege of any sort to any operator, even though he pays well for it, is a constant source of trouble. If a certain firm has the exclusive passenger-carrying rights on the airport and another operator is seen by him to load in what is apparently, but not in reality, a load of pay passengers, there immediately arises in his mind the thought that he is being deprived of something which is rightfully his, and nothing will cause trouble and dissatisfaction more quickly than such a feeling. Unjustly the airport management immediately receives the blame and bears the brunt of the ill-will thus incurred. If an airport grants unrestricted commercial privileges to all those basing on the airport, it then only becomes the management's problem to see to it that planes based at other ports do not encroach upon the home plane operators' rights, which is a comparatively easy matter.

The more nearly approaching ownership of hangars and hangar sites that can be developed the simpler the airport management's problems will be, for it is natural for every one to take pride in ownership and look at things from a much different point of view. If an operator owns his own hangar or still better his hangar and the site upon which it stands, together with an interest in the flying area, he feels that he is part of the airport organization and will be more careful to guard and promote its interests. It also encourages him toward a more permanent development and a better standard of operations. He becomes a permanent function of the airport, and, although spasmodically perhaps not so lucrative, is in the long run a much greater asset than a tenant who is here today and gone tomorrow.

One of the most necessary policies in airport management is to make each operator feel that those in charge of the airport take a personal interest in his activities and evidence a desire to be of help in developing his business or of assistance in solving his problems. A much closer relationship can be established along these lines if the airport manager or at least some of his staff are recognized pilots. If such is the case many incidental problems can be dealt with more intelligently and the opinions of the airport management are likely to be more highly respected. Being a pilot, although desirable, is by no means the only qualification that members of airport managerial staffs should possess. They naturally have many problems of construction, engineering, and general business with which to cope, which require a diversified experience.

With an airport built on the foundations outlined the problem of maintaining an amicable relationship between the aircraft operators and the airport management is greatly simplified. The author is frank to admit that with the fundamental set-up which



FIG. 6 FACTORY OF BACH AIRCRAFT COMPANY AT LOS ANGELES AIRPORT



FIG. 7 SERVICE, STORAGE, AND REPAIR DEPOT OF ANGELUS AERO CORPORATION AT THE AIRPORT

exists on many airports throughout this country a harmonious organization would, at many times, be very difficult to maintain. In these cases the fault does not necessarily lie with the management, its personnel, or the operators, but rather with its policies which are basically wrong. Unfortunately many airports have been developed by political administrators with little thought except that of building a monument to their administration or have been developed by private capital with little thought of what the future held in store for them.

To make a drastic change in policy, although apparently for the better, even in an airport's infancy is oftentimes disastrous. This has been tried in the cases of privately owned airports which have been taken over by municipalities. As an example, a certain large midwestern airport which was privately owned permitted all operators to market gasoline. Upon the acquisition of the airport by the city near which it was located, the new management of the airport decided that the revenue derived from the sale of gasoline should go rightfully to the city, and immediately took steps to bring about this change. When the operators were not only deprived of this revenue, but also had to pay a larger price per gallon for the gasoline which they themselves consumed than they did before municipal control, immediate dissension resulted. Although in some cases these operators carried on extended operations with bases at other airports where they were not permitted to market gasoline and were perfectly content with the arrangement there, yet it was one of these operators in particular that raised the biggest protest. The result has been a belligerant attitude between the operators and the municipal management, although in the minds of both they are entirely justified in the stand which they have taken. In the long run it may be found that although considerable revenue can be gained by the city in the sale of the gasoline the loss of good-will more than counteracts this apparent gain.

This is cited just as one case of scores that may arise in a change of policy. The airport that is intelligently built with an eye to the future, instead of merely looking two or three years ahead, will always have fewer operator-managerial problems.

Individual consideration must be given to various classes of operators as outlined in the opening paragraph.

(1) *Transport Companies Operating Large Fleets of Planes.* Such an operator is one of the backbones upon which the industry is built. His demands are many and in most cases do not dovetail well with the activities of other types of operators. One of his principal difficulties is in merchandising. His clients are paying prices for transportation considerably in excess of other means and therefore feel entitled to all possible courtesy and comforts while at the airport. It is therefore highly necessary that the airports from which transport companies operate have suitable terminal facilities available at which various conveniences are to be had. A comfortable waiting room is necessary, with clean and modernly equipped restrooms. Conveniently situated, should be also a clean restaurant where good food is served at reasonable prices, and tobaccos, confections, and current periodicals should be obtainable. Some one should be at hand at all times when airline passengers are present, competent to give full information on the many questions likely to be asked. There is no branch of service that will mean more to the airline operator than judicious attention to all of their clients' needs.

Air-transport hangars should be so located and fenced that the general public has no access to them and so that they are not bothered by the usual airport hangers-on.

Many airports ground all other flying activity upon the arrival or take-off of transport planes. This may be very satisfactory from the transport operators' point of view, but works an unnecessary hardship upon all other operators. When the flying area of the particular airport in question is of sufficient size, the enforcement of the Department of Commerce rules requiring that all ships in the air remain at least 300 feet apart and the granting rights of way to transport ships in the vicinity of airports should be entirely sufficient. Care should be taken, however, that traffic does not interfere with the activities of transport ships on the ground to the extent that they may be delayed in approaching the passenger terminal, and congestion, if any, at that point should be eliminated, as close operation to schedule is highly important to the success of the air-transport operator.

(2) *Air-Service Operators Having Fleets of Taxi and Charter Planes.* Such operators depend largely for their business upon the solicitation of customers from those drawn to the airport through interest in aviation and general flying activity, and of course the repeat business derived from their clientele. Their problem, as of the transport operators, is principally one of merchandizing, and they should be accorded every possible opportunity of contacting those that visit the airport. For this purpose parking areas should be provided at locations giving a good view of the general flying and as free from unpleasantness as possible. It is highly desirable, but oftentimes impractical, from an operation and safety point of view, to permit the spectators to get as close to the operating planes as possible.

On almost every airport it will be found that the spectators are likely to congregate at certain spots, and each operator feels that if he can park his plane nearest that spot he will be advantageously located. As airports grow this will of course be impos-

sible, and it has been found that rather than lose the good-will experienced in the past through a change in policy along such lines it is advisable to require all operators to park their planes in some area immediately in front of or adjacent to the hangar in which their equipment is stored. With this definitely understood in the development of an airport from its start, much future hard feeling can be eliminated.

A public address system operated by the airport company for the benefit of all of its commercial operators or for those who subscribe to its maintenance can do much to build up the trade.

(3) *Schools.* The problem of intensified student training on any airport where transport operations take place, particularly if the port is of limited size, is a serious one. One of the customs frequently practiced during the giving of landing practice is for the instructor to give his student verbal instruction immediately upon landing out in the middle of the field prior to another take-off, this instruction lasting in some cases several minutes. The pilot of a plane in the process of landing often questions whether this standing plane with idling engine is going to take off again or turn to go back to its hangar. This oftentimes confusing situation can be prevented by requiring all ships headed in a downwind direction to keep moving. If they wish to remain stationary, they must head upwind, incidentally giving the pilot vision in the direction of approaching and landing planes.

Certain very definite lanes within the immediate vicinity of airports should be prescribed for the flight operations of student planes. The carrying of tape streamers attached to the rudder by solo students prior to securing a private license easily designates the plane as one piloted by an amateur, and right of way should be given to such planes by all except transport ships.

Hangars in which school operators are based should be easily accessible to the public, and it is oftentimes desirable that they may be thrown open to the public, for many schools obtain their students from casual airport callers.



FIG. 8 AIRPLANES AT NIGHT ILLUMINATED BY FLOOD LIGHT

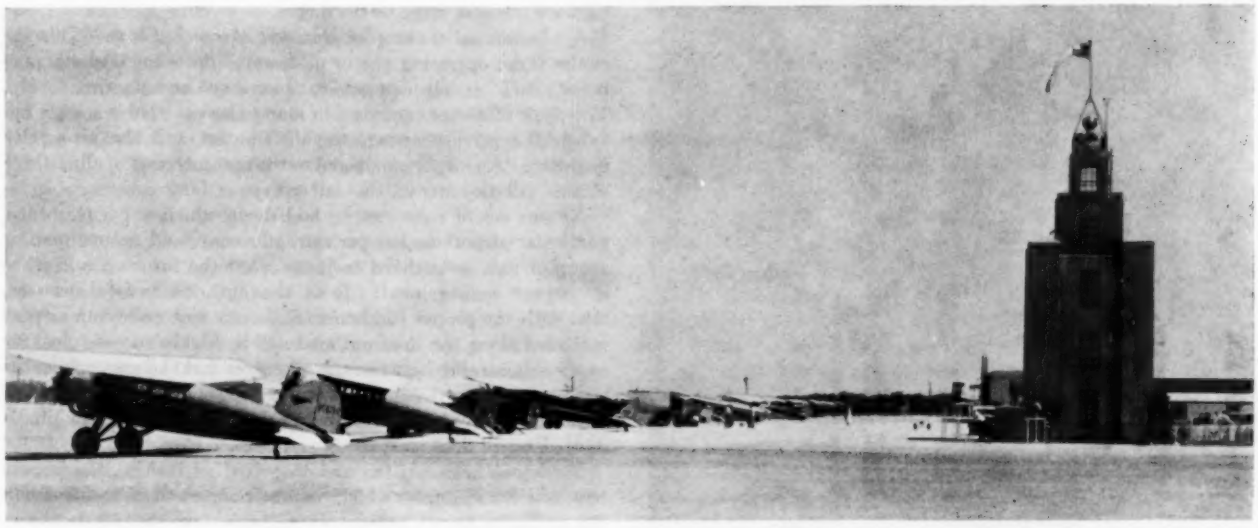


FIG. 9 LINE OF PLANES IN FRONT OF CONTROL AND ADMINISTRATION TOWER

(4) *Dealers and Distributors.* It is greatly to the advantage of dealers and distributors to be located on a port where intensive selling activities are taking place. The modern purchaser of aircraft no longer buys the first plane that he sees which is satisfactory and at the price he wishes to pay, but he will spend considerable time looking at various makes and models. The port having the distribution of the largest variety of planes will attract more buyers than any other. This is being fully realized by many distributors in that the tendency has been lately for them to concentrate as much as possible on one large major airport. The airport fortunate enough to get this type of business soon becomes recognized as a leading aircraft merchandizing center and quickly attracts the other dealers who did not have th e forethought to locate on it in the first place. However, such a situation on an airport immediately makes more problems for the management. Care should be taken that distributors are not located so close to each other that there can be any feeling that

one may steal another's prospects. It is essential that the public at large have easy access to the distributors' quarters and display rooms. He must be provided with office space and given sufficient opportunity to display his merchandise. Airport hangers-on should be discouraged from interfering in any way with the distributors' clients.

While on the subject of hangers-on, this type of individual, although seemingly at many times obnoxious, is an asset to the industry. These people are usually super-airminded and are often the go-between in many deals. They usually give a willing



FIG. 10 FACTORY OF COMMERCIAL AIRCRAFT CORPORATION AT LOS ANGELES AIRPORT



FIG. 11 CONTROL AND ADMINISTRATION TOWER WHICH HOUSES METEOROLOGICAL INSTRUMENTS, DISPATCHER'S ROOM, AND FIELD OPERATIONS OFFICES

hand when called upon and can be made generally useful. Of course, there are the out-and-out airport bums who certainly are not assets, and steps must necessarily be taken to eliminate this type of individual.

(5) *Manufacturers.* On airports where industry is encouraged, manufacturers of aircraft, engines, and other commodities are usually found. It is well that such industries be so located that they do not come in direct contact with the other airport activities. Their premises should be suitably fenced and protected.

There are many facilities which an airport can well afford to install for the use of aircraft manufacturers and others, such as weighing scales capable of weighing large aircraft as units, compass-swinging bases, etc. Not only the manufacturer but the dealer and distributor finds railroad trackage essential. Restaurants should be provided which serve good meals at low prices for factory employees. Given these conditions in a good labor market, together with suitable airport test facilities, one has the principal requisites which an airport management should provide to attract manufacturers. Of course, the geographic location of the airport is also a prominent factor.

(6) *Private Owners.* The needs of the private owner are varied. He wants his plane stored where it will not be tampered with by others and where it may be kept clean and ready to go at all times. Service should be supplied which makes it possible for him to have his plane on the line, warmed up, and ready to go at any designated time. He also wants his friends to be shown the proper deference when they come to the airport, either with him or calling for him. He usually has a certain personal pride in his plane and wants his acquaintances to see it and know that it is his. Particular care should always be exercised by hangar crews in handling all planes between the line and into the hangars, not only of private owners, but of all ships. All airport operators have been confronted with the problem of an occasional minor damage to aircraft in this process, and it is money and effort well spent to repair such damage at the airport's expense and immediately report same to the owner.

(7) *Individual Owners of Business Aircraft Fleets.* This includes those operating one or more small fleets for business purposes, such as oil companies, accessory manufacturers, etc. This type of owner can come in many classes. He is a cross between the private owner, the distributor, and the air-service operator. An airport managed to the best interests of all of these classes will also attract this latter type of business.

No one set of rules can be laid down which will govern any particular airport to 100 per cent efficiency, and no one man or group of men is qualified to judge what the future has in store in airport management. It is the author's belief, however, that with the proper fundamental set-up and policy an airport managed along the lines outlined will be highly successful so far as the relationship between the operators and the management is concerned. The business is in a highly experimental stage, however, and only the future can determine the correct procedure along all lines.

Much has been written and theorized on making the airport pay, but the examples which may be pointed to of such cases in this country are at present exceedingly few and far between. Perhaps an outstanding example along these lines is the rapid growth and development of the Los Angeles Metropolitan Airport.

This project was originally conceived and backed by the finances of a group of Los Angeles and Hollywood business men. The ideal of their conception was a port planned and operated to meet all needs of commercial aviation. In the short space of thirteen months after the port was opened for operation it has been placed on a paying basis. When it is taken into consideration that the entire plan and layout had been developed with the future rather than the present in view, this rapid progress can be considered unusual. There are many fundamental theories upon which the airport has been developed, the strict adherence to which has been largely responsible for its success.

The first of many problems which had to be considered was that of location, and the site of 394 acres located in the heart of the San Fernando Valley was purchased only after a careful analysis of its merits with respect to accessibility to the center of the buying power of the metropolitan area, meteorological con-

ditions, cost of the land, and cost of preparing the land for airport purposes.

The next problem was that of laying out the field in such a manner that it would adequately serve all requirements imposed upon it for future flying activities and building development without the unnecessary waste of any acreage.

It was then necessary to install needed improvements, such as runway surfacing, hangars, lighting equipment, restrooms, restaurant, and administration facilities, in conformity with the scope of potential business. No effort was made to install any improvements for the sake of display or effect, but purely those that were essential to the contemplated activities, with the thought continuously in mind as to what the future might hold in store.

The next step, then, was merchandizing the airport. In order to do this effectively, it was necessary to have a set-up which was not only diversified but fundamentally sound. The plan of development made it possible for manufacturers and operators and other allied industries to procure sites for their activities by either lease, lease with an option to purchase, purchase for cash, or on time. Such a plan enabled those locating on the Los Angeles Metropolitan Airport to get actual title to the property on which their buildings were located if they so desired. For those not wishing to put up their own buildings the airport company is supplying the necessary buildings on a lease or a lease with an option-to-purchase basis. This diversified plan has enabled enterprises and allied industries to avail themselves of the airport's facilities in whatever manner most suited to their particular requirements.

The next problem has been that of exploitation. The airport was dedicated on December 16, 1928, with a suitable program which was publicized through the press as extensively as possible. This launched a year's program of letting the public and the aviation industry of this country hear of the Los Angeles Metropolitan Airport with great frequency.

In order to do this, it was felt that among other things it would be highly desirable to obtain as many record-making flights at the port as possible. At no time did the airport, however, in any way sponsor any such flight, but it was felt, as was later justified, that if facilities were made available and equipment installed that would tend to make such record flights more successful if attempted at this port than at other ports, activities of that nature would naturally be attracted.

The first of a series of such records to start from the port was the famous flight of the "Question Mark." Before this was completed, Miss Bobbie Trout took off and established the world solo endurance record for women. A few weeks later, Captain Frank Hawks took off from the Metropolitan Airport for his first transcontinental non-stop record, and early in the spring Herbert J. Fahy broke the men's solo endurance record here. In the month of June, Frank Hawks again used the Metropolitan in establishing his non-stop round trip record between New York and Los Angeles. The following month the writer was fortunate enough to establish a new record for altitude, carrying 1000 kilograms pay load, using a Bach Air Transport. The month of August saw the start of Henry Brown, Lee Shoenhair, Roscoe Turner, and the ill-fated John Woods in the Los Angeles-to-Cleveland non-stop air derby. At the close of the year, Misses Bobbie Trout and Elinor Smith established a world refueling endurance record for women, and a few days later Amelia Earhart established a new women's world record for speed by attaining an average of better than 184 miles per hour over the mile course at the port.

The establishment of nine records recognized by the F.A.I. during the year in no small way contributed largely to the exploitation of the port. Exploitation was further developed by building up a friendly relationship with the various newspapers, magazines, and news services. Every single item of news, no matter how small, was immediately transmitted to the press. No attempt was made to cram "publicity" down their throats, with the result of building up a very friendly relationship with the press. A conservative advertising program was carried out through the distribution of literature, the advertising in trade magazines, and occasionally by newspaper advertising.

Business has not been built at the Los Angeles Metropolitan Airport merely by exploitation. So far as has been possible, every branch of the aviation industry and allied industries, whether apparently suitably located or not, has been interviewed by representatives of the airport company with the idea of selling them on locating at the port. These activities have not only been local but in some cases national in scope.

Getting business is one thing and keeping it is another. The



FIG. 12 GOODYEAR BLIMP AT SANDERS SERVICE STATION, LOS ANGELES AIRPORT

airport company has stayed out of all commercial activities which in any way might be competitive with those located here on the port and is therefore in an admirable position to remain neutral on the many problems which daily arise in the operation of all ports.

Every month has shown an increase in business. Starting in with nothing on December 1, 1928, the Los Angeles Metropolitan Airport on January 1, 1930, could claim the following: There were 95 airplanes permanently based at the port; six hangars capable of housing 125 planes; two airplane factories; one engine factory; dealers and distributors representing 14 makes of aircraft, who have sold twice as many airplanes in the last six months as have been sold at any other airport west of the Rocky Mountains; four schools; four gasoline filling stations; three service and repair shops; and many private owners.

Revenues have been derived from many sources, the principal source of income being from the rental or lease of hangar space, hangars, and ground space. Other income has been derived from rental of the restaurant, refreshment stand, office, and shop space. Revenue in no small amount has been obtained from motion-picture companies while on location at the port. This form of activity has also proved quite lucrative to the various operators, shops, and restaurants. Certain portions of the airport which are held for future development bring a fair revenue from farming.

The year of 1929 showed an operating cost of approximately $52,000, with an income of approximately $42,500. Contracts at hand and business located on the airport as of January 1, 1930, show an income of approximately $60,000, with little reason for much increase in operating costs over the preceding year. Without any increase in the present business, a profit of $8000 for the year 1930 can reasonably be expected.

An analysis of the income of Metropolitan Airport shows that approximately 75 per cent of the revenue is derived from rentals, the remaining 25 per cent being the revenue from sales of land to manufacturers and operators, less the cost of improvements and alterations.

At this time, approximately 140 acres of the flying area are being sown to alfalfa, which from time to time during the coming year will be irrigated and cut. Of course, it will be necessary to cut this alfalfa much more frequently than if grown for agricultural purposes, yet it is anticipated that revenue from this source and other agricultural activities will approximate $20,000.

Fully 80 per cent of the floor space erected by the airport company is occupied, and with the same increase as has in the past been enjoyed it will in a short time be filled to capacity. The investment in buildings and improvements on the part of the airport company represents $130,000, while there is an investment slightly in excess of this amount which has been spent in building improvements on the part of companies located at the airport.

There has been no one single factor which has been responsible for the financial success of the Los Angeles Metropolitan Airport, but the many contributing factors as previously outlined have all had their place in doing their share toward the development.

Airplane Instrument Vibration

Report of the A.S.M.E. Special Research Committee on Airplane Vibration With Special Reference to Instruments[1]

THE problem of airplane vibration with special reference to instruments is one of vital interest to both aircraft manufacturers and operators, since safety in flying depends to such a large extent on accurate instruments. The present investigation is one of a preliminary character only, and had for its purposes: (1) The securing of information on difficulties encountered in practice, (2) the collecting of opinions as to possible remedies, and (3) formulating suggestions for research.

With these purposes in mind, the committee sent out a questionnaire to pilots, airplane operators, aircraft manufacturers, aircraft instrument makers, and others in the aviation industry.

Questionnaire and Analysis of Replies

The questionnaire and analysis of replies follow:

1 *Have you experienced any difficulties with airplane or airplane-engine instruments due to vibration?*

Almost every reply stated that instrument difficulties due to vibration had been experienced.

2 *If so, on what types of airplanes?*

The airplanes on which instrument vibration occurred covered all types, the reply as to type being dependent upon the particular operator's experience. It is safe to say that instrument vibration difficulties are not limited to any one type, or to a few types, of airplanes.

3 *What was the general construction of the airplane in which such difficulties were experienced?*

Such difficulties were not limited to any one type of airplane, nor were they limited to any one type of fuselage construction. Difficulties were experienced in all types of fuselage construction, whether of welded steel tubing, riveted dural, or composite metal and wood. All phases of structure were evidently covered in the replies and no particular type of fuselage structure seemed to be free from vibration difficulties.

4 *What was the character of engine mounting?*

The engine mount was constructed of the same material, generally, as the fuselage. Some engine mounts were detachable, while others were not. Wooden engine bearers were employed with some engine mounts, while others had the engine mounted on rubber blocks.

5 *What engines were employed?*

All engines, whether radial, in-line, vee, etc., were used on the planes which had instrument vibration troubles. Those who had experience with both radial and other types of engines stressed the fact that radial engines gave the greater trouble, while the six-in-line gave less trouble. These comparisons were made with engines in different airplanes; no particular investigation was made to determine if the trouble still persisted if a different type of engine was substituted in the same airplane. Nevertheless, since the airplanes are more or less similar in construction, the diagnosis as to which type engine gave the better results may be assumed as correct.

6 *What p opeller was used?*

Both metal and wooden propellers were used. If excessive instrument vibration trouble occurred due to the propeller, it was usually traced to propeller unbalance and therefore could easily be corrected. In no case was instrument vibration difficulties thought to be due to some inherent fault in propeller design.

7 *Where was the instrument board mounted?*

The instrument board location aft of the engine was generally dependent upon the location of the cockpit. The conventional place was slightly in the rear of the fire wall. (The fire wall was usually located as close to the engine as possible.) The instrument boards were usually suspended between the two upper longerons.

8 *What was the distance from the instrument mounting board to the engine mount?*

The distance from instrument board to engine varied from 7 in. in the rear of the fire wall (which was the closest of all cases cited) to as far away as 10 ft. The average distance in the rear of the engine mount was about 2 ft.

9 *What were the material and thickness of instrument board?*

Aluminum or duralumin were the common materials used in the construction of the instrument boards. A few instrument boards were reported to be of plywood construction, while one operator mentioned basswood. The common thicknesses for metal boards were either 0.020 in., the thinnest used, or 0.050 in., the most common thickness, or 0.091 in., which was the thickest used.

10 *What was the method of attaching the instrument board? Were mountings flexible or rigid?*

A few instrument boards were bolted directly to the longerons or cowling of the fuselage, while a few were mounted on felt washers or sponge rubber. The majority of the instrument boards were rigidly mounted, although one or two were reported to be flexibly mounted.

---

[1] The personnel of the Committee is as follows:

Klemin, Professor Alexander, *Chairman*, Daniel Guggenheim School of Aeronautics, New York University, New York, N. Y.

Aldrin, Captain E. E., Manager, Aviation Department, Standard Oil Development Co., New York, N. Y.

Backus, Harold A., Metallurgical Laboratory Inc., Philadelphia, Pa.

Baugh, Lieut.-Commdr. H. V., General Inspector of Naval Aircraft, Eastern District, New York, N. Y.

Budwig, G. G., Chief of Inspection Section, Department of Commerce, Washington, D. C.

Campbell, C. W., Cambridge Instrument Co. Inc., New York, N. Y.

Colvin, Charles H., President and General Manager, Pioneer Instrument Co., Brooklyn, N. Y.

Depew, Captain Richard H., Jr., Sales Engineer, Fairchild Airplane Manufacturing Corp., New York, N. Y.

Durand, William F., Emeritus Professor, Mechanical Engineering, Stanford University, Cal.

Hagenberger, Lieutenant A. F., Instrument and Navigation Unit, Matériel Division, Air Corps, Wright Field, Dayton, Ohio.

Ileman, E. A., Chief Engineer, Consolidated Instrument Co. of America, Inc., New York, N. Y.

McCarthy, C. J., Chance Vought Corp., Long Island City, N. Y.

Newell, Floyd B., Engineer in Charge of Aeronautical Instrument Division, Taylor Instrument Co., Rochester, N. Y.

Smith, Wesley L., Superintendent, Eastern Division, National Air Transport Inc., Air Mail Field, Cleveland, Ohio.

Presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

11 *Did the instrument board form part of the fire wall?*

In the majority of cases, the instrument board did not form part of the fire wall. One operator replied that it was impractical to have the instrument board form part of the fire wall.

12 *What was the exact character of the instrument trouble experienced?*

Excessive vibrations caused oscillations of the compass card so that it was practically impossible to read. All instruments having indicating pointers were difficult to read since the indicator vibrated so badly that readings were inaccurate.

In some cases, continued and excessive vibrations caused the indicating hand to drop off if it was not held in place securely by a small cotter pin.

13 *If the trouble was eliminated, how was it eliminated?*

The answers were few on this subject, probably indicating that either the trouble had not generally been overcome, or no particular attempt had been made to overcome it. The difficulty in trying to eliminate instrument vibration usually lay in determining the real cause.

A few eliminated the trouble by flexibly mounting the instrument boards, while others did the direct opposite and rigidly mounted the boards. One eliminated the trouble by changing engines, but this method is not expedient.

14 *Was the vibration difficulty experienced at all engine speeds, or at specific engine speeds?*

Difficulty was generally experienced at all engine speeds. Some noticed difficulty at varying engine speeds, and still others found that vibration difficulties were experienced at one or two engine speeds.

15 *Did all instruments vibrate at the same or at different engine speeds?*

There were very few replies to this question. Two correspondents found that various instruments vibrated at different engine speeds, while three found that the instruments vibrated at the same engine speeds.

16 *With what instruments was vibration difficulty experienced?*

The compass seemed to be affected most severely of all instruments. Altimeter and tachometer readings were also affected by vibration.

17 *What was your diagnosis of the trouble?*

The diagnosis by the correspondents were equally divided among five causes:

1 *Faulty Engine Mounts.* The engine mounts were either too rigid or too flexible. There seemed to be no agreement among the correspondents.

2 *Faulty Instruments.* Improper balancing for dynamic conditions were thought to be the cause of inaccurate readings.

3 *Faulty Instrument-Board Mountings.* The boards were considered to be of cheap construction, and improperly mounted, either too rigidly or too flexibly.

4 *Engine.* Some engines have inherent vibration weaknesses which affect the entire aircraft structure.

5 *Propeller Balance.* Wherever propeller unbalance was found to be the cause of instrument vibration, the operator was able to remedy the difficulty, so that this may not be considered an inherent fault.

18 *What are your general recommendations as regards elimination of instrument vibration difficulties?*

(*a*) *Structure.* A few suggested that the fuselage structure might be made more rigid by more bracing; the structure in the vicinity of the instruments especially should be well braced. Changes in structures, if made at all, should only be made if all other methods have failed.

(*b*) *Engine Mounting.* Rubber or fiber mounting for engine bearers were generally favored in order to obtain a slightly more flexible engine mounting.

A few consider a more rigid engine mount desirable.

(*c*) *Mounting of Instrument Board.* The consensus of opinion was that a flexible instrument board mounting was essential. The flexibility could be obtained either by rubber, fiber, or spring mountings. The degree of flexibility would depend upon the particular installation and would require some investigation.

Locating the instrument board as far away from the engine as possible is usually a very good means of eliminating vibration difficulties. Most of the present-day instrument trouble could perhaps be diminished by proper location.

(*d*) *Mounting of Instruments.* The opinions were, on the whole, very contradictory; both rigid and flexible mounting of the instruments found equal favor. However, if the instruments themselves were not flexibly mounted, then the instrument board should be.

(*e*) *Improvement in the Instruments Themselves.* It was suggested that the instruments could be improved (1) by better damping and balancing of component parts and (2) by more rugged construction of the instruments so as to be unaffected by vibrations. It was conceded that the more rugged the instruments were made, the more sluggish they would likely become.

General Impression of Replies

A survey of the replies to the questionnaire shows that comparatively little is known as to the exact cause or causes of instrument vibration. In some cases, the answers were too brief to analyze the whole properly. However, the questionnaire did accomplish its primary object. The following items stand out:

1 That instrument vibration is not limited to any type of plane, fuselage construction, engine mount, or engine or propeller make.

2 That changes in fuselage structure or engine mount are not considered essential.

3 That instrument vibration is not necessarily limited to specific engine speeds.

4 That practically all instruments are equally affected by vibration.

5 That partial elimination of vibration difficulty may be obtained by placing the instruments as far away from the engine mount as is conveniently possible.

6 That elimination of vibration difficulties may be found by proper mounting of the instrument board and instruments, and perhaps further refinements in the instruments themselves.

Suggestions for Further Research

Further research for the mounting of instruments and instrument boards perhaps should more properly be taken up by some governmental agency rather than by the aircraft manufacturer or the aircraft-instrument manufacturer. Refinements in the instruments, of course, can be more easily made by the instrument manufacturer.

Several suggestions for research have been made:

1 Records of vibrations of the instrument boards of various airplanes should be made. These records could be studied for intensities and frequencies of the vibrations affecting instruments. After the intensities and frequencies had been classified, they could be duplicated in the laboratory, and tried on various mountings of instruments and instrument boards. In this manner the comparative merits of flexible or rigid mountings of either instruments or instrument boards could be definitely and progressively studied until the difficulty had been eliminated. Flexible mounting could be obtained by use of rubber, fiber, or small steel springs combined with a small dashpot.

Steel springs have natural vibration frequency of their own, so that any spring system would have to be thoroughly tested out before the vibration of instruments could be overcome.

2 Experiments should be made with the instrument board mounted on steel springs whose tension can be adjusted. It would be by this means that the instrument board could be adapted to any condition met with an airplane.

3 Experiments could be conducted on different weights for instrument boards. It is possible that the instrument board is insufficiently loaded; a flexible mounting might then not effect much improvement unless it were too flexible to be practical. The loading of the instrument board could be varied to increase its natural period, and then combined with a flexible mounting to eliminate vibration difficulties.

4 Further refinements in instrument construction could be carried out by the instrument manufacturer. It has been suggested that an application of the principle of the Lanchester vibration damper may help those instruments which employ indicating pointers. The instrument pointer could be mounted on its pivot and driven through some simple frictional or hydraulic connection operated by a somewhat heavier pointer. The Lanchester vibration damper would not, however, take care of dynamic unbalance and therefore would practically be useless.

Hydraulic systems are not reliable since they are affected by temperature and humidity variations.

As soon as a definite program for research is successfully conducted, other improvements will naturally suggest themselves.

Grateful acknowledgment is made to all those who contributed replies and information to the committee.

Discussion

E. H. Hull.[2] Until airplane engines and propellers can be made nearly vibrationless by proper design and balance or isolated from the fuselage by elastic supports, a device such as is suggested in paragraph 6 of "General Impression of Replies"—namely, an elastic suspension of the instrument board—will be desirable.

Referring to "Suggestions for Further Research," the frequencies present in the instrument board, with their approximate relative amplitudes, can be quickly determined by Frahm-type vibration tachometers. The frequencies of response of the instrument parts such as pointers can be determined by observing them, while in vibration, through a stroboscope.

Knowing the lowest impressed frequency which causes trouble, a frequency of suspension of the instrument board can be chosen, using the expression

$$e = \frac{1}{\left(\frac{\omega}{\omega_c}\right)^2 - 1}$$

where the transmissibility e is the ratio of the amplitude of the board after suspension to the impressed amplitude, ω is the impressed frequency, and ω_c is the natural frequency of the board on its elastic supports. Any elastic material may be used in the suspension. The board itself should be made as rigid as possible to avoid flexural vibrations in itself.

Helical steel springs "have a natural frequency of their own" when the ends are held and the middle part vibrated as in valve-spring surge, but this frequency is very high and should not be troublesome here. The instrument board will show a natural frequency (the ω_c mentioned) when mounted by any elastic means, such as rubber, springs, cork, etc. It is necessary for the impressed frequency to be well above this natural frequency of the board on the mounting before any benefit can be gained from the elastic mounting.

Stops incorporating heavy damping for limiting the motion of the board would be desirable at take-off and landing. Connections, electrical and mechanical, to the instruments mounted on the board must be flexible.

When a body is supported between two springs, its natural frequency in the line of the springs is not dependent on the initial tension in the springs. Thus the mounting cannot be varied to meet requirements of different stiffness along the spring axis by varying the spring tension. The spring constant itself must be changed.

Loading the instrument board will decrease its natural frequency since $\omega_c = \sqrt{\frac{c}{M}}$, when c is stiffness and M mass.

Light or heavy boards can be made to give the same isolation providing the natural frequency of each, determined by its elastic support, is the same. If the board is made heavy, the stiffness of the supporting means may be increased in proportion, maintaining the same natural frequency on these supports, and hence the same isolation. The isolation obtained in any case depends on the natural frequency of the board on its mounting; the lower this frequency the better will be the isolation.

J. Ormondroyd.[3] The question of proper mounting for airplane instruments comes in the field of general vibration theory. In order to treat the question rationally, the source and frequency of the disturbing force must be known and the elastic and mass properties of the instrument mounting must be known. The problem of adjusting the instrument mounting to the known disturbances is one which has been solved in a tremendous number of engineering problems in other fields. The probability is overwhelming that the disturbances come from the engine rather than from any aerodynamic instability at the airfoil surfaces. Since an airplane engine operates most of its time at two more or less fixed speeds, its full speed and its cruising speed, it is possible to predict at least the frequencies of all the disturbances which can arise at the motor. These disturbances will have engine r.p.m. frequency or some multiple of that frequency. The possible sources of disturbance are:

1 Unbalance—one cycle per revolution

2 Torsional critical speed—some multiple taken in half steps of r.p.m.

3 Torque reaction—firing frequency

4 Crankcase vibration—one cycle per revolution.

There are known methods for reducing unbalance, torsional critical speed, and crankcase vibration to unimportant magnitudes. In most engines to be found in modern airplanes these steps have been taken. Torque reaction, however, is a phenomenon so inherent in the operation of the internal-combustion engine that the only way to get rid of it is to stop the engine. However, since torque reaction is inevitable, instrument mounting should be so designed that the frequencies from torque reaction at the running speeds are not in resonance with the instrument on its mounting.

Technical literature is filled with papers on the subject of spring mounting. It is to this field that the designers should turn their efforts. In line with this, probably the most important step which can be taken would be toward the accurate measurement of the frequencies and amplitudes of the disturbance actually found at the instrument boards. Any number of commercial instruments for the recording of vibrations are already in existence, and any number of special electrical devices for measuring these quantities have already been devised by various manufacturers who have had vibration troubles in their own equipment.

[2] Research Laboratory, General Electric Company, Schenectady, N. Y. Jun. A.S.M.E.

[3] Experimental Division Manager, Westinghouse Elec. & Mfg. Co., Philadelphia, Pa.

The group of manufacturers interested in airplane instruments should be warned that they must stand on their own feet in the solution of this problem. While the Government is very active in certain centers, such as the Brooklyn Navy Yard, Wright Field, and Langley Field, in the study of vibration problems, it does not have the same immediate interest in their solutions as those interested in the commercial exploitation of the parts affected.

The writer believes that the questionnaire method of starting the solution of any real technical problem is fundamentally wrong. The committee should get a man who can study the theory of vibration and work out its immediate application of the problem in hand. The questionnaire merely dodges the real issue by making what is distinctly one man's business, everybody's business, and therefore nobody's business. Surely there is somebody in the airplane industry or in the airplane instrument industry who can go deeply into the study of vibration theory and bring order out of the chaos of ideas which is shown to exist in the answers to the questionnaire.

S. J. Zand.[4] It has been the writer's experience that instruments perform very satisfactorily on some ships, but, due to the excessive vibrations of other ships, the same instruments fail. The problem therefore is to find out exactly the source of these vibrations and eliminate them as much as possible. This is similar to the solution which Mr. John Lee suggested of eliminating noise in the airplane cabin. "It would be much wiser," he said, "to remove the source of noise, rather than dampen the noise by insulating the cabin." Similarly, it would be much wiser to eliminate airplane vibrations rather than try to eliminate the vibrations of the instrument board alone.

The problem of airplane vibrations and its relation to instrument performance is much more complex than generally realized. Any elastic system or any elastic body which is suddenly subjected to an external force, or if a continuously applied force is suddenly removed, will vibrate; which means that the elastic forces of the system will no longer be in equilibrium with the loading.

The airplane is a structure which consists of many individual parts, and each part will be subject to what is called natural vibrations, which in turn will produce on the instrument board a resultant vibration of quite a complicated nature.

To solve this problem theoretically would be a task great enough for a mathematician of the highest skill. The writer does not want to say that a theoretical solution is impossible, but anybody who has had experience with differential equations will agree that the interpretation of the same for practical purposes is very difficult and very often impossible.

It is the writer's conviction that the engine is the main offending part. This is due to the difficulty of balancing any internal-combustion engine. In the case of radial engines, perfect balancing is impossible, at least at the present time.

Empirical methods in accord with the recommendations by the committee are favorably regarded, and for this purpose an apparatus faithfully recording the vibrations is the first item necessary.

There are a number of very fine vibrographs on the market, but, owing to their weight and size, none is particularly suitable for aircraft use.

The size of such an instrument should be the same as of all other aircraft instruments, and the weight should be similar to that of one of the standard aircraft instruments, so that substitution can take place without materially changing the loading of the instrument board. Such an instrument is not easy to construct as it involves a number of parts which have to be assembled in a very small case.

In order to have a reference point which will not move, a long-period pendulum is necessary. Such a pendulum ought to have considerable length, which is in direct contradiction to the statement made regarding the small size. The writer is firmly convinced, however, that by using differential, harmonic, or even gyroscopic pendulums, an apparatus recording the vibrations of an airplane can be constructed in the standard airplane instrument size, but it is, as explained, a rather difficult proposition.

The stiffness of instrument boards should be studied. Any attempt to load instrument boards to increase their natural period should be discouraged; adding weight to any existing part of the airplane is poor practice, and even if it should prove one of the solutions, it will not be acceptable to the airplane manufacturers.

Owing to such obvious reasons that need not be mentioned here, the use of dashpots as a dampening medium for instrument boards is entirely out of the question. The suggestion referring to the use of steel springs should be taken with reserve. It should be borne in mind that steel springs produce changeable magnetic disturbances which might affect the compass. Furthermore, steel springs corrode. Brass or Monel metal could be used more advantageously.

There is no difficulty in figuring the natural frequency of springs. It can be done in a few minutes by the following procedure: Denoting k, the load necessary to produce a unit extension of the spring, then the static deflection of the spring under the weight W will be $\delta = W/k$, and at once the period $2\pi\sqrt{\frac{\delta}{g}}$ and the frequency $2\pi\sqrt{\frac{g}{\delta}}$.

The k is a constant of the spring and can be either measured or else found from the diameter of the spring, diameter of the wire, and characteristics of the material used; a simple mathematical calculation and any engineering handbook gives all the necessary formulas.

The best solution to overcome excessive instrument-board vibration is to suspend the same pendulously. If the instrument board should have the same period of vibration as the r.p.m. of the engine in the case of a rigid suspension, then the same instrument board, in order to be immune to vibrations, ought to be suspended in such a way as to have a period of $1200/60 \times 1/5 = 4$ sec. (assuming the lowest flying r.p.m. to be 1200). It is at once apparent that to obtain such a low period, considering the space available for the spring suspension, is not easy.

In paragraph 4 of "Suggestions for Further Research," the use of a Lanchester damper is advocated. This is highly impractical, as the Lanchester dampener is used to dampen tortional vibrations of crankshafts, i.e., of bodies of considerable mass and revolving at high speeds. The pointer of an instrument is the lightest part and revolves very slowly. For example in a rate-of-climb indicator the pointer revolves with a maximum speed of about 1/2 r.p.m., and in 10,000 ft. altimeters, about 1/5 r.p.m. These values are maximum for commercial airplanes.

When one hears about improvements in instrument design suggested by people outside of the instrument industry, he is naturally inclined to be slightly skeptical. The majority of engineers, outside of the instrument makers, hardly realize the complexity of the problems. Aircraft instruments are compromises, and a very brief illustration will prove this statement.

The main difficulty when designing instruments for aircraft use is the fact that the power available to produce indication is very small. In an altimeter, for example, for a difference

[4] Aeronautical Engineer, Pioneer Instrument Company, Brooklyn, N. Y.

of 1000 ft. the available power is equal to about 1 in. of mercury pressure difference. For a range of 10,000 ft. there will be, due to the logarithmic form of the pressure-altitude curve, about 6 in. of mercury. The movement of the diaphragm, due to that pressure difference, is not larger than $^1/_{32}$ in. This movement, in order to be read with ease, has to be amplified tremendously, from 100 to 400 times. Thus it is seen at once that any attempt to introduce additional friction to dampen out errors due to vibration would be ill-considered. Friction must be eliminated as much as possible in order to achieve the amplification of motion aforementioned.

If one looks at a sketch of an altimeter he sees a great similarity to a vibrating reed. That the instrument makers are able to produce very reliable instruments which will stand some vibration is not a mere coincidence, but the result of very careful design and a great many calculations.

When an instrument performs satisfactorily on the majority of airplanes, but fails on some of them, it is not the fault of the instrument, but generally the failure can be traced to terrific vibration of the instrument board or parts of the structure near it.

The work of the committee is appreciated, as it will no doubt give the incentive both to the airplane motor and instrument manufacturers to study the vibration problem thoroughly. It is unfortunate that the committee could not obtain quantitative and qualitative data on vibrations. Without this information, very little can be accomplished.

Through the proper cooperation of the airplane industry with the instrument manufacturers, and possibly with the help of Government agencies, such as the Bureau of Standards, a very satisfactory solution of the problem can be obtained, and, needless to say, instrument manufacturers will be delighted to cooperate.

Authops' Closure

Many thanks are due to Mr. Hull for his valuable suggestions on practical methods for further research. It is hoped these suggestions will be acted upon by experimenters in the field.

Mr. Ormondroyd's reference to technical literature will be studied further, and it is hoped, at a later date, to issue supplementary data on this literature.

Mr. Zand's discussion is equally as valuable and will undoubtedly provide stimulus for further work in the field.

AER-53-14

Airplane Cabin Engineering

By PIERRE FREYSS,[1] NEW YORK, N. Y., and JOHN F. HARDECKER,[2] PHILADELPHIA, PA.

The intense interest in the development of the interior design of the airplane is the national consequence of the competition between the airplane, the Pullman car, the automobile, and the motor bus for the patronage of the general public. Appealing as the plane does to that class whose time is of extreme value, it must offer a degree of comfort, convenience, and luxury comparable with other means of transportation.

While essentially a problem of the decorator, interior design demands the whole-hearted cooperation of the aeronautical engineer. It must be always in mind from the inception of the structural design, so that the maximum opportunity for eye appeal is assured the decorator. Wall, ceiling and floor treatments, hardware, sanitary equipment, accessories, chairs, windows, doors, soundproofing, and fireproofing, all are within the scope of interior design. All must be blended into a harmonious whole, keeping ever in the forefront of the design the fundamental aeronautical requirements of minimum weight and maximum space utilization.



Pierre Freyss



John F. Hardecker

MODERN airplanes are no longer mere flying machines; they are a mode of transportation rapidly gaining equality with long-established contemporaries in many respects and already far exceeding them in certain phases. As a fitting climax to this rapid cycle of evolution, the proponents of air travel have become aggressively interior-conscious. Luxury evolved after literally scores of years in other means of transportation is demanded in a few short years, not as a consequence of mere pride or ambition, but as a necessity of travel competition.

Flying is rapidly ceasing to be an adventure. The ownership and utilization of a plane as a means of transportation is a matter of convenience and comfort. The development of large transport cabin planes has made it possible to travel by air with a security and comfort comparable to the automobile and motor boat. It is always a proof of stabilization when an industry so fundamentally mechanistic as aircraft reaches a point where, in addition to satisfactory functioning, there arises a distinct demand for elegance, style, and beauty.

This demand for interior styling is a natural consequence of the expansion of the prospect lists for aeronautical sales, either of planes or transportation. The youthful enthusiast who spends freely of his time about flying fields, anxiously soaking up technical details of aircraft construction and operation, ever willing to do anything to get a flight or get into aviation, is still worthy of cultivation in the general scheme of things aeronautical, but he is not the best prospect for sales. His sympathetic understanding of the many ramifications of a highly technical profession is indeed gratifying to the ego of the aeronautical specialist, but his is, primarily, an ambition to find in aviation a means of livelihood.

The real prospects are the individuals of means and large corporations. These prospects have neither the time nor inclination to master the details of the theory of flight, structural analysis, or flight operations, but they depend to a large extent on eye appeal in establishing their confidence in air transportation.

This attitude is not a consequence of indifference or laziness, but a natural derivative of the popularization of other means of transportation. Surely the utilization of a Pullman car as a means of travel does not demand an intimate understanding of railroad construction and operation, nor the ownership of a private automobile that its possessor must be an expert on gasoline engines. More and more, these competing means of transportation depend upon the luxury and comfort of their interiors for sales appeal. The more the airplane can suggest the same degree of ease and refinement in its interiors, the better able it becomes to meet travel competition.

It would therefore appear that the problem of interior design might be relatively simple at first glance if style standards are adopted which are already stabilized by competing transportation agencies. Immediately following this thought the question of weight enters the mind, and the designer is face to face with perhaps the greatest problem to be encountered in plane interiors. Practically every material or part evolved in quantity production for other industries, and therefore decidedly attractive in price and availability to the airplane manufacturer, is automatically barred on this single count—it is too heavy. This single factor alone is responsible for the fact that originality is necessary from the very inception of interior design, an originality which requires extreme ingenuity in an industry which at present is far from mass production possibilities.

Following closely on the heels of the weight inhibition is the restriction on design imposed by space limitations. Luxury and comfort are most easily accomplished and suggested by the generous utilization of space; to achieve the same effect in the decidedly restricted confines of an airplane interior calls for extreme skill and artistic legerdemain. In addition, and each a distinct problem within itself, is the need for soundproofing, fireproofing, and the providing of the many accessories to comfort so readily supplied in other transportation mediums, which in the airplane become a distinct problem in specialized interior design.

Preliminary Engineering Considerations

While this matter of interior design is essentially the problem of the decorator, he must be assured a close degree of coopera-

[1] With F. Schumacher and Co., New York, N. Y. Served in French Army from 1914 until the end of the War. Received four citations for French Croix de Guerre and Medal. Studied at Hautes Etudes Commerciales, Paris. Joined F. Schumacher and Co. in 1925.

[2] Chief Draftsman, Naval Aircraft Factory, Philadelphia, Pa., and has served this firm since 1917 in various engineering and executive capacities. Received Civil Engineering degree from Cornell University in 1917.

Contributed by the Aeronautic Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

tion from the aeronautical engineer if he is to accomplish the most desirable results. No greater mistake in management can be made than to design a commercial passenger plane purely on the basis of aerodynamical and structural considerations, and on its completion, call in the decorator and defy him to make an attractive interior out of the resultant cut-up, odd-angled interior spaces.

Windows and doors should be a primary consideration in the preliminary layout of the plane design. Windows should be preferably long and narrow, in order to accentuate the apparent length of the cabin, and odd shapes should be avoided. It is also highly desirable that no construction members show through the window opening, as the general public does not have a sufficiently adequate conception of the strength inherent in tubular members of small diameter. Doors should not only be of regular shape, but they should be sufficiently large for natural entrance and exit, and should be made sufficiently thick to inspire confidence, as to the public they often suggest



FIG. 1 CABIN OF BURNELLI, 20-PASSENGER AIR TRANSPORT. ARMCHAIRS TAKE ANY POSITION. OWING TO THE ALL-WING TYPE OF FUSELAGE THE CABIN IS VERY WIDE



FIG. 2 LUXURIOUS SIKORSKY AMPHIBION. PRATT AND WHITNEY PURCHASED THIS PLANE FOR USE OF ITS EXECUTIVES



FIG. 3 BUSINESS PLANE UPHOLSTERED IN CHASE VELMO MOHAIR VELVET



FIG. 4 CONTROL CABIN OF CURTISS CONDOR AIRPLANE

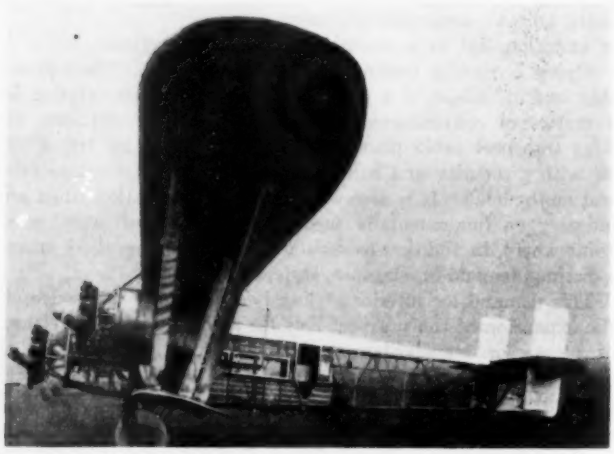


FIG. 5 BALSAM WOOL APPLIED TO CABIN FOR INSULATION

the only protection against being suddenly catapulted off into space.

In arranging for lavatory space, due regard should be given to the dimensions of available sanitary units to avoid costly special installations. Floors should be adequately supported and of such section as to give a solid feeling to the feet of the passenger walking over them, as nothing so readily suggests flimsiness to the lay mind as a floor that gives with every step. Windows that are intended to be adjustable must be readily so, and this at least is one respect in which the railroad car is not to be taken as an example of desired practice. Provisions for ventilation and heating are part of the interior design charged to the aeronautical engineer.

All controls and operating mechanisms, while desirable of concealment behind the walls of the cabin, should be so arranged in coordination with the interior finish that they may be readily reached for routine inspection and emergency repair through removable panels that suggest a routine function rather than a hurried tearing down of the plane interior that is suggestive of fear and danger to the passenger. In fact, the military practice of making a layout and mock-up of fixed equipment and useful load in the development of a plane design may well be followed by the commercial-plane designer in his planning of interior design installations.

The foregoing, while indicative of a considerable effort on the part of the aeronautical designer in achieving a satisfactory interior, must be regarded in the light of a task which, well performed, bears fruit in every single unit produced of a given plane model. This may to some extent suggest a monotony of interior design, but it must be borne in mind that once these essentials are properly established for any given model, the decorator by his skill in choosing appointments and finishes can add infinite variety to each individual plane unit. In fact, many planes purchased for private use are custom decorated.

Decorator's General Problems

Avoidance of monotony is one of the decorator's problems which he must be careful not to exceed by over-emphasizing pattern and color, lest the interior disturb the sensitiveness of people subject to air sickness. The decorator must create an interior into which the psychological factor of security enters, and build effects which indirectly assure the naturally timid with their recurrent thoughts of their long distance above the ground. He therefore has to choose tones and textures that have an appearance of substance and reliability and by handling style and line give an assurance of maximum strength. The whole interior should put the individual into an intimate atmosphere, confronting him with materials and fabrics he is accustomed to, and which conceal any unfamiliar members or construction which, to his excited nerves, may suggest fragility. The reduction of noise, the single remaining factor that suggests breakability, is now receiving serious consideration by all aircraft manufacturers, and real progress is being made.

The finished product of the decorator's art is at once a study in harmony and contrasts. It cannot be reduced to a simple engineering formula applicable in all cases. Each plane may be said to be an individual study, an attempt to apply the decorator's skill to the interpretation of the individual wishes of the owner much as the interior decorator functions in finishing the interior of a home, hotel, or club. Therefore no attempt will be made to draw detail word pictures of individually finished interiors. Even the photographs offered, with their accompanying descriptions, are not a sufficient presentation of the assembled effect, for they do not show conclusively for ready perception the wonderful crowning effect of color in creating the true beauty and luxury of the modern airplane interior. Therefore, the major materials, chiefly textiles, which are used for walls and ceilings, will be treated only in a general way, leaving for detail treatment the less obviously visible treatment of the interior, and also the many accessories and fitments which form such a significant part of the coordinated whole.

Wall and Ceiling Finishes

When choosing textiles, it is easier to find suitable decorative fabrics for walls and ceilings with harmonizing contrasts as the plane grows larger in size. These textiles are in a large proportion combination rayon weaves with modernistic designs, as modern art is the most suitable for the decoration of the airplane. The rayon weaves give the glint of silver best. Silver plays a large part in the aeronautical color scheme, as the aerial atmosphere favors its neutral and restful tone. Wool damasks,



Fig. 6 Aluminum Alloy Reclining Airplane Chair



Fig. 7 Magnesium Alloy Straight Chair

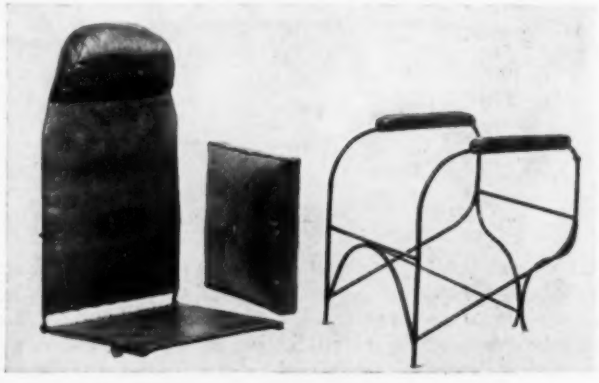


Fig. 8 Knock-Down Type of Construction Used for Airplane Chair

cotton, rayon, and rough-weave Rodier fabrics are most successfully used in the bigger transport planes for sidewall panels. In weight these fabrics vary from 7 to 12 oz. per sq. yd., the general and best weight being from 8 to 9 oz.

The usefulness of a woven fabric has been questioned by many an engineer. Some advance the opinion that it will soil and cannot be cleaned; others fear its inflammability. This latter has been overcome by a special treatment which makes it entirely fireproof, and which will be presented later. Yet fabrics have won their way because they are light in weight, strong, decorative, and do not soil more readily than other materials.

They have withstood the test, and many a large transport plane inspected after two years of service shows fabrics which have neither faded nor stained.

Fabrikoid, an imitation leather, which is a washable fabric, is of course more suitable, but up to the present this lacquered fabric has been produced only in plain colors. These colors are of the same tone as the Duco colors. So instead of being subdued as are the woven cotton and rayon fabrics, they are very decided in coloring. The decorators have tried in many ways to render Fabrikoid more decorative by air-brushing designs on it or even embroidering it with modernistic designs as was done in one cabin of the *Commodore* plane. Yet it has been impossible to obtain the highly decorative soft effects so greatly sought after for plane interiors. Therefore attention has been directed to developing new designs in light and strongly woven cotton and rayon fabrics. The woven fabric has a decided advantage in absorbing sound, a property which the lacquered Fabrikoid lacks.



FIG. 9 REED CHAIRS OF UNIVERSAL AIRPLANE CHAIR COMPANY

The best solution is to combine the use of woven fabrics and Fabrikoid. Places like sidewalls where a decorative effect has to be obtained are covered with a modernistic woven fabric, and Fabrikoid is used on the ceiling, in the lavatory, galley, and entrance. In many cases Fabrikoid is used for seat covering as it is a very strong and resistant fabric which can be washed, while the softer woven fabric is subject to replacements.

Mohair pile fabrics, or mohair velvets, find considerable favor in small planes, where they are utilized in a manner similar to the decoration of an automobile interior. Genuine leather, in upholstery finishes, is sometimes used for chairs or sidewalls in small planes. Many cabins use veneers or plywood for walls or ceiling, finished in natural-wood finishes or treated with the many decorative design finishes used in modern building interiors. In others, plymetal, with its finish surface of aluminum or duralumin, cored with light woods such as balsa, is utilized.

FLOOR COVERING

Carpets are used on the floor. The latest contribution to deaden the sound is the new rubberized carpet, with a heavy pile of wool backed by rubber. This carpet is very easy to place, as contrary to the ordinary carpet, cuts are not joined by sewing. After the carpet has been cut, the two straight edges are pushed together, and a tough web of strapping, impregnated with a special cement, is laid along the back of the joint, overlapping it for several inches on each side. When the cement has set, the joint is firm and unyielding, the resilient back and the strapping having been firmly fused, while on the face of the carpet, the thick pile meshes together and the seam is invisible. As the resilient back is waterproof, the carpet can be scrubbed on the floor. If damaged by stain or burn, the offending spot can be quickly cut out, another piece inserted, and not a trace of the joining will show.

FIREPROOFING TEXTILES

It is highly desirable that the textiles used in plane interiors be fireproofed. The processing of textiles by fireproofing formulas should leave the fabric as nearly as possible in its original color and texture. This can be done in most cases, but as chemicals are made a part of the cloth, in some cases there is a slight change in the color. This change usually consists of a slight darkening.

The fireproofing of textiles is usually done by one of two methods. The first is to impregnate or coat the fibers with materials that do not burn, and in that way exclude the air and raise the ignition temperature. The other method is to impregnate the fibers with chemicals that liberate a gas when they are heated, this gas cutting off the air supply, so that the fabric does not ignite. Of course, both these methods do not prevent the burning of a cloth, but they do eliminate the rapid burning of the fabric and therefore reduce the fire hazard.

WATERPROOFING OF TEXTILES

It is highly desirable to waterproof or showerproof the fabrics used in amphibions, flying boats, and seaplanes. This is usually done as a precaution against water damage of the fabrics. The

showerproofing of textiles also involves the impregnation of the fibers with chemicals that make them resistant to water. In looking at an unprocessed woolen or cotton fabric under the microscope, one notices a decided absorption when it is brought in contact with water. The same fibers that have been showerproofed, by the "Cravenette" process would not absorb water and if viewed under the microscope would show some beads of water clinging to the outer surface.

There are a great many methods of showerproofing cloths and their effectiveness is as varied as the processes. Most showerproofing processes consist of some surface coating of the cloth which has no permanence and is not very efficient. The "Cravenette" process of showerproofing involves the saturation of the fabrics in especially designed solutions, so that these chemicals become a definite part of the yarn, and render the fabric water-resistant as long as the cloth lasts and it is not subjected to any vigorous cleaning process. In addition to preventing the absorption of water by the fabric, this process also prevents, to a very large degree, the spotting of a cloth when water strikes it.

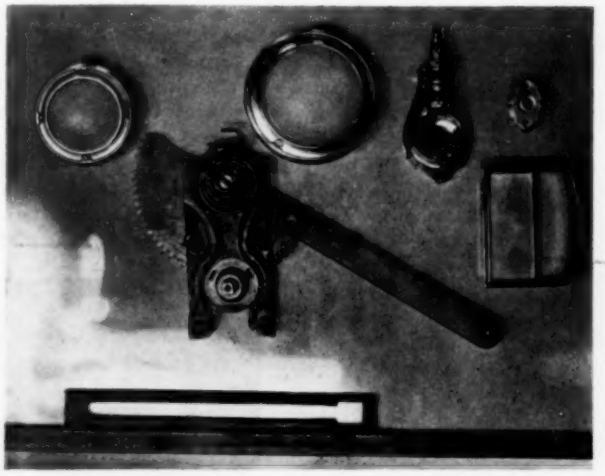


FIG. 10 SOME TYPES OF AIRCRAFT HARDWARE



FIG. 11 SLEEPING ACCOMMODATIONS

CABIN INSULATION

One of the big problems to be met in passenger comfort is cabin insulation. While the major function of cabin insulation is the reduction of noise, important contributing advantages are cold insulation and fire protection. Those insulating materials which are highly inflammable should be barred. While there is a great deal of development work going on in the reduction of noise at the source, the present success in abating the noise nuisance lies in the reduction of the noises that find their way into the airplane cabin. Therefore a good airplane-cabin insulation should have three properties; reduction of sound penetration, absorption of the sound that penetrates, and lightness of weight.

The Bureau of Standards in conducting sound-insulation experiments uses a unit of measurement called a "decibel." In a scale which ranges from 0 to 108 decibels, it places the sound comparable to the rustle of leaves in a gentle breeze at 10 decibels.

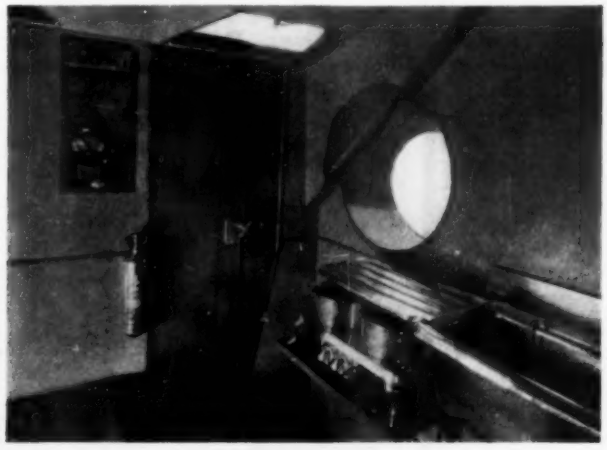


FIG. 12 KITCHEN IN KEYSTONE PATRICIAN



FIG. 13 LAVATORY FINISHED IN LIGHT GREEN FABRIKOID

an average whisper 4 ft. away at 20; conversational-speech range at 35 to 65, train noise at 65, New York subway train noise at 80, and the noise in an open cockpit plane at 90 to 100. By proper insulating methods it has been shown that a decrease in sound level of about 32 decibels can be attained. Generally speaking this is a reduction in the loudness of sound as heard in an open-cockpit plane to a level of that of a railroad coach in motion.

The most efficient insulation material commercially utilized, on the basis of weight for the same degree of sound insulation and volume of space filled, is dry zero. Dry-zero airplane blanket is a batt of ceiba fiber bound up with cheesecloth. The in-

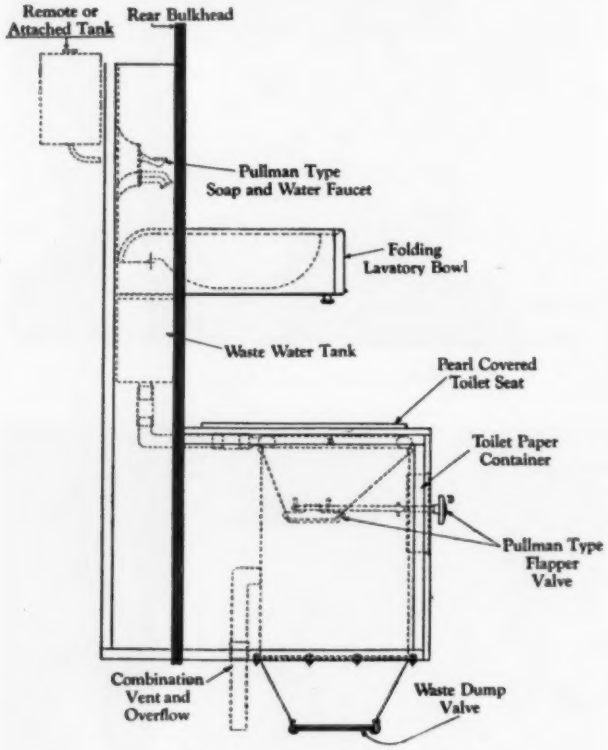


FIG. 14 IMPERIAL CABIN TOILET

tensely fine and permanently resilient tubular fibers of ceiba are "grained," that is, laid parallel and across the line of sound transmission or heat flow. Its resistance to heat flow is 25 per cent greater than that of cork, and it absorbs only 49 per cent as much moisture from humid air as cork. The material weighs about 0.2 lb. per sq. ft. in 2-in. thick layers, which is approximately $1/8$ the weight of cork. The material is highly flameproof.

Balsam wool, the next most popular and efficient insulation, is made of clean new wood fiber from Northern coniferous trees. By mechanical and chemical means the wood is broken down into individual fibers; fine, hair-like, hollow tubes. After being saturated with chemicals to render them non-inflammable and resistant to decay, and treated with an adhesive to secure a permanent bond, these fibers are felted into a fleecy mat which looks and feels like sheep's wool. In balsam-wool airplane insulation this mat is lined on one side with waterproof Kraft paper and on the opposite side with open-mesh cloth. Balsam wool weighs 0.3 lb. per sq. ft. 1 inch thick and 0.18 lb. per sq. ft. $1/2$ in. thick.

Both of these materials are easily installed, being literally tucked into place after being fitted with a simple pair of shears. In wooden fuselages they are nailed to the wooden struts, a braid being placed over the nails to protect the interior fabric. On metal tubular structures the material is wired in place, supplemented by nailing through wood fairing strips where available. It is to be noted that continuous hard materials (such as micarta, plymetal, etc.) when used for interior panels impair the sound-absorption qualities of the insulation, and that loosely woven fabrics are best for interior finish in this regard. It is possible to lower the noise level by 8 or 10 decibels by taking this simple precaution.

Other substances such as hair felt, cotton, and cork have been tested and used as insulation materials, but they are not as efficient as the foregoing. In France, seaweed is used for this purpose, being very light in weight and non-inflammable. It is kept together by fireproofed thin-woven fabric.

Inasmuch as windows are not so effective as the walls in dampening sound in a well-insulated cabin, it is necessary to use rather heavy glass with a thickness of $3/16$ or $1/4$ in. From a safety angle it is highly desirable that this be laminated glass. Briefly, laminated glass consists of two layers of glass cemented together by an inner layer of tough, flexible, colorless material, molded by heat and pressure into one clear transparent plate, which neither jars nor blows can force apart. If the glass breaks, no splinters will fly, all broken particles adhering to the flexible inner layer. Celluloid or pyralin is generally used for the inner layer. Safety glass $1/4$ in. thick weighs about 50 oz. per sq. ft., a relatively high weight, so that it is desirable to hold windows to a minimum size consistent with passenger comfort and enjoyment.



FIG. 15 LAVATORY AND WATER FLUSH TOILET WITH BUCKET CONTAINER

FURNITURE

The cabin airplane chair probably contributes more to the comfort or discomfort of the passenger in a transport plane than any other single item. Early chairs at best were rather crude affairs, very light due to their wicker construction, but otherwise not very practical. The building of seats for any purpose is one that requires considerable study of the human anatomy so that the proper comfort factors may be incorporated. A chair that may be very comfortable when tried out in an office for a few minutes may prove to be very irksome when one is forced to sit in it for any great length of time. As rattan did not adapt itself readily to a strong and permanent connection to a structural member of a hull or fuselage, attention was naturally diverted to metal tubing for the framework. There is, of course, no universal airplane chair for all uses, but fortunately several organizations have specialized in this field, so that there is a reasonable range of choice, and particular designs may be modified in detail dimensions to fit particular planes. Brief descriptions of some of the leading chairs follow.

Adapto Steel Chairs. (Grand Central Wicker Shop.) This self-reclining chair is easily made of mild carbon-steel tubing of 0.035 in. wall thickness, and lends itself readily to airplane re-

quirements. The mechanism is simple, automatic, and light. Whereas a chair frame is usually constructed in one part, this is made in three, i.e., a seat chassis, a seat, and a back. The back pivots from the chassis while the seat is connected at two friction points with the back. A chair of this type in which the restless passenger can satisfy his whims with an infinite number of positions is highly desirable. In choosing the upholstery fill, cotton has been discarded because of its weight. Down has a slight advantage in weight and comfort, but kapok is considerably cheaper and possesses water-resisting properties which are desirable as a life preserver in a flying boat. Fabrikoid is used for covering due to its strength, lightness, cheapness, and washability. Repair costs caused by tears or burns are reduced to a minimum by having each unit of the chair upholstered separately and laced. The complete weight is $7^{3}/_{4}$ lb.

Steel Arm Chair. (Grand Central Wicker Shop.) The steel tubular arm chairs required by the large transport planes such as the Ford tri-motor, merely necessitates a reconstruction of the chassis. The front and rear legs are arched a foot or so higher than on the side chair chassis, thus forming two arms which are upholstered in the same manner as the seat or back. This incorporates all the comforts of a heavy living-room chair and weighs but 13 lb. While no two planes generally call for the same chairs, it is a simple matter to change heights and widths.

Aluminum-Alloy Reclining Chair. (Aluminum Company of America.) This chair is fabricated from strong aluminum alloys using both riveting and welding as fabrication processes. Tubing of 0.036-in. and 0.040-in. wall thickness is used. The total weight of the chair is $12^{1}/_{2}$ lb. These chairs are also available in a double-chair type.

Aluminum Alloy Deck Chair. (J. G. Brill Company.) Both the adjustable and stationary chairs developed are constructed principally of heat-treated, $^{3}/_{4}$-in. aluminum-alloy tubing. To this framework is attached a one-piece leather cushion support and back. The cushion and headrest are padded and upholstered in leather. In the adjustable type there is a push-button directly under the arm at each side, the pressing of both buttons at the same time releasing the back. The stationary type weighs 13 lb. complete and the adjustable type $19^{1}/_{4}$ lb.

Magnesium Alloy Straight Chair. (Dow Chemical Company-Kelton Aurant Company.) This magnesium-alloy tubing chair is fabricated by welding, the use of a light braided fiber as filler bringing the total weight with seats and pads down to approximately 6 lb.

The custom-finished private plane offers unlimited opportunity for the decorator to exercise his skill. Built-in desks at which the business man may dictate his letters while traveling are cleverly contrived to take little space in the plane and are most useful. Complete galleys for the preparation of food on electric equipment are features of some private planes, while all others have one or more sleeping berths, somewhat similar to those in a railroad Pullman car. Combination ice boxes and table cabinets are features of others. In fact, the designers of air yachts have taken many a thought from pleasure boats or Pullman cars, and made them as practical for air as for water or rail travel.

Sanitary Equipment

The development of this very primary need in modern planes



Fig. 16 Fokker Super-Universal Plane With Engine Exhaust Pipe Arranged for Heating

may be divided into two distinct phases of development. The first installation in which all the requisites are installed as separate units as shown in the current planes of the day, and the latest, a brand new development, in which all the requirements are grouped into a single compact unit before installation. In addition to the obvious restrictions of weight and space, there is the further fact that government regulations rigidly prohibit the discharge of any waste while in flight. There were really no precedents available in other travel conveyances when this development was first inaugurated, so that the entire matter of lavatory equipment may be said to have been a pioneering undertaking.

In most of the installations made with separate individual units, it has been necessary to build a great part of the equipment as special, designed to fit the space available. Nickeline and aluminum are used to build wash stands and water tanks. Wash stands are equipped with either two faucets, one faucet, or none at all, some types being foldable when not in use, while the weight ranges from 1 to $5^{1}/_{2}$ lb. Water tanks are of various forms, flat rectangular, triangular to fit corners, quarter cylinder, and half cylinder, according to space available. Varying capacities, often of 4 to 5 quarts, and weights of $1^{3}/_{4}$ to $6^{1}/_{2}$ lb. are utilized.

Aircraft hoppers are of the chemical type, ranging from 16 to $25^{1}/_{2}$ in. in height, with diameters from 10 to 13 in. and weights from $8^{1}/_{2}$ to $19^{1}/_{2}$ lb. Some are of the folding closet type,

while others are built with slanting bottoms to fit the sloping floors found on many planes at their point of installation. The inner buckets are readily removable for cleaning.

The single-unit cabin toilet is a self-contained unit, designed from the three predominating angles of minimum weight, minimum space, and artistic design. It is patterned after the lavatory toilet-chair combination used in the recently introduced bedroom cars of the Pullman company. There are three outstanding advantages to this unit, namely, it is complete within itself, combining water and liquid-soap storage, lavatory, and water-flush toilet in a compact article that can be moved and placed in any desired location; secondly, it occupies only a part of the space now occupied by the conventional type of corner bowl and open toilet; and, thirdly, it is equipped with the Pullman-type flapper valve and water sealed against all odors.

Its pleasing appearance allows the use of the toilet room as a dressing room without the feeling of distaste that would ordinarily prevail. The modernistic design as a whole is carried into detail in the shape of the Venetian mirror, side lights, and toilet seat. The back of the chair (which forms the bottom of the lavatory) is upholstered in rich velour or leather, and the color scheme is followed in the finish of the pearl-covered toilet seat, frame, and covering of the unit. A variety of colors are provided to harmonize with the interior decoration of the planes. The panels may be of perforated metal, solid sheet in tile or painted colors, or of special sheet materials. Complete units, together with 5-gal. capacity tanks, are 14 in. by 22 in. by 63 in. in dimension and weigh about 32 lb.



FIG. 17 EXHAUST FROM THE CENTER MOTOR RUNS UNDER THE FUSELAGE WITH STOVES AND HEATING CYLINDERS

AIRCRAFT HARDWARE

The development of suitable aircraft hardware for cabin interiors is, strange to say, but a recent accomplishment. Its history has been rather peculiar. At first the airplane manufacturer was literally forced to shop at the corner hardware store and accept the weight penalty. Then individual plane manufacturers and a few specialty concerns made crude attempts at light-weight design, entirely lacking in any artistic conceptions. The advent of the decorator into the field necessitated an about face and the penalty of excess weight was again paid by utilizing automobile hardware which was pleasing in design. Only recently have the automobile-hardware manufacturers interested themselves in aircraft hardware, with the result that at last there are available small, artistic, light-weight hardware units, usually made of aluminum or aluminum alloy except where design or wear limitations preclude their use for certain component parts.

To enumerate in detail all the various types of aircraft hardware now available would be rather tiring. Therefore only a very abbreviated attempt will be made to give an indication of the various fitments which are customarily classed under this general term. Door-locking equipment ranges all the way from cylinder door locks with automatic safety-lock features, to simple latches and bolts, with external handles usually streamlined in form. Window equipment includes various forms of window regulators, window stops, and regulator handles, usually, though not always, available in light alloy materials. Aluminum-alloy hinges of all forms are readily procurable. Lighting equipment includes various sizes of dome and torchère light fixtures. Robe rails, baggage racks, ash trays, and various accessories complete the list of special light-weight hardware of artistic design developed especially for airplane interiors.

HEATING AND VENTILATING

Heating and ventilating equipment where installed, while perhaps more exterior in its visible applications, is decidedly interior in its needs and effects. While the necessity for heating is naturally influenced by the routes operated over or the territory ordinarily selected for flying operation, it is decidedly important from the viewpoint of passenger comfort, as extreme cold is reached at high altitudes even in temperate climates. It may be cited as a matter of interest that on the Western transcontinental route of the Boeing system the temperature sometimes varies from 35 deg. below zero to 135 deg. fahr., and the flying altitude from sea level to as high as 12,000 ft. when crossing the mountain ranges.

In heating, the source of heat is the exhaust from the motor.

This is carried from the various cylinders of the motor in a collector ring to the base, where a long steel pipe is brought down under and along the fuselage to carry these hot gases. Stoves or heating cylinders, usually of duralumin, are placed around this exhaust pipe, and direct fresh-heated air up into the passenger cabins through appropriate conduits and floor registers. The heating cylinders are belled out at the front to insure an adequate supply of fresh air, and various types of control valves are used to regulate the amount of heat admitted to the various parts of the cabin.

Ventilation is not such a serious problem in so fast a moving transportation medium. To a great extent it is at present merely a matter of opening and closing windows, but in addition some

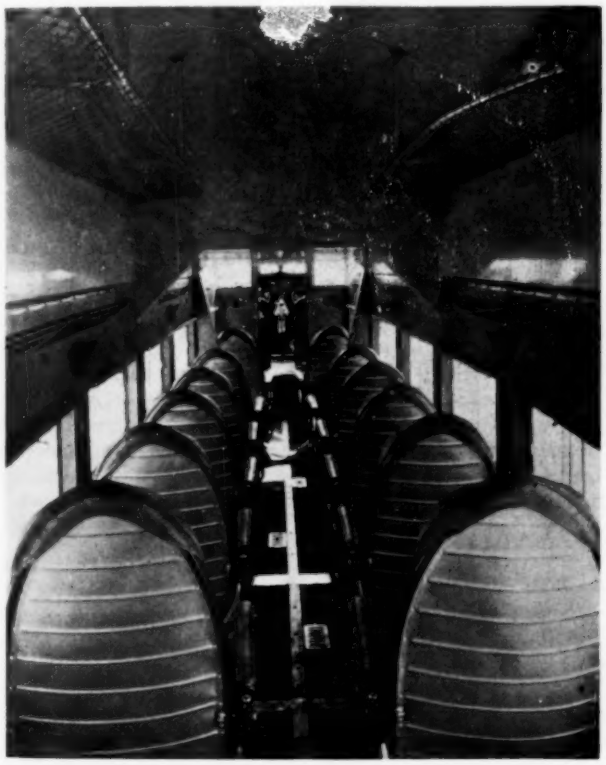


FIG. 18 FORD TRI-MOTOR PLANE '20 WITH FLOOR REGISTERS FOR HEATING

installations use an ordinary light-weight grill-type ventilator with an external flat scoop at certain desirable locations.

RÉSUMÉ

It is hoped that the foregoing will give some conception of the many problems involved under the heading of interior design. Its accomplishment is essentially a problem in coordination of many materials, many parts, and the work of many individuals. This job of coordination belongs most properly to the decorator. His is the responsibility to create in this rather limited space—and it is limited, judged by the standards of other means of transportation—an artistic appeal that makes of flying not only the last word in speed but in luxurious comfort as well. It is necessary not only to keep abreast of style trends, but many times in advance of them, so that their specialized application in aircraft materials and fitments will appear simultaneously with their more readily general application in other industries.

Air transportation is still at the stage where the first flight of an individual is significant; everything about the exterior of a plane is novel to his perceptions, the medium through which he is transported is new to him as such, and the speeds with which it is done are also generally far in excess of what he is accustomed to. Only one familiar aspect can be graciously bestowed upon him, an interior which is as luxurious and comfortable as his own home or club, and which immediately places him at his ease.

To attain this perfection in harmony of line, color, and utility in an industry which economically has not yet felt the advantages of true mass production, is a difficult task. That it has been accomplished in so fitting a manner, as shown by the accompanying illustrations, is a tribute not only to the decorators who have directly created these artistic conceptions of interior design, but also to the many specialty organizations operating primarily outside the field of aeronautics, which have had faith enough in the future of aviation to anticipate the ultimately profitable volume demand for specialized aviation products by making them available well in advance of that eventual demand.

The rapid evolution of aeronautical technical development makes it imperative that the finishing of the interior of any particular plane be accomplished in a minimum of time. There is usually very little time left after the completion of the structure of a plane for the interior design to be carried out. Therefore while this should be executed under the direction of an experienced decorator, who has a knowledge of the mechanical problems involved, the actual work is best done by the personnel of the airplane manufacturer under a specially appointed and experienced foreman. The decoration of the interior of a plane is a specialized function, best performed by those who are thoroughly familiar with plane construction and fully aware of the many intimate special problems involved.

Discussion

JOHN G. LEE.[3] After so admirable a presentation, the writer dislikes to take the opposition view. Nevertheless, he feels that any stress on cabin interior decoration is premature. He has no quarrel with special interiors for individuals who buy an airplane as if it were a yacht. They have a right to anything that makes them comfortable or suits their fancy. But for the ordinary transport plane to be used on a transport line the case is different, as also for the small plane for the private owner, where we cannot afford to make every one different.

The truth is that the airplane is not as far advanced as we like to think. The present-day airplane is about as far developed as the steamboats that immediately followed Fulton. In other words, they are clumsy and unsatisfactory in many ways. To spend much time on the decoration of a Fulton steamboat is obviously ridiculous. When as radical a device as the Autogiro can seriously be considered as a competitor of the airplane in any particular, then the art of airplane design is anything but stable. Or, more specifically, when we do not know whether to make a fuselage of shell construction with rounded sides or to build it of tubular members, then we are not ready for the interior decorator.

The writer would be glad to see all of the interior design thrown overboard. Obviously, however, we cannot do that. Airplanes compete with luxury in other forms of transportation, so we must supply at least the essentials of comfort. We may differentiate between comfort and decoration. This modern decoration is usually overemphasized. The designs are large and bold, and do not show well in a constricted space. One cannot get away from them. They are disquieting, to say the least. What decoration there is should be restful and simple.

[3] Fairchild Airplane Mfg. Corp., Farmingdale, N. Y.

The authors mentioned the use of a "mock-up," or full-size dummy structure, as a help for laying out the cabin interior. This is a most excellent idea. In the last plane which we designed at the Fairchild factory we used this method, and saved ourselves an untold amount of difficulty in the cabin design.

The subject of soundproofing is of great importance. It has always seemed, however, that we have gone at it backwards in permitting so great a source of sound as the power plant, and then trying to prevent that sound from entering the cabin. If we were to put the necessary time and money into research on a silent propeller, the result would be much better. For the propeller is really the most serious offender. The problem should be relatively easy. We know that a geared propeller is much quieter than a direct drive, and that micarta and wooden propellers are quieter than metal. That is something to start with at least.

To silence the exhaust is by no means sufficient. The human ear can record only a certain volume of sound, to put it roughly. If the volume of sound exceeds this amount, enormous reductions can be made and yet the effect will scarcely be noticed. Some time ago we were much interested in mufflers at the Fairchild plant, and finally obtained an excellent silencer. From the ground the noise of the plane was much reduced, but in the cabin there was no improvement. After flying wide open at 5000 ft., we put the plane in a dive and throttled back the motor until we obtained the same high speed and same r.p.m. as in wide-open level flight. There was no difference in noise between the two conditions. In other words, the silencer was effective, but no improvement resulted. In fact, the only comment which shed any light on the subject was the pilot's remark. "This is the noisiest engine I ever rode behind. You can hear the parts rattle around inside it."

We had silenced one of the noises, which allowed others to be heard, but did not sufficiently reduce the general noise level. If we can do something to eliminate these noises at their source, then the decorators' soundproofing will be much more appreciated.

While on the subject of comfort, the writer wishes somebody would define what "adequate comfort" is. We get as many answers as we ask people. For example, are toilets really necessary in present-day aircraft? If so, how big can an airplane be before it requires a toilet? Another point, are the reclining chairs really such an asset? The writer thinks not. They have a nasty habit of collapsing when you put your hand on the back of them in walking up the aisle. Also, when you are seated, and the pilot "gives it the gun" your seat flattens out like an operating table. To prevent this, some sort of clamp or clip is required for the passenger to operate, and most passengers are scared to tamper with any adjustments on an airplane.

It is surprising how much apparatus a human being requires to have carried around with him. The average passenger thinks that we are transporting nothing but himself. Actually we are carrying an accumulation of weights, about as follows: Passenger's baggage, 35 lb.; seat and safety belt, 15 lb.; laminated window glass, 10 lb.; upholstery, rugs, soundproofing, 15 lb.; toilet and toilet compartment, 10 lb.; total, 85 lb. Thus, for every 170-lb. passenger we have to carry 50 per cent more of unpaid weight.

As things stand now, air transport does not pay. Before it can pay we will have to at least double our pay load. That does not mean to double the size of the airplane. It means to double the pay load for the same size of equipment. Furthermore, air transport is unpunctual and unreliable. If the weather is bad, the flights are postponed, Otherwise, they are unsafe. We will have to make air transport a dependable service. When we are confronted with problems as serious as these it seems premature indeed to lay great stress on cabin interior design. Let us keep our interiors simple, and think more than twice before we add anything more for the airplane to carry around.

Airplane-Wing Trussing

By RICHARD M. MOCK,[1] WARNEMÜNDE, GERMANY

The paper compares and analyzes the existing types of wing bracing in conventional airplanes. The internal structure of airplane wings is closely linked with the external system of support, and therefore the paper covers some of the methods of internal bracing, but is mainly devoted to systems of external bracing. The efficiency of the wing trussing is a function of the size of the machine. For a low-wing design the wire-braced wing seems quite efficient, providing the span is not too great compared with the height of the plane. For a larger span, subject to high wing loadings, the high-wing monoplane is very efficient, though there are many points in favor of both the braced and the cantilever wing. For the very large plane, with a normal fuselage, the biplane bracing is advantageous. On large designs where a part of the load can be carried in the wings the monoplane is better aerodynamically.



OF ALL the possible variables in airplane design, the system of bracing for the wings perhaps has been exploited the most. It seems that man's imagination has no limit in designing the trussing of members to support the lifting surfaces of the airplane. This paper will confine itself chiefly to existing types of bracing and will attempt to compare or roughly analyze them. Only the conventional airplanes, as in practical use, will be discussed; the Ente or Canard type, as well as the Autogiro, will not be included.

The internal structure of airplane wings, especially in monoplanes, is so closely linked with the external system of support that it is believed best first to cover some of the current methods of internal bracing and then to devote the remainder of the paper to some of the systems of external bracing.

Internal wing bracing may be divided into two basic schools. In one, the entire air loads on the wing are taken by a bracing inside the covering of the wings, and in which the covering is used only to give shape to the airfoil. In the second, the wing has a covering of rigid material which can be utilized to take stress and therefore to eliminate a corresponding amount of material inside the wing. In addition, there are some designs using a combination of these two.

The simplest type of internally braced wings consists of the popular two-spar design, with compression members parallel to the flight direction. These compression members are between the spars and have diagonal wires or tierods between, completing the truss. This system has been varied by having single diagonal wires in one direction and double wires in the other, depending upon the loads, or by having a complete double system with the double wires attached together on one fitting. A third is to have the wires as far apart as possible, and thus give the wing great torsional rigidity. A variation of the last is to have the wires go from the top of one spar diagonally across the bay to the bottom of the other spar. However, this is somewhat complicated and is not desirable in most wing designs. For the wires it has been proved quite efficient to employ tierods of solid cross-section and threaded terminals. In some less expensive airplanes hard piano wire is used, wrapped at the ends and soldered. Another type of internal interspar bracing used in wings of two-spar design has the entire internal wing bracing of rigid members that can take either tension and compression (such as tubes), and thus eliminate wires completely. In this type of bracing it is usual to have the system made up of similar members, arranged in a Warren truss, with none of them parallel to the flight direction. This usually eliminates the complications of compression members with airfoil shape, and yet does not complicate the method of supporting the aileron hinges should there be an aileron attached to this wing. However, it is sometimes found necessary in wings with tierod bracing because of wing, tanks, etc., to use a diagonal tube in one bay. Of all of the mentioned systems the most modern for two-spar designs seems to be that of having all ribs alike with an internal bracing of tierods spaced far apart and with compression members placed between the ribs so that it is not necessary to give them a special contour.

A basically different system of bracing a wing in torsion is that developed by the Mono-Spar Company, Ltd., of England. The system of bracing is to have a single spar in the wing with tierods from the top and bottom of the spar to tubes or ribs in the chord direction. These diagonal wires, crossing at the spar, take part of the torsional and drag loads. (See Fig. 1.)

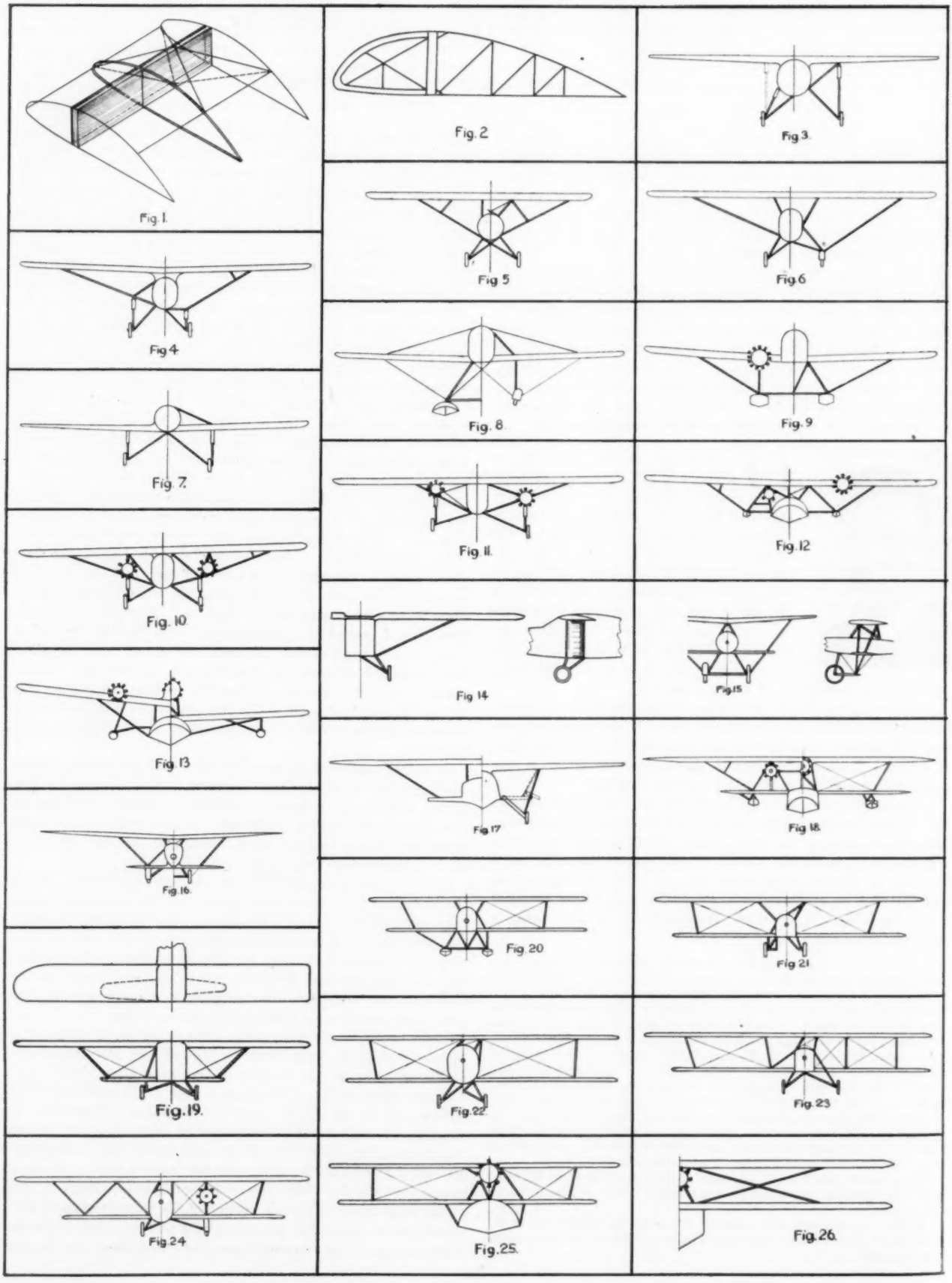
There are many variations of the methods of internally taking the air loads on the wing when it is decided to use the covering to take some of the stress. The prime method is to have a single spar that has strength both in bending and torsion, and can thus carry all of the loads without additional members. The English braced single-spar wing has been used to a limited degree, and in Germany the single-spar incorporating a stressed covering has been used on gliders as well as on power machines.

The German single-spar system utilizes the depth of the wing to take part of the beam loads and the skin to take the torsion and drag loads as well as part of the beam loads. The most common construction of this type is that used in some German gliders and in the B.F.W. powered machines. These have a wing profile with a fairly thick leading edge utilizing a single vertical member about 30 to 35 per cent behind the leading edge and the curved rigid covering over the leading edge to act as a spar. (See Fig. 2.) This thin covering is given its shape by ribs of the usual type. The rear portion of the wing consists of ribs, usually fabric covered to give the proper profile. Sometimes the inner portion of this covering is metal to give a better base for the

[1] (At time of presentation) Consulting Aeronautical Engineer, Ernst Heinkel Flugzeugwerke. (Now assistant to A. H. G. Fokker, New York, N. Y.) Jun. A.S.M.E. Mr. Mock is a graduate of the Daniel Guggenheim School of Aeronautics, College of Engineering, New York University. Before completing his college course he was in the inspection and experimental departments of the Ware Radio Corporation, New York, N. Y., and in the inspection department of the Litho-Etching Corporation, New York. In 1926 he was in the shops of the Loening Aeronautical Engineering Corporation, New York. Later he became technical editor of *Aviation*. In June, 1928, he joined the Bellanca Aircraft Corporation, New Castle, Del. In September, 1929, he went to Germany as consulting aeronautical engineer, supervising the design and construction of a metal flying boat. He returned to the United States this year.

Presented at the Fourth National Aeronautic Meeting, Dayton, Ohio, May 19 to 22, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.


Fig. 1.
Fig. 2
Fig. 3.
Fig. 4
Fig. 5
Fig. 6
Fig. 7.
Fig. 8.
Fig. 9
Fig. 10.
Fig. 11.
Fig. 12
Fig. 13
Fig. 14.
Fig. 15
Fig. 16.
Fig. 17
Fig. 18.
Fig. 19.
Fig. 20
Fig. 21
Fig. 22.
Fig. 23
Fig. 24
Fig. 25.
Fig. 26.

attachment of the wing to the fuselage. Another method, which also has been called a single-spar type of construction, is that employed by Rohrbach, in which the central portion is used as a beam and has vertical members at the place where the usual two spars are located. The top and the bottom are closed in by sheet metal, and thus complete a box to which the leading and trailing edges are hinged.

Wings of the type used by B.F.W. are quite light and allow a very high aspect ratio. (The single spar B.F.W. M-20 has an aspect ratio of 10.) However, it must be admitted that, in their present state of design, wings of this type are not cheap to manufacture in spite of the fact that in principle there are fewer structural elements.

Somewhat similar to these are the wings with two or more internal spars in which the skin takes the drag and torsional loads, and in addition sometimes takes part of the beam loads. The simplest of this type of construction seems to be that of having the spars and usual ribs all covered with a stressed covering, thus forming the airfoil desired. Somewhat more complicated than this is to have more than two spars and a lattice-truss system between the spars, and thus give the wing its shape and also act as part of the bracing. Such wings are usually indeterminate, and are therefore sometimes quite difficult to calculate, as well as being costly to construct.

Stressed covering is normally limited only to cantilever wings, and for this reason the wing is usually tapered so that the aerodynamical lift increases toward the root and thus gives the combination of an aerodynamically efficient wing with sufficient depth at the root to take the large bending moments. However, because of the difficulty of attaching the covering, wings with stressed covering are usually quite expensive, especially if metal is used. Internally braced wings without a stressed covering are, in general, with parallel leading and trailing edges and have no taper in thickness ratio. Therefore they are somewhat cheaper, since all ribs are alike. A simple variation of the stressed covered wings is that used on some Heinkel land types of wooden construction in which there are parallel beams of the normal type supported torsionally by rigid box ribs. A plywood covering for the under side of the wing completes the structure, and the top is covered with fabric in the usual manner.

Before going to the subject of the various methods of external wing bracing, mention must be made of the tendencies in airfoil design, which it seems will materially affect the future systems of internally bracing airplane wings. There seems to be today an increasing tendency in the use of thick profiles having high-lift characteristics and allowing high wing loadings, and also to have the proper depth for sufficient spars. More and more the industry is coming to use airplanes with a short center of pressure travel, thus reducing the torsional loads on the wing. Where a constant center of pressure is not possible because of the aileron loads, when a usual aileron has been employed, it has been proposed to balance this torsional load by another aerodynamic force caused by a second external member, and thus reduce the resulting load to only a local one and a bending load on the beam. However, this has not been worked out to a practical point. In some modern designs it is usual to have inset ailerons and a constant center of pressure airfoil for the wing tip, and thus reduce the possibility of flutter in normal flight with great accelerations or in flight in which certain vertical currents or "bumps" are met.

Cantilever wings usually have a fairly thick section inboard because of the high bending moments. (See Fig. 3, left.) For this reason external bracing has sometimes been used to reduce the bending moments, and consequently to allow a shallower profile to be used. It is common in some designs to have a wing that is quite like a full-cantilever wing in all respects except that about one chord length outboard from the root a strut goes from the fuselage to the wing, thus relieving the bending moment and allowing a more shallow airfoil. (See Fig. 3, right.) Such a wing usually approaches the full-cantilever wing in aerodynamic cleanliness, and sometimes allows a lighter structure, though the aerodynamic interference of the strut and the costs of more members and more fittings sometimes overbalance the advantage of the shallower profile.

A structurally lighter development is the externally braced monoplane wing in which the strut point is some 120 to 135 per cent of the chord from the tip, thus allowing a very light type of structure, providing that the base of the external support has sufficient depth to allow a good angle between the strut and the wing and thus not cause excessive axial loads in the wing beams. Structurally, the lightest variation of this type is to have the wing taper according to the bending moments, and thus give a wing with maximum chord and thickness at the strut point and with a short chord and thin section at the fuselage, giving a low weight and increased vision without the complication of cutouts or abnormal wing tapers.

Mention should be made of the wing system of the German Rohrbach Roland and the Beardmore Inflexible, also of Rohrbach design. In these a very high aspect ratio has been incorporated, and a tension cable is used to support the single-spar wing from the fuselage, and thus have a braced wing for normal wing loadings. Because of the short center of pressure movement, this single cable puts little torsion in the wing.

The most popular type of wing bracing in America and now also finding great favor in Europe is that of the high-wing monoplane with struts from the bottom of the fuselage attached to the wings at an efficient point. (See Fig. 4.) This point varies with the depth of the fuselage, the dihedral and the pressure distribution on the tip, and the depth of the spars available. Bracing of this type has the great advantage that, without being excessively heavy, it allows a wing without taper, eliminating the costs of the dissimilar ribs, tapered spars, etc.

Of all of the possible types of wing bracing it is believed that there are more basic variations in the externally braced monoplane than in any other system. Most airplanes using this bracing are of the high-wing type as they allow the external bracing to be under tension under the most severe flight conditions. In addition, the external bracing is under the wing, allowing less aerodynamic interference than when the wing is low and the external bracing is above, with the bracing members of large dimensions because of their high compression loads. (See Fig. 7, right.) In most monoplanes of the braced high-wing type the members go from the lowest point on the fuselage to the wing. As the axial load in the wing beams is a function of the cosine of the angle between these wing struts and the wing, it is desirable to have this angle as large as possible, and yet have the strut attachment point far enough outward to relieve efficiently the bending moment caused by the air load. Outboard of this strut point the wing is designed in the same manner as a cantilever wing, and for this reason, if costs permit, it is advantageous to have the wing tapered from here to the wing tip. The design of the wing spars between the strut point and the fuselage or cabane is a function of the axial load caused by the external bracing, the air loads producing bending, the length of the bay, and the available section for spars. Thus it can be seen that, in addition to having the angle between the struts and the wing as large as possible, it is advantageous to have the length of the compression bay as short as possible. This has been carried out very nicely in the wing system of the new Boeing monoplane fighter and the French Wilbault 130 by having the cabane struts arranged in such a manner as to reduce the slenderness ratio for the inner bay of the spars. (In Fig. 5, left, is the Wilbault 130 and right,

the Boeing monoplane system.) It is also quite customary now to reduce the buckling length of the external wing bracing by having short struts between this bracing on the wing such as is shown in many of the accompanying sketches. (For example, see Fig. 3, right, the system of the Bellanca "Pacemaker.") In order to reduce the axial loads in the wing as well as the loads in the external bracing, it is advantageous to attach the external bracing struts as low as possible. This is carried out very ingeniously in the French Ateliers des Mureaux type 4 C-2, two-seater fighter (see Fig. 6), where the wings are braced directly to the landing gear. In this plane the wing-strut angle is exceptionally good, though it must be admitted that the landing-gear design is somewhat complicated and there is a great aerodynamic interference between the numerous struts supporting the landing gear to the fuselage.

In most of the American high-wing monoplanes of the cabin type in which the wings are braced externally, the landing-gear shock absorbers are attached to the wing struts, and sometimes an additional strut goes from this attachment point to the upper longerons of the fuselage. (See Fig. 4.)

This action allows the slenderness ratio of the external bracing to be somewhat reduced. In most American commercial designs the external bracing consists of steel tubes, though in many high-speed planes tension streamline tierods have been substituted and additional tierods braced above the wing take the reversed loads. Of course this means that the wing must be below the fuselage in order to have a high point of attachment for the tierods above the wing surface. In the Schneider Cup designs (the wing system of which is shown in Fig. 8, left) and the Travel Air type S racing plane (Fig. 8, right) this has been carried out quite successfully. It is the opinion of the author that this type of bracing has great possibilities as it allows a very light structure without the additional resistance of the usual struts. In both of the racing designs referred to, a very efficient angle for the under tierods was obtained by having them attached to the landing gear. However, for very rough usage, tierod bracing has the disadvantage of being somewhat flexible, and for this reason the Heinkel single-engined and multi-engined low-wing designs (for seaplane designs for heavy sea duty) have substituted tubes for the tierods and thus have eliminated the upper wires. (See Fig. 9.) In these designs the external strut bracing is afforded at a very good angle because of the wide tread of the floats. This type of bracing is also carried out on the Consolidated flying boat (Fig. 12, left) and the French three-engine Paulhan-Pillard E-5 flying boat (Fig. 12, right).

Another method of reducing the slenderness ratio of the external bracing efficiently is shown in Fig. 13, where the additional bracing also carries part of the wing-float loads. In multi-engined, externally braced monoplanes the inner bay of the wing spars and the external bracing can be relieved of great loads by utilizing the additional struts for the engine support as part of the wing bracing and thus reduce the slenderness ratio. (See Figs. 10 and 11.)

In all of the drawings mentioned only the front bay has been shown, and in most cases the airplane is shown as a land machine. However, most of these designs are adaptable for seaplane installation by attaching floats to the landing-gear fittings. However, though the objection to attaching the wing bracing to the landing gear has been somewhat overruled by the use of very good shock absorbers, care must be taken to have sufficient rigidity when including the wings in a float-bracing system.

In externally braced monoplane wings with parallel spars it is simplest to have the external bracing, as seen from above, coincide with the spar center lines and thus relieve the wings of any chord component forces such as may occur when the external bracing is carried to one point as on some folding-wing planes or on some high-wing flying boats. However, in these cases it is believed that the advantage of easily folding wings, allowed by having the struts at one point of the fuselage, or the reduced water resistance, in case of a high-wing flying boat, overbalances the additional weight caused by chord-component forces from the external diagonal wing strut. On the other hand, there is the advantage of great rigidity in having the two lower attachment points of this external bracing far apart longitudinally and thus give as great a base as possible to take the torsional drag loads in the wing. However, in such designs it is usually necessary to have diagonal bracing between the wing-support struts, which then makes the wing structure indeterminate in the chord direction, complicating the stress analysis considerably and increasing the costs accordingly.

In conventional monoplanes the external bracing is usually of streamline form. However, many manufacturers are now adapting the principle developed by Bellanca of having the external bracing with a lift or airfoil section, thus carrying some of its own weight without any great difference in resistance. Streamlining of wing-brace struts, when there are no short auxiliary members between the external bracing and the wing (such as mentioned and as illustrated in Figs. 4 and 5), means that the large section for the external bracing members must continue to the wing and thus cause a large amount of aerodynamic interference, especially if the strut is given a lift section. However, by the use of these short auxiliary struts, the large cross-section for the external bracing does not continue to the wing and allows the airfoil section to be carried to this auxiliary strut attachment point and then have a small streamline section continued to the wing. Thus induced drag is reduced. While on this subject it might be interesting to recall the wing bracing of the Bellanca CF monoplane (Fig. 14) developed in 1922 in which the external bracing supports formed the spars for a lower auxiliary wing that braced the main wing to the fuselage. Although the aerodynamic interference of this auxiliary or lower wing was very high, this was somewhat compensated by the increased lift and lateral stability of the plane. Incidentally, it is the opinion of the author that the long column members for external bracing for monoplanes for land use only can be made structurally more efficient in wood than with steel tubing. This is because of the form factor of the wood column and the possibility of better attaching the ribs for the fairing and covering the entire unit with doped fabric. However, it must be admitted that the steel-tube type of bracing with a metal fairing, though somewhat heavier, is cheaper.

Similar to the externally braced monoplane is the wing-bracing system used on the Nieuwport-Delage planes, shown in Fig. 15. In this design the landing gear is of a through-axle type with the fairing for the axle having a lift section. From the rear of this fairing goes a forked strut to the upper wing. The strut is supported laterally in the middle by a short wing from the bottom of the fuselage. Both aerodynamically and structurally, this type of design having a large upper wing and a very small lower wing is quite small, the weight of the external bracing is not excessive and is properly proportioned, and in addition it allows more lifting area. However, in the method of carrying this out in the Nieuwport-Delage machine there is a considerable number of struts below the fuselage, which it is believed causes much induced drag. Somewhat similar to this in structure, though of cleaner design, is the French Breguet sesquiplane (Fig. 16, right), in which more of the usual biplane characteristics are incorporated, with the power wing strut point braced to the landing gear by wires. The upper wing has considerably more span than the lower wing, and for this reason the interplane strut, a single streamline member, slopes outward so as to eliminate excessive overhang in the upper wing. In the French Caudron sesquiplane (Fig. 16, left) the interplane bracing consists of stream-

line N-struts attached to the lower wing above the landing-gear attachment points and then has additional struts from this landing-gear attachment point to the upper fuselage longerons. Thus the struts act as part of the landing-gear structure as well as supporting the wings.

The idea of having the lower wing of a sesquiplane assist in taking the landing and starting load was also carried into the flying boat by Dornier, who has a small stub wing at each side of his main boat hull. (Fig. 17, left.) In addition to bracing the upper wing this stub wing provides lateral stability on the water and gives sufficient planning action for starting. In the first Fokker F-XI amphibion (Fig. 17, right), the sesquiplane idea or stub-sponson idea was also incorporated, providing for the amphibion gear as well as to be used for starting. However, in this design it is understood that the upper wing supported the lower, and not the lower wing the upper, as is usual. Fig. 18 shows two methods of utilizing the lower wing of a sesquiplane in supplying a convenient support for wing floats on the flying boat, whereas quite long struts might have to be used for the side floats if the



FIG. 27 HEINKEL HE-9, HOLDER OF FIVE WORLD'S RECORDS

plane was designed only as a high-wing monoplane. The system shown on the left in Fig. 18 is similar to that used by Sikorsky. In the Buhl sesquiplanes the lower wing has such great taper that it might be said to be triangular in form. In the larger of the Buhl types (shown at the right of Fig. 19) the landing gear is supported to the wing in a method similar to that in the Caudron.

Two designs also incorporating the landing gear with the wing bracing are shown in Fig. 20. Both of these planes are Heinkel single-engine water training machines in which the landing loads are carried through parts of the wing structure. In the design shown at the left no tierods are used.

For the land types of purely braced-wing biplanes there does not seem to be as much variation in the system of trussing the wings except perhaps in the arrangement of the struts for the cabane. Figs. 21 and 22 show two of the most popular methods of carrying this out. In a biplane, it is agreed that in a multi-bay design it is possible to have some reduction in weight, though the complication and aerodynamic resistance are, of course, greater. The Martin 74 type of bracing (Fig. 21, left) eliminates the additional bracing of the two-bay design and yet incorporates much of its efficiency and at the same time provides for a landing gear of fairly wide tread. The Boeing mailplane 40-C (Fig. 23, left) is practically of two-bay design with the inner bay braced with tubes instead of wires. A variation in the usual tube- or wire-bracing biplane design is in the Parnall Elf (Fig. 24, left) in which a complete tube truss is used to provide greater rigidity and allow the wings to be folded. In multi-engined biplanes it is usual to have the outboard engines form part of the interplane bracing, and thus have a multi-bay design, utilizing the engine bracing as an integral part. This is carried out in the Boeing transport (Fig. 24, right). Fig. 25, left, shows the usual method of bracing a biplane flying boat, while Fig. 25, right, is a type of bracing used on some German seaplanes in the war.

Perhaps one of the most interesting types of wing bracing developed in recent years is that first used in the Bellanca K and later developed in the Bellanca TES. (See Fig. 26.) It is difficult to classify this type of wing in the category of either biplane or sesquiplane, as the wing bracing in no way resembles the sesquiplane as the term is generally used. In endeavoring to incorporate a large span and yet have monoplane efficiency, Mr. Bellanca has used what might be called a two-bay design, with



FIG. 28 BELLANCA TES



FIG. 29 BELLANCA K

the inner bay being a normal biplane with usual dihedral and the outer bay being a braced-wing monoplane, incorporating the advantages of the Bellanca CF and yet eliminating the aerodynamical interference by having the supporting wing stub some distance below the upper wing. In order to provide an angle as efficient as possible for the bracing wing, or auxiliary wing, the upper wing of the inner bay is given some positive dihedral, and at the same time the lower wing of the inner bay is given negative dihedral. Thus a deep box for the external wing bracing is obtained, at the same time providing for support of the landing gear.

From the foregoing it may be seen that the efficiency of the wing bracing is a function of the size of the machine. For a low-wing design the wire-braced wing seems quite efficient, both aerodynamically and structurally, providing the span is not too great when compared with the height of the plane. For a larger span, subjected to high wing loadings, the high-wing monoplane is very efficient, though there are many points in favor of both the braced and the cantilever wing. Also in airplanes of about this size and for flying boats with side floats the sesquiplane

type of bracing offers many advantages structurally, while when one considers the very large plane, with a normal fuselage, the biplane bracing is advantageous. However, on large designs where a part of the load can be carried in the wings the monoplane is distinctly better aerodynamically.

Discussion

W. E. Savage.[2] The paper discusses almost all the externally braced wing trussings used at the present time, but very little was mentioned of the internally braced wings. It is believed that "interference," which accompanies externally braced wings, should have been included in this paper, since interference must be carefully considered in developing wing-trussing design.

The paper, which is not highly technical in character, will serve as a ready reference for the different type of wing trussings used to date.

[2] Aeronautical Engineer, Wright Field, Dayton, Ohio.

AER-53-16

Design and Development of Seaplanes for Transatlantic Service

By DR. E. RUMPLER,[1] BERLIN, GERMANY

The author takes up in succession the various problems involved in the design and construction of a multi-engined airplane capable of making a transoceanic non-stop flight of 5000 miles and carrying a pay load of 21 tons. Among the topics dealt with are: Location of engines and propellers; influence of load distribution on weight of wing structure; aerodynamical researches of the Göttingen laboratory on a model of the author's plane; hydrodynamical starting tests in tank at the Hamburg Shipbuilding Research Institute; influence of various factors on size of pay load; maximum power outputs from oversize engines at various altitudes; necessity of greatly increased airplane speeds; etc.



IN DEALING with the subject of long-distance airplanes it will only be possible, because of space limitations, to discuss a few of the problems involved. I shall confine myself, primarily, to long-distance water planes, but much of what I shall say relative to these is also applicable in a corresponding sense to land machines. I shall also try to make clear what I say by reference to practical examples.

In order to have a starting point, I shall base what I have to say on the design of a transoceanic airplane, especially so because tests carried through in the research institutes of Göttingen and Hamburg pertained to such a machine.

Location of Engines and Propellers

In the design of my transocean flying boat, the first problem that presented itself was to find the most favorable location for the engine with respect to accessibility, the lowest air resistance, and the greatest security. All three of these requirements point toward locating the engine entirely within the wing.

In the interest of safety, it is also generally understood that the power plant should be subdivided into several small units in order that equilibrium shall be affected to a minimum extent in the event of one unit's failing, and which would be of little influence relative to the continuation of the flight.

The opinion occasionally stated that several power units cannot be started simultaneously or kept at a uniform engine speed is erroneous, being disproved by the large Dornier flying boat, which has twelve engines.

By arranging the engines within the wing structure, the question as to the adoption of air cooling is decided in the negative. Therefore we must use the well-tried and reliable water-cooling system in long-distance airplanes until adequate means are found in the shape of a carefully designed system of tubes that conduct the flying wind into the interior of the wing to the cylinders of the motors.

The question as to the maximum propeller efficiency must be settled by taking into consideration the disturbing influences set up by other constructional parts. Very much depends on the ratio of the propeller diameter to the thickness of the wing. If propellers of relatively large diameter were to be employed, the question of their location either in front or to the rear of the wing would—in view of the experience we now have—not be highly important if only the propeller efficiency had to be regarded; but when propellers having a relatively small diameter in relation to the depth of the wing are to be employed, the location is more of a problem. Owing to the subdivision of the power plant the latter is frequently the case at the present time and will be increasingly so in the future. By locating the subdivided propellers at the trailing edge of the wing, the air current surrounding them as well as the fore-and-aft propeller currents, will suffer relatively the least deviation. For this reason the best reciprocal effect between the wing polar curve and propeller efficiency is to be expected.

The design of the tailless airplane that was successfully demonstrated last year at the Tempelhof airport was based on similar concepts.

As we see from this, the most favorable location of the engine clashes somewhat with that of the propeller, and therefore it may be said, in general, that the employment of special transmission mechanisms becomes unavoidable. This is a problem in itself, and it has to be solved in a manner insuring a minimum of mechanical-efficiency losses, as such losses in the transmission—as will become evident later on—cannot be countenanced in a long-distance airplane. For this reason I shall give no attention to arrangements other than those in which the engines are inside the wing, and the propellers are located above or below them.

In view of what has just been said, it would appear that the best compromise is to place the propellers behind the wings in line with the engine crankshafts and connected to the latter by means of a simple shaft. In the event that a reduction gear is employed, the shaft will of course lie in the direction of the gearwheel on the countershaft. Such an arrangement is simple and its mechanical efficiency will be the highest possible, as additional friction losses are avoided.

Also, the noise problem must be given attention. In the near future this is sure to be regarded as highly important, perhaps no less so than the arrangement of the motors and propellers. It is indeed unreasonable that passengers in a flight of about twenty hours' duration should be subjected to the incessant thundering of the engine. Doubtless many having thus been inconvenienced would desist from using an airplane on a long journey again.

The noise problem can be solved only by locating the engine a sufficient distance away from the passengers and arranging between them soundproofing material. The best possible solution that occurs to me is to locate the engines and propellers behind the passengers.

In a lecture which I delivered in 1926 at the annual meeting of the Scientific Aeronautical Society of Germany at Düsseldorf, I

[1] Rumpler Motor Company.

Presented at the National Aeronautic Meeting, Dayton, Ohio, May 19 to 22, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

outlined my plan for a transatlantic service, the crossing to be effected in stages. Since then, however, the requirements of a transatlantic service as also the technical possibilities have changed somewhat.

5000-Mile Non-Stop Flight a Necessary Design Condition

In order to take full advantage of the great speed of aircraft it will be advisable to make the transatlantic flight in a single stage from a European to an American port. Another circumstance necessitating the crossing in a single stage is to be found in recent politico-economical developments. Owing to concessions which the Portuguese Government has recently granted to an air-traffic company, it will, for some time to come, be impossible to

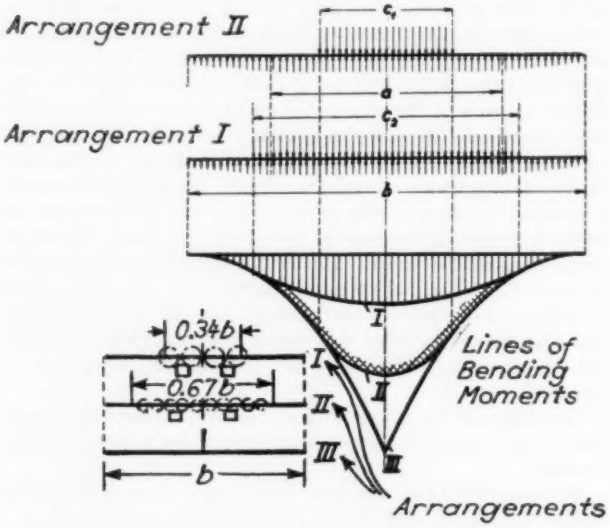


Fig. 1 Curves of Bending Moment

effect intermediate landings on the Atlantic islands under Portuguese sovereignty, namely, the Azores and Cape Verde Islands.

The shortest distance between England and Newfoundland is approximately 2360 miles. But the full advantage of air traffic will only then be attained when it becomes possible to link up directly the most important commercial places. The distance from an English port, for instance, Plymouth to New York, amounts to approximately 3350 miles, and from Hamburg to New York to about 4100 miles.

In practical air service one must always consider the possibility of contrary winds retarding flight and thereby virtually increasing the distance to be covered. Assuming that such a counter wind of, say, 26 miles per hour, is blowing during the *whole* of the crossing—which is a very unfavorable supposition to make—the hourly speed of the plane will then be less by this amount. Later on I shall demonstrate that in consideration of such possible counter winds it is desirable to choose a plane that develops a speed as high as possible. This will not only mean a shortening of the flying time, but actually also a decrease in the amount of fuel to be carried.

An airplane capable of a speed of 144 m.p.h. and heading against a wind of 26 m.p.h. will actually cover only 118 m.p.h. Taking a distance of 4100 miles and dividing it by this speed, we find the plane will have to keep in the air for 34.4 hours. In other words, the airplane has to carry sufficient fuel for a flight of this duration. In calm air this airplane would be able to cover 4950 miles with the same supply of fuel.

In consideration of these facts, I have so designed my flying boat as to enable it to make a non-stop flight of approximately 5000 miles.

Influence of Distribution on Weight of Wings

I should now like to say a few words relative to the influence of weight or load distribution on the constructional weight of wings.

Fig. 1 shows the various possibilities of distributing the weights or loads. In arrangement I the weight is distributed over 0.67 of the wing span and in arrangement II over 0.34 of the span, while in the third arrangement the total load is concentrated at a single point. It is assumed that the span and the weight in all three instances are equal.

The air forces, the sum of which of course is equal to the total weight, it is assumed, are equally distributed over the central part of the wing, while toward the wing tips it is likewise assumed that the load per unit of wing length imposed by the air forces progressively decreases in accordance with the German building regulations.

The diagram shows the curves of bending moment for the three arrangements. The curves meeting at a point represent the moment produced solely by the air forces. The same curves apply of course also to arrangement III. In a long-distance air-

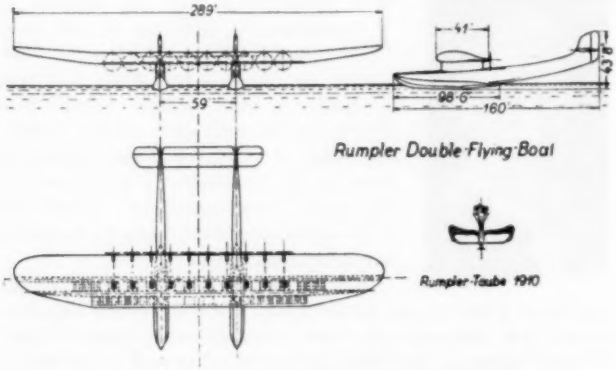


Fig. 2 Rumpler Double Transoceanic Flying Boat

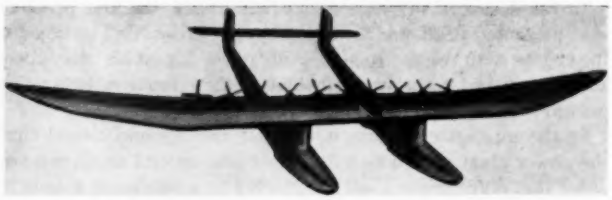


Fig. 3 Rumpler Double Transoceanic Flying Boat

plane such an arrangement will hardly be considered advisable today.

Curve I constitutes the moment of forces for arrangement I and curve II for the second arrangement, and it is seen that by just about doubling the length over which the loads are distributed an extraordinary reduction of the moments is obtained. The maximum moment for arrangement I is about four-tenths of that in arrangement II and about a quarter of that in III.

The weight of the wing, which in the first place depends on the bending forces, will in arrangement I be very noticeably smaller than in arrangement II, and much more so than in III. A rough estimate already shows that in the first plane, according to arrangement I, a saving in weight is effected of approximately 20 per cent, that is, about 11,000 lb. over that of arrangement II and of 30 per cent or 19,800 lb. over III.

This diagram conclusively demonstrates that in the case of long-distance airplanes, which are expected to carry a maximum load a maximum distance, an extensive distribution of loads similar to arrangement I should be selected as this insures the smallest constructional weight, and therefore the largest paying load.

Assuming that the other important features of my transoceanic airplane, which is shown in Figs. 2 and 3, are sufficiently well known, I shall confine myself to saying that my design with two hulls is attracting increased attention. Ten independent engines of the high-compression type, each developing normally 1000 hp., are to be employed. The most important data of the plane are given in Table 1.

TABLE 1 DATA ON RUMPLER TRANSATLANTIC SEAPLANE

Specifications

Total weight	232.000 lb.
Pay load	42.300 lb.
Fuel load	75.000 lb.
Wing area	9,850 sq. ft.
Span	289 ft.
Overall length	160 ft.
Overall height	43 ft. 3 in.
Wing chord	41 ft.
Diameter of propellers	15 ft. 9 in.
Lower edge of propellers above water	13 ft. 2 in.
Thickness of wing	8 ft. 7 in.
Inside height	7 ft. 3 in.
Ratio of chord to span	1 : 7.7
Specific fuel consumption	0.352 lb. per hp-hr.
Propeller efficiency	0.8
Flying altitude	13,100 ft.
Gliding figure	15.1
Gliding angle	3 deg. 45 min.

Constructional Figures

	Beginning of flight	End of flight
Wing load	23.6 lb. per sq. ft.	15.9 lb. per sq. ft.
Power load	23.2 lb. per hp.	15.7 lb. per hp.
Ratio of power to area	1.01 hp. per sq. ft.	1.01 hp. per sq. ft

Performances

Maximum Speed:	
Beginning of flight	183 m.p.h.
End of flight	187 m.p.h.
Average	185 m.p.h.
Average touring speed	144 m.p.h.
Radius of action	4.970 miles
Landing Speed:	
Beginning of flight	79.6 m.p.h.
End of flight	66.5 m.p.h.
Propeller speed	900 m.p.h.
Torsional vibration starts at	2160 r.p.m.
Flexional vibration starts at	290 r.p.m.

The fuel consumption I have taken at 0.352 lb. (0.16 kg.) per hp-hr. as I am hoping to be able to employ Diesel engines. Although perhaps so low a consumption has hitherto not been attained, I believe the research work carried through to develop suitable Diesel engines for aircraft will eventually lead to a complete success.

Apart from low fuel consumption, Diesel engines have the invaluable advantage over normal gasoline engines that they offer positive safety against fire and have great reliability.

The polar curve on which this calculation is based is shown in Fig. 4. It has been developed from measurements made in the research institute at Göttingen and is based on my own experience extending over many years.

Aerodynamical Researches in Model of Author's Flying Boat

Let us now turn to the aerodynamical researches which are being carried on with a model of my transoceanic plane. This investigation to find the most suitable construction for a long-distance airplane has already led to good success.

The basic requirement is to obtain as high as possible a value for

$$\left[\frac{c_a}{c_w}\,\eta\right]_{max.}$$

One of the most important auxiliary conditions is to reduce the relatively high starting and landing speed which is given by the necessarily high wing load. This is obtained by a high $c_{a\ max.}$-value. For starting, the additional requirement of a high propeller force during the starting process and especially toward its end, is to be considered.

While a large $c_{a\,max.}$-value is easily obtained by a corresponding choice of the wing section, the requirement of a maximum degree of efficiency during the start as also of a large $\left[\frac{c_a}{c_w}\,\eta\right]_{max.}$ value offers various difficulties.

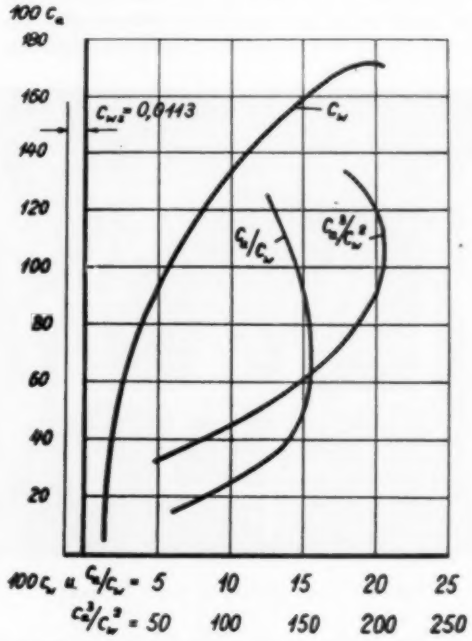


Fig. 4 Polar Curve Assumed for Performance Calculations

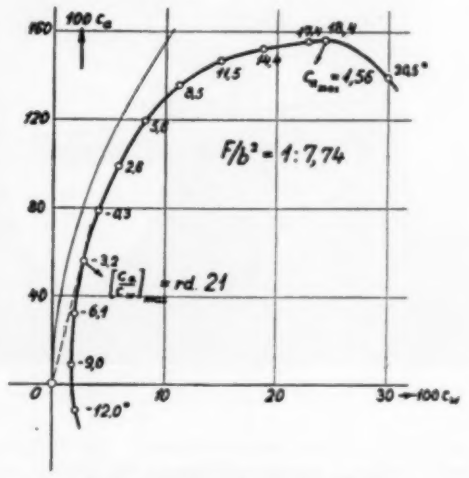


Fig. 5 Polar Curve of Wing

In order to satisfy the last requirement the body resistance and the induced and section resistances must be as small as possible, and the propeller efficiency at the economical speed as large as possible. Also the highest values of c_a/c_w and of η must be reached at the same ratio of velocity to the peripheral speed of the propeller. The requirement of large propeller push forces at starting calls for an arrangement of the propellers in which the efficiency during the low starting speeds shall be as large as pos-

sible. But, as is well known, it is very difficult to obtain an arrangement of this kind which is satisfactory at low as well as at higher speeds.

At this point I should also like to mention various matters respecting propeller efficiency. As is well known, it is not proper to state the efficiency of a propeller by itself, since when a propeller is mounted on an aircraft, each is influenced by the other. The push and torque of a propeller depend on its location on the

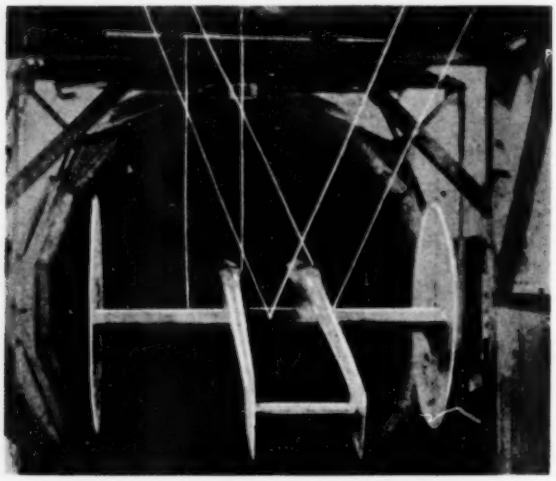


FIG. 6 PARTS MODEL IN WIND TUNNEL

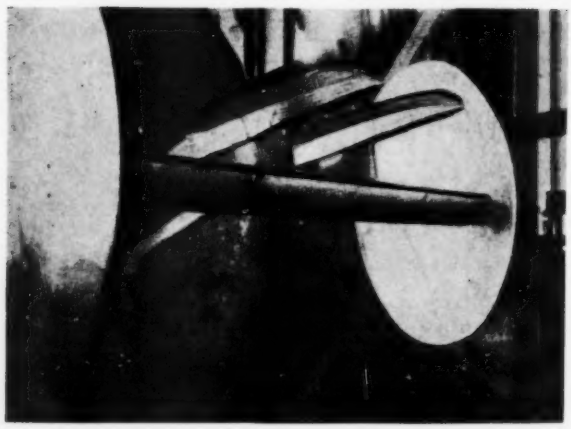


FIG. 7 PARTS MODEL IN WIND TUNNEL

aircraft, and therefore also does its efficiency. On the other hand the propeller causes a change as regards the lifting properties and resistance of a wing as compared with a wing having no propeller. By itself the degree of efficiency established from the push and torque of a propeller on an aircraft does not indicate the quality of the propeller, for the changes in respect to lifting force and resistance must also form part of the calculation. Later on I shall describe how, under the conditions obtaining in economical flight, these difficulties can be avoided. For the other matters I would refer to the interesting work of Beetz and Seiferth.

In conformance with these considerations an agreement was made with the Institute at Göttingen to carry on researches on the wing by itself, the control surfaces alone, various propellers by themselves, a part model with the propeller running, and on the complete model. As it would take too long to report on this work, which has not yet been completed, I shall limit my remarks to the results obtained with the wing and on the part model with propeller running.

Fig. 5 shows the wing polar curve, which is very satisfactory both in regard to $c_{a\ max.}$ and $[c_a/c_w]_{max.}$

Unfortunately it was not possible at Göttingen to carry through tests with a complete model wing with propellers running as the size of the wind tunnel there limits the size of the models. Even if it were possible to test a very large-scale model with propellers, the sizes of the latter and the values obtained with them would still be so small that the results could not be regarded as reliable. For this reason the tests were made with a large-scale part model and propeller. Figs. 6–8 show this model hanging in the Göttingen wind tunnel. The model consisted of the central wing section provided at each end with an elliptical disk. Inside the wing two small electric motors belonging to the Aerodynamical Research Institute of Göttingen were provided, each driving one propeller. Two hulls were fitted to the wing section. They were each provided with rudders and common stabilizer and elevator.

Fig. 9 shows polar curves of the part model which served as the basis for the propeller tests. The one indicated by open circles refers to an arrangement in which the disks sit square on the wind ends and without filleting. At $c_a = 0.8$ it begins to be irregular. The other polar, however, showing the result obtained by filleting the joints between the wing and disks is uniform in curvature, and also much more favorable in other respects. Thus clearly demonstrating the great importance of avoiding sharp corners.

In the tests with the part model about to be described the influence of alterations to the height of the propeller position, the rearward position of the propeller, the propeller inclination, and the propeller diameter is being or has been established.

Fig. 10 shows some of the results obtained in these tests on a part model built to a scale of 1:30, in which the propeller was

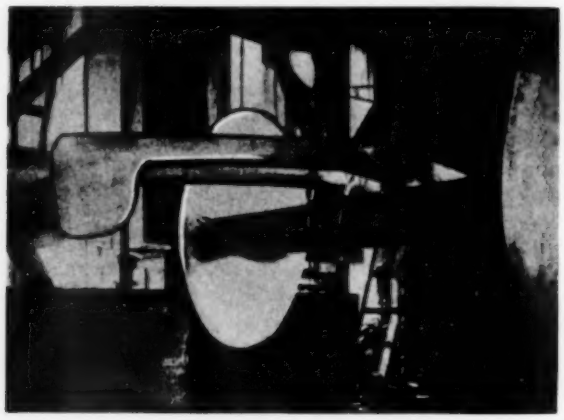


FIG. 8 PARTS MODEL IN WIND TUNNEL

located 5 mm. to the rear, the shaft was 10 mm. above rear edge of the wing, and the propeller had an angle of inclination of 1.8 deg. to the wing chord with the wing set at 8.8 deg. Similar results have been obtained with all other angles of incidence and with all other propeller sizes that were examined.

The abscissas are the ratio of forward velocity to the peripheral speed of the propeller. As ordinates are used the c_a, c_w, and c_m values of the wing area of the part model, also the k_s and k_d values in relation to the circular surface swept by the propeller, and also the η values.

The c_w values here are the resistance figures of the wing with the

propeller running, and therefore do not depict the real resistance of the part model but the reduced resistance caused by the propeller rotation. For this reason the value may be a negative one. At the intersection of the c_w curve with the axis of abscissas the resistance is equal to the propeller pull.

The polars shown in Fig. 11 cover the results of these tests with running propellers. Each polar is based on a definite progression.

By developing this further we arrive at the diagram of Fig. 12, the most important curve of which gives the $(c_a/c_w)\eta$ values for horizontal flight. The peak of this curve corresponds to the most economical flying condition with the propeller arrangement under examination and has a value of 10.2 at $\alpha = -1$ deg.

In a similar manner the values obtained with various positions of the propeller are established and in this way that position of the propeller is being determined for which the $(c_a/c_w)\eta$ curve has the greatest maximum.

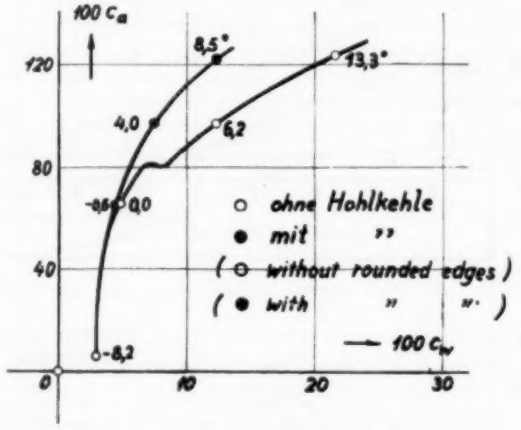


Fig. 9 Polar Curve of Parts Model

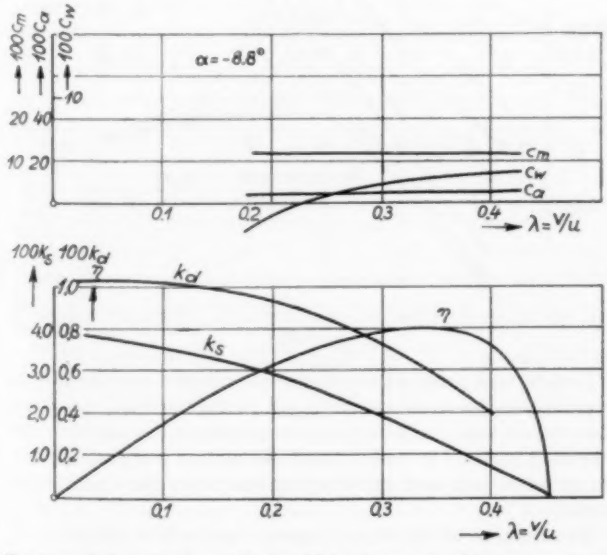


Fig. 10 Parts Model Tests With Running Propellers

These researches are not yet complete and new series of experiments employing a larger-scale model and larger propellers are in progress. The values thus obtained will be greater and the experimental results more reliable.

The results obtained thus far clearly show that it is quite possible to develop by means of systematic experiments a long-distance airplane that will fulfil the aerodynamical demands mentioned at the outset and therefore providing the best solution of the problem in hand.

Hydrodynamic Starting Tests

I shall now discuss some hydrodynamic starting tests that have been carried out by the Shipbuilding Research Institute of Hamburg on a mutually agreed schedule.

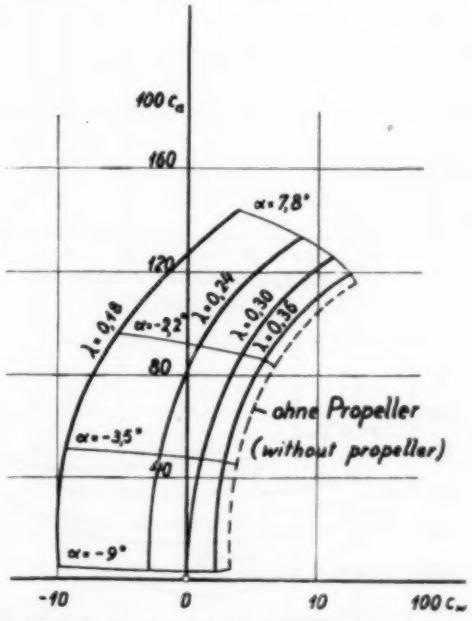


Fig. 11 Polar Curves With Running Propellers

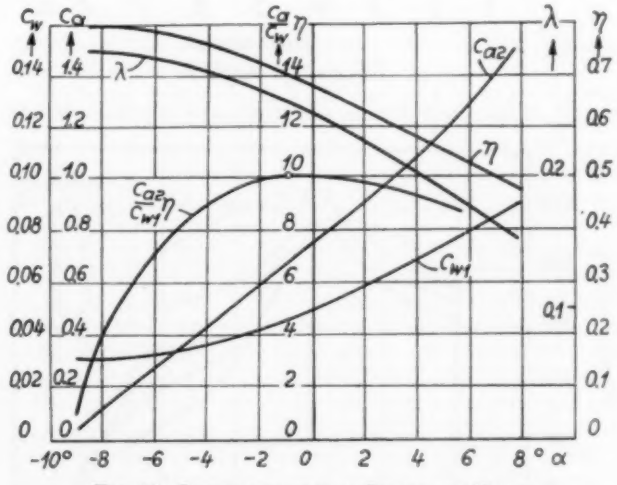


Fig. 12 Interpretation of Propeller Tests

In these tests it was assumed that the load on the boat decreases directly as the square of the speed from the moment of starting to that at which the plane lifts out of the water (Fig. 13).

It is possible to calculate the take-off speed on the basis of the aerodynamical tests, and in the present case it amounted to 130 kilometers or 81 miles per hour. It was therefore assumed that at this speed the wing carried the total weight and that the load on the hull would be zero. The load for the speeds between zero and 130 km. per hr. may then be ascertained by a parabola, the ordinate of which at 130 km. per hr. or 36.1 m. per sec. will be equal to the total load.

On recommendation of the Hamburg Institute a model scale of 1 to 25 was adopted. According to Froude's model law the loads had to be reduced in the ratio $1:25^3$ and the speeds in the ratio $1:\sqrt{25}$ or 1:5. In the present instance the model load therefore works out at $105{,}000/(2 \times 25^3) = 3.68$ kg. and the take-off speed at $36.1/5 = 7.225$ meters per second. These figures are entered on the diagram and the parabola is based on them.

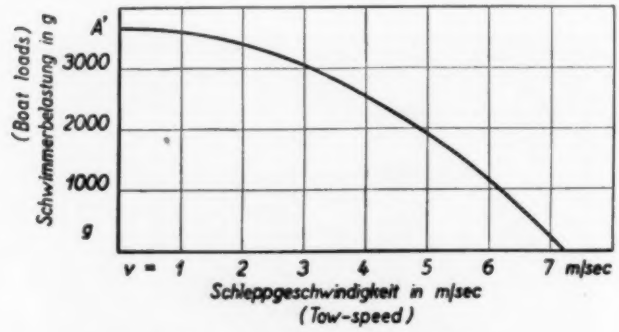


FIG. 13 LOAD CURVE OF BOATS MODEL

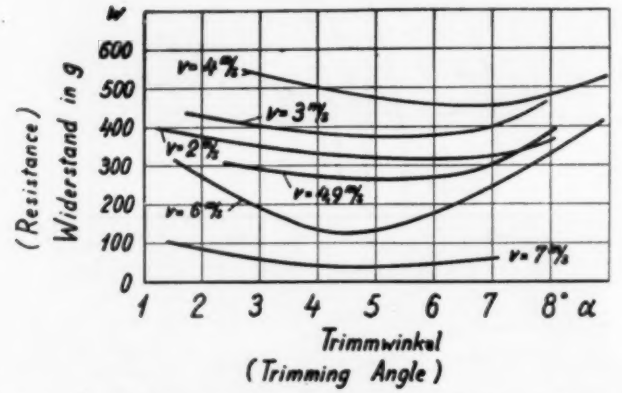


FIG. 14 RESISTANCE CURVES FOR DIFFERENT CONSTANT SPEEDS

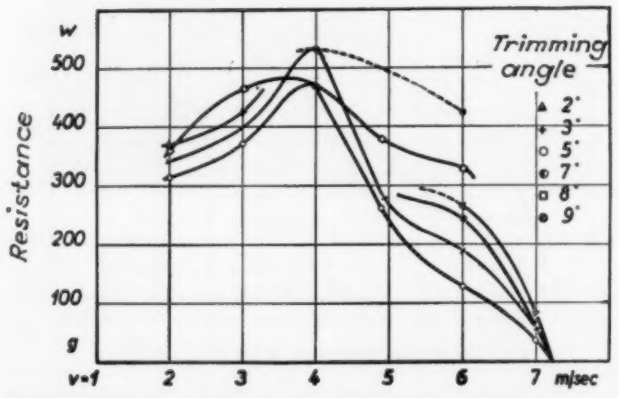


FIG. 15 RESISTANCE CURVES FOR DIFFERENT CONSTANT TRIMMING ANGLES

The experiments were carried through with the boat under a definite load and at a speed for this load corresponding to the decrease of load in the square proportion referred to and at various trimming angles.

In the tests the resistance at every speed and the various trimming angles was measured and plotted in Fig. 14. From the resulting curves the resistances belonging to the same trimming angle were then deduced. These values then were combined to form the resistance curves shown in Fig. 15, which are plotted against the speed. Each curve corresponds to a different trimming angle.

I have purposely left out of the diagram the curves for 4 and 6 deg. as otherwise the graph would not have been so clear. In this model, as can be seen from the views, it is advisable to keep the trimming angle below 8 deg. The most favorable angles are 4 and 5 deg. during the whole starting procedure. Up to $v = 5$ m. per sec., angles of 4 to 7 deg. are practically of equal value. At higher speeds 4 to 6 deg. should be maintained. In this case the second resistance rise in the Ru 93 hull graph would be avoided. The ratio of maximum resistance to flying weight only amounts to 12.5 to 13 per cent for the trimming angles $\tau = 5$ to 8 deg., which alone are practical.

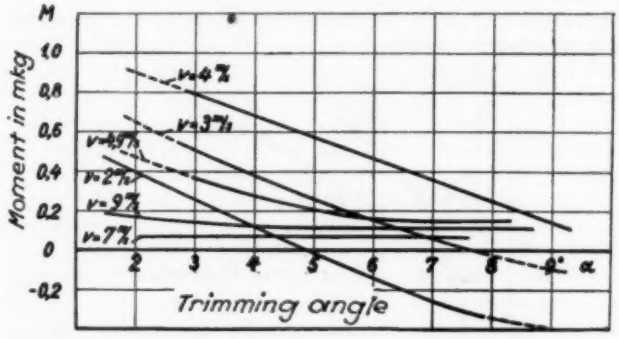


FIG. 16 MOMENT CURVES FOR DIFFERENT CONSTANT SPEEDS

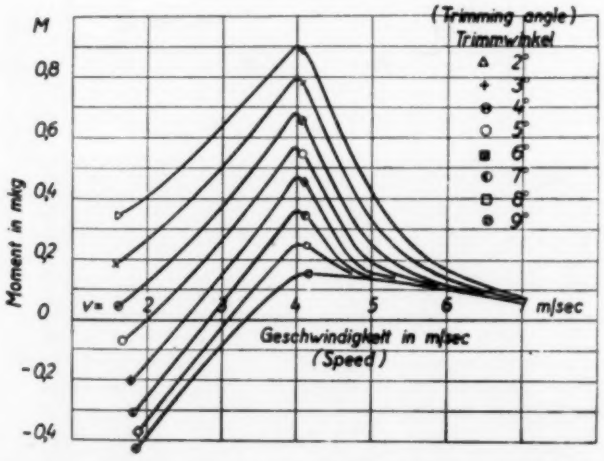


FIG. 17 MOMENT CURVES FOR DIFFERENT CONSTANT TRIMMING ANGLES

In these tank tests the moment at every speed and the various trimming angles were measured and plotted in Fig. 16. From these curves those moment values belonging to one and the same trimming angle were then deduced and plotted in Fig. 17 against the speed, so that each curve corresponds to a different trimming angle.

The moment of the water force is brought into relation with the line perpendicular to the central plane of the hull that joins the two lateral corner points of the step. Positive moment means that the machine must be pressed down so as to obtain the running angle.

As is evident from Fig. 17, the position of the hull may easily be influenced with increasing speed. The moment differences which cause the change over from 4 to 9 deg. at a speed of 6 meters per second are already so low that the accuracy of the

measuring devices employed no longer suffices to record them. From this it is manifest that the hull, after passing the critical speed, can be made to adopt every desired position by operating the elevator correspondingly. It will therefore be very easy to force the plane into a position at which the sum of the water and air resistances is at a minimum.

With the aid of these results it is now possible to formulate the actual starting process; at any instant at the start the sum of all moments must be equal to zero. The moment of the water forces is known for every trimming angle from Fig. 17. We also know the angle of pitch between wing and hull, and it is therefore possible to determine for each trimming angle the angle of incidence of the wing corresponding thereto. It is then possible to calculate the value of the air-force moments from the results of aerodynamical tests at the angle of incidence thus found and at the assumed speed. Based on the particulars given regarding the propeller, it is easy to determine the propulsive force at the speed under consideration. In this we are assuming that each motor will develop 1400 hp. during the start. As the position of the propeller is also known, it is possible to determine the moment caused by the propulsive force. Assuming that the point of application of these forces coincides with the center of gravity of the airplane, then only these three moments will arise. If it does not, however, we have to take into account the further moments arising from the weight and of the masses.

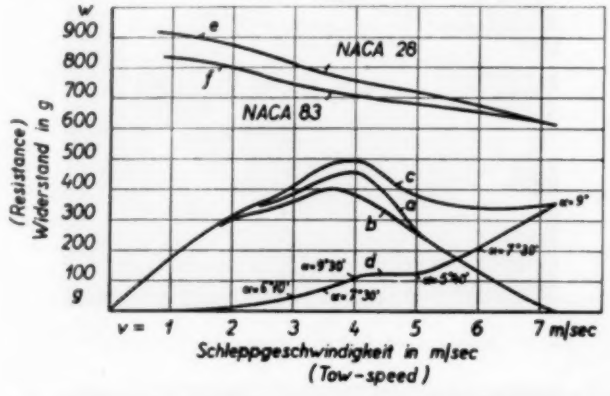


FIG. 18 BALANCE BETWEEN PROPULSIVE FORCE OF PROPELLERS AND RESISTANCE

The trimming angle of the airplane at which the sum of all these moments is zero is established experimentally. At the instant at which the airplane glides at the given speed it will automatically assume the trimming angle thus ascertained. Fig. 15 will then indicate the water resistance corresponding to this trimming angle.

If we carry through this calculation we arrive at the diagram in Fig. 18, in which curve *a* represents the water resistances. In this we are assuming that the airplane trims automatically only up to a speed of 5 meters per second. At greater speeds than this, this model would assume a very small trimming angle (slightly more than 0 deg.) if left to itself. But, as we have seen from the diagram of resistances at various trimming angles previously shown, the resistance increases very considerably with the speed and with a decrease in the trimming angle. In order to avoid an increase in the resistance it will be advisable to force the airplane into a greater trimming angle by operating correspondingly the elevator as soon as a speed beyond 5 meters per second is attained. The influence of this has been considered in constructing curve *a*.

In some airplanes in which the elevator is not effective at this speed, it may be desirable to provide a vertically operating water rudder in order to insure that the proper trimming angle is maintained. This is especially to be recommended for craft in which the control surfaces are located as high up as possible in order to prevent water and spray from touching them. In such places the stabilizer and elevator are sometimes located above the propeller airstream, so that it requires a relatively high speed for them to become effective.

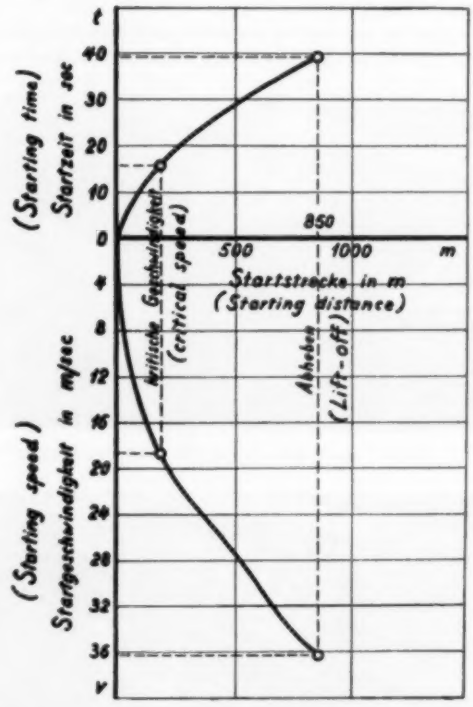


FIG. 19 STARTING DISTANCE, STARTING TIME, AND STARTING SPEED

The water rudder should be located at a point on the float where it will be submerged constantly. Also it must be arranged as far away from the step as possible so as to exert the highest possible moment with the smallest possible surface and application of power. The best position would probably be at the rear step or in front of the forward step.

The use of a water rudder, then, is of course only justified when it is possible to obtain with it a trimming angle at which the saving in air and water resistance is greater, as compared with the arrangement employing no water rudder, than the resistance of the water rudder. The latter should also be retractable into the float in flight so that it causes no additional air resistance.

The application of the values thus obtained to full-scale construction is attended with difficulties, since as yet there is no generally recognized law for such calculations that meets all requirements. However, English researches, as well the experience of the German research institutes, show that the actual maximum water resistance is considerably lower than the value obtained by converting model results in accordance with Froude's law; furthermore, compared with the model results it tends in the direction of smaller speeds.

On the basis of the experiences of the Hamburg Shipbuilding Research Institute I have arrived at a resistance curve *b* (Fig. 18) which will apply to full-scale operation, and which I call the corrected water-resistance curve.

From the trimming angles that result in a self-trimming aircraft, the angle of incidence is obtained by adding the angle of pitch to the trimming angles.

It is now possible to determine from the aerodynamical tests the resistance corresponding to the angle of incidence. This resistance is shown in curve *d*. The sum of these two resistances constitutes the total resistance during the starting period at any given moment. These total resistances are plotted against the speed and are represented by curve *c*.

The propelling force of an airscrew is a function of the speed. In the curves *e* and *f* the propelling forces of two American propellers (N.A.C.A. 28 and 83) are converted to correspond to the scale of the model and for five propellers. The former propeller has better qualities as regards starting; the latter, for economical flight.

It is quite generally acknowledged that the difference between the propelling force of the airscrews and the total resistance constitutes the power available for accelerating the airplane. This of course must at all times be positive, as otherwise the starting would be retarded. The power must also be as great as possible in order to shorten the time and the starting distance. It will be seen in Fig. 18 that the difference between the propelling force and the total resistance is quite large, and the airplane will therefore have a good acceleration when starting.

Referring to Fig. 18, I should like to point out that the angle of incidence α is indicated at various points on the air-resistance curve. It is arrived at, as already mentioned, by adding to the trimming angle the angle of pitch between the hulls and wing, which in the present instance is 2.5 deg. It will be seen that the angle of incidence constantly increases up to a certain speed and then decreases. From this it is evident that the decrease of the load on the float is directly in the square proportion of the speed, which we assumed at the outset does not hold true in the actual starting process. If the reduction of the load on the hull did proceed thus, it would mean that the airplane would

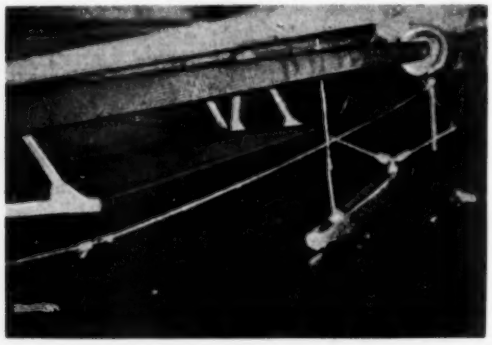


Fig. 20 Towing speed, 3.04 m. per sec.; trimming angle, 3° 12′

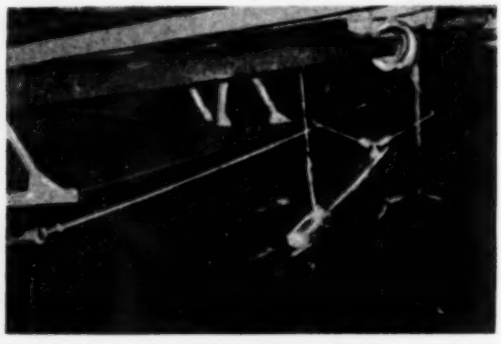


Fig. 21 Towing speed, 4.02 m. per sec.; trimming angle, 3° 42′

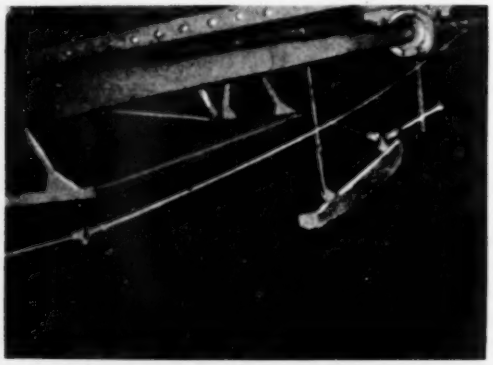


Fig. 22 Towing speed, 4.02 m. per sec.; trimming angle, 6° 2′

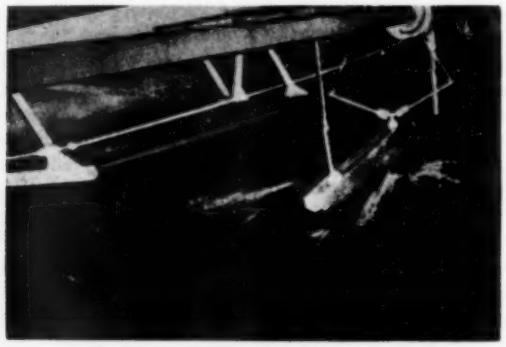


Fig. 23 Towing speed, 5.00 m. per sec.; trimming angle, 5° 6′



Fig. 24 Towing speed, 6.00 m. per sec.; trimming angle, 4° 30′

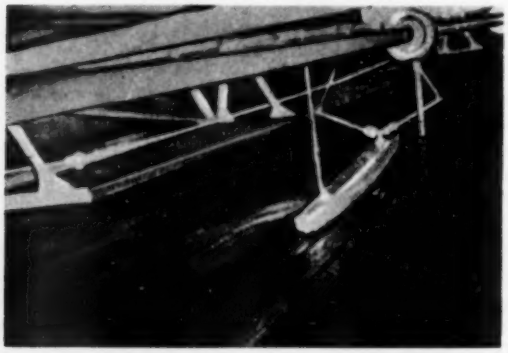


Fig. 25 Towing speed, 6.00 m. per sec ; trimming angle 6° 12′

FIGS. 20–25 TANK TESTS WITH MODEL HULL

have a constant angle of incidence from the moment of starting to the moment that the craft rises out of the water.

In the further tests which are being carried on with the object of ascertaining the counter influences set up between two hulls (when such are employed), this circumstance will be considered and the load carried will be progressively decreased to correspond with the actual aerodynamical conditions.

From Fig. 18 we find that the power available for the acceleration of the airplane is equal to the difference between the propulsive force and the total resistance, depending on the speed. Designating this force by P,

$$dt = \frac{G}{g} \times \frac{dv}{P}$$

in which G/g is the mass of the airplane, t the time, and v the velocity. From this we derive

$$t = \frac{G}{g} \int \frac{dv}{P}$$

and obtain the starting time as a function of the velocity.

If we know the speed as a function of the starting time we can determine the starting distance from

$$s = \int v dt$$

Knowing now the starting time and also the starting distance as a function of the velocity, it is possible to derive also the starting time as a function of the starting distance.

In Fig. 19 the velocity and time are plotted against the starting distance. In this case the propulsive force of the propeller N.A.C.A. 83 has been taken; this propeller is less suitable for starting than the N.A.C.A. 28, but it is given the preference since it is more efficient in flight.

It will be seen from this graph that the critical speed, that is, the speed at which the water resistance reaches its highest value, is attained after but a very small percentage of the starting distance has been traversed. The lift-off is effected after a distance of 850 meters has been covered or respectively after 38 seconds has elapsed, which are results that must be regarded as exceedingly satisfactory.

Tank Tests With Hull Ru 98

Figs. 20–25 show views of a number of tank tests carried out with the hull Ru 98. In Fig. 20 the speed is 3.04 meters (9.97 ft.) per second, which is less than the critical speed. In Fig. 21 the speed is 4.02 meters (13.18 ft.) per second. The hull already glides on the step. In Fig. 22 the hull is moving at the same speed as in Fig. 21, but at a greatly increased angle. In Fig. 23 the speed is 5 meters per second. In all these views it will be seen that the lateral wave caused by the step is very low and that the spray hardly reaches half the height of the hull. Therefore, there is no danger that the propellers, which are situated a considerable distance above the upper edge of the hulls, will be moistened by spray. This favorable condition is to be ascribed to the form of the gliding bottom at the step. The surface of the water at the rear of the hull shows a depression so that there is also no danger that the tail of the hull, which is drawn high up out of the water, will be moistened either.

Fig. 24 was taken with the hull moving at a speed of 6 meters per second and just about to take off. It is seen gliding on a very narrow strip of the front step. In Fig. 25 the speed is the same, but the trimming angle is larger.

Summarizing, I should like to say that it is quite possible, by dint of systematic research work, to determine a form of the hull or float which will insure a short start even in the case of airplanes with a high wing load. This is exceedingly important, as in this way the unknown problems necessarily arising in the design of a new plane type are considerably reduced and the prospects of success greatly enhanced.

Turning now to certain questions of general importance, neither small nor great risks may be run in the construction of a long-distance airplane. Almost all mistakes made have aggravated adverse effects. The worst faults are those having a cumulative effect which must be integrated in order to determine the total of adverse influences at work. A long-distance airplane must be a high-efficiency craft but not necessarily a high-powered craft, for there is a difference between the two. Naturally the high efficiency must be coupled with absolute safety in service.

Influence of Various Factors on Size of Pay Load

In what follows I shall discuss the influence of various factors as they affect the size of the pay load.

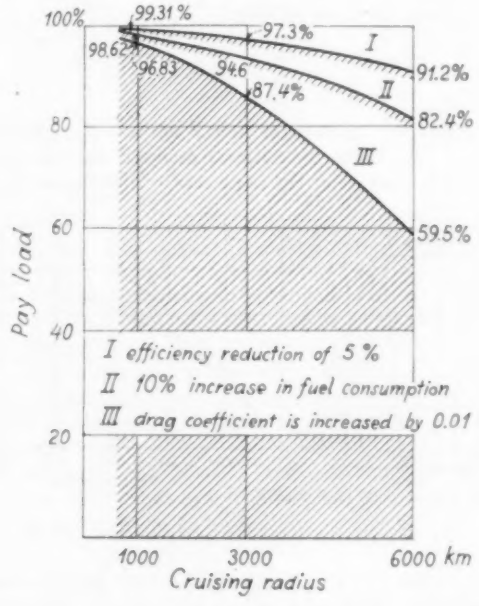


Fig. 26 Influence of Faults With Respect to the Pay Load at Various Cruising Radii

Let us compare three airplanes for flying distances of 1000, 3000, and 6000 kilometers, respectively. These distances are considered as being covered at normal touring speed, that is, in flight with the approximately most favorable gliding figure. All planes, in order to have a simple comparable basis, shall have the same paying load as the first or basic plane. Also we shall assume that the wing load, the starting speed, and also the ratio between load and starting weight in the three cases are identical, and that all three planes have the same aerodynamical qualities and the same fuel consumption per horsepower-hour.

First, let us assume that for some reason or other the propeller efficiency is reduced by 5 per cent, which will correspond to an increase of the average power output by the same percentage. This increase of power output will increase the fuel consumption and require that approximately slightly more than 5 per cent of extra fuel be carried, the weight of which has to be deducted from the paying load. The latter will, therefore be reduced in the various airplanes by

1700, 530, and 133 kg., respectively, corresponding to
8.8, 2.7, and 0.69 per cent of the pay load.

Second, let us assume that the fuel consumption will be 10 per cent greater than foreseen. In this case the 6000-, 3000-, and

1000-km. planes will respectively have to carry the following additional amounts of fuel:

3400, 1060, and 266 kg.

The pay load of 19,300 kg. (42,460 lb.) will be reduced by these weights or by

17.6, 5.4, and 1.38 per cent.

Third, unsuitable location of the engines, radiators, etc. may create an undesirable increase in the resistance. We shall assume that owing to this the coefficient of parasite resistance is increased by 0.01. We have assumed a gliding factor of 14 and a lift coefficient of $c_a = 0.6$ so that the coefficient of the total resistance has the value

$$c_w = \frac{0.6}{14} = 0.043$$

If now this value be increased to 0.053, we find that the airplane will have a gliding factor of only

$$\frac{c_a}{c_w} = 11.4$$

This requires an increase in power output of approximately 14/11.4 = 1.23 times the original amount, and the fuel consumption will increase at the same rate. This means the 6000-,

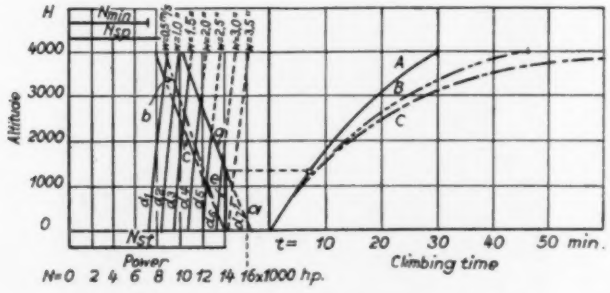


FIG. 27 POWER BALANCE AND CLIMBING TIMES

3000-, and 1000-km. planes will respectively have to carry the following additional weights of fuel:

7800, 2440, and 610 kg.

so that the pay load will be reduced by

40.5, 12.6, and 3.17 per cent, respectively.

These calculations are represented graphically in Fig. 26. The reduced values of the pay load as a consequence of various faults are plotted against the flight distances of the various airplanes.

Curve I shows the values for a reduction of the propeller efficiency of 5 per cent, curve II shows the conditions obtaining when the fuel consumption is increased by 10 per cent, and curve III those when the coefficient of parasite resistance is increased by 0.01.

These curves distinctly show that the pay load decreases when these three faults appear and also the law of this reduction. In other words, the remaining pay load will, in the event of mistakes of whatever nature, sink absolutely and relatively much quicker in the case of long-distance airplanes than in that of planes with a normal radius of action.

MAXIMUM POWER OUTPUTS FROM OVERSIZE ENGINES AT VARIOUS ALTITUDES

Fig. 27 presents a comparison of the maximum power outputs available from oversize engines at different altitudes with the power requirements of the airplanes they belong to when starting, when climbing to height of 4000 meters (13,000 ft.) at various speeds, and when flying horizontally.

These calculations are based on a long-distance airplane equipped with ten engines each developing nominally 1000 hp.

In the left-hand portion of the diagram the output of the engine is plotted against the altitude. The maximum output of the engine at an altitude of 4000 meters or 13,000 ft. is 10,000 hp., and increases at lower altitudes until it reaches 16,300 hp. at sea level. Curve *a* depicts this more distinctly.

Curve *b* shows the power output at various altitudes when a constant power reserve of (16,300 — 14,000)/16,300 = 14.1 per cent is assumed.

Curve *c* shows the power output at various altitudes when a constant and absolute power reserve of (16,300 — 14,000) = 2300 hp. is assumed.

Let us now discuss the power output of 14,000 hp. assumed for the start, which includes a considerable surplus of power.

The power requirements of the airplane for various climbing speeds ranging from 0.5 to 3.5 meters per second are shown by the curves d_1 to d_7 for varying altitudes. These curves are based on the assumption that the airplane always flies with the value of (c_a^3/c_w^2) a maximum.

Horizontal flight at an altitude of 4000 meters is demonstrated by two examples. For flight with a minimum of power this has been calculated at 7100 hp., and for economical flight the power requirement has been calculated at 8000 hp., both of which values have been entered in the diagram.

Having now established the motor power on the one hand and the power requirements on the other it is easy to compare the surplus power under various flying conditions.

When starting we have a power output of 16,300 hp. as against 14,000 hp., in which latter case, however, it will be remembered there is already included an abundant reserve of power. This is highly desirable as it will enable a quick start to be made, even in a high sea. The fact that the high power surplus is not actually required when starting or flying low is very important, as it means that the engine and gears will not be strained, and thus tends to insure great security.

The large surplus of power in reserve when starting will therefore only be required in rare instances as when the sea is running very high.

In order to determine the surplus of power available at any altitude, say, 2000 meters, with the airplane climbing, it is only necessary to compare the abscissa of curve *a*, Fig. 27, for the given altitude with the abscissa of the one of the curves d_1 to d_7 for the same altitude. We see that this difference is a positive one in the case of d_1 to d_5, and negative in the case of d_6 and d_7. This means that it is possible to climb at a speed of about 2.5 meters per second but not, however at a higher rate.

Correspondingly the same reasoning applies to curve *c*.

In order to determine the surplus of power in horizontal flight at an altitude of 4000 meters or 13,000 ft. it is only necessary to find the difference between the abscissa of the curve *a* at 4000 meters, that is, 10,000 hp., and the power requirements of the airplane in economical flight and in flight with minimum power. As shown by the diagram, 2000 hp. are available in the first case and 3000 hp. in the second, or 30 per cent.

I should like to remark here that this figure applies to the beginning of the flight. It increases very considerably in the course of the flight and reaches approximately 60 per cent toward the end.

Summarizing, I should like to point out that the surplus power shown on this diagram by the difference between the power output and the power demand of the airplane constitutes virtually a balance sheet of work. The decrease in this surplus shows that the power reserve required for horizontal flight at high altitudes differs in no great measure from that needed when climbing and starting. The assumption that a very great surplus of power

is required when starting, making for a small actual reserve of power, does not therefore hold true in airplanes that for reasons of safety are also provided with a considerable surplus of power for horizontal flight at high altitudes.

For this reason it appears to me that the auxiliary devices to aid starting, such as towing by another airplane, catapulting, starting rockets, etc., recently so much discussed would perhaps constitute a complication for airplanes of the type described having an oversize power plant. On the other hand, it is quite possible that such starting aids will be very useful for airplanes constructed from other points of view than that of long-distance flying. Everything depends on whether or not normal horizontal flight is to be performed at great altitudes with a large power reserve. If such reserves are required, as is certainly the case for a long-distance airplane, then, as shown by the diagram, it will only be necessary to draw on these reserves for a few minutes during the start and the climb.

The Problem of Speed

Before concluding I shall refer briefly to the speed problem. What I have to say in this respect will not be new to experts, but it may possibly interest the general public and aid in spreading a better understanding of the problem involved.

The distances to be traversed as well as contrary winds are entirely beyond our influence. They are given facts in the problem, and the latter ever so powerful, they must not constitute an impediment to the start and flight of a long-distance airplane.

Only the speed of the airplane is under our control, and this must be as high as possible in order to counteract effectually the influence of the opposing winds. How essential this is will be readily seen from the diagrams of Fig. 28.

The upper one of these diagrams shows the actual traveling speed in kilometers per hour of three airplanes capable respectively of speeds of 100, 200, and 300 kilometers per hour plotted against the speed of the counter wind in meters per second. As is self-evident the absolute deductions from the traveling speeds for the various forces of counter winds are the same for all three airplanes. With a counter wind of 25 meters per second blowing, 90 kilometers have to be deducted from the speed of each of the three airplanes. The 300-km. plane therefore advances at the speed of 210 km. per hour, the 200-km. plane at 110 km. per hr., and the 100-km. plane at 10 km. per hour. This is shown on a percentage basis in the middle diagram. While the 300-km. plane loses 45 per cent of its speed, the 100-km. plane loses 90 per cent.

The adverse influence of contrary winds, however, becomes more manifest still if we consider how it increases the time required to execute a flight of definite length. This is shown graphically in the lower diagram, in which the distance to be covered is taken as 6000 kilometers or 3730 miles. This is the distance, it will be remembered, which was assumed in constructing the basic plane. In the case of the 300-km. plane the traveling time increases from 20 to 28.6 hours; in that of the 200-km. plane, from 30 to 54.5 hours; and in that of the 100-km. plane, from 60 to 600 hours. The increase in time with a counter wind of 25 meters per second in the fastest airplane amounts to 8.6 hours, in the second to 24.5 hours and in the third to 540 hours. It is possible also to read from the diagram the increased flying time required for counter winds of lesser velocity.

Comparing the three diagrams we distinctly see the prolongation of the flight in the case of fast aircraft is small, whereas with slow machines of the fine-weather type it increases with extraordinary rapidity.

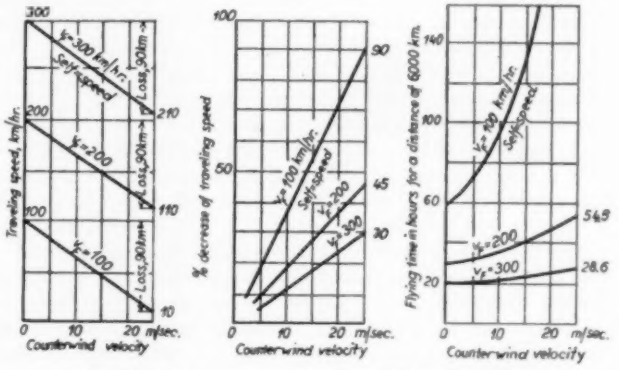


Fig. 28 Influence of Counter-Wind Velocity

Mention might also be made of the fact that airplanes with a high range of speed and good climbing qualities make it possible for their pilots easily to climb to altitudes in which the counter winds have a minimum velocity or in which they reverse their direction. That such searching for more suitable altitudes must not be left to chance but be aided by instruments goes without saying.

From this we see that the speeds of modern aircraft must still be increased very considerably, and I go so far as to assert that in case this is found not to be possible or only partially so, the future of long-distance air traffic will not be very promising.

Four years ago when I stood before the assembly of the German Scientific Aeronautical Society and delivered a lecture on transoceanic flying boats, these aircrafts were not objects of general interest. Today, however, it is quite different.

Today no one asks whether such a long-distance airplane is worth building, but how it is to be built. Its construction or development at the moment interests engineers all over the world to a very great degree. Further, the daily press in educating the general public to understand the developments taking place, and is creating for us engineers the basis on which future successes will be built.

The great problem has been recognized; it is being attacked and its solution is assured.

AER-53-17

Some Aspects of the Seaplane

With Special Respect to Its Development in Germany

By FREDERICK P. SCHUETTEL,[1] NEW YORK, N. Y.

After a brief historical résumé, the author deals, respectively, with the efficiency of the seaplane, constructional features, design of floats and boats, engine arrangement, beam load, seaworthiness, service questions, official requirements to be met, etc. Many tables, charts, and illustrations are included.



ONE of the world's greatest aircraft designers said once in a discussion in Berlin, in the fall of 1929: "Why do we talk so much about seaplanes if we are able to build multi-engined landplanes that can fly with one engine out of commission and that can thus cross the oceans with reasonable safety?" And, indeed, he had such valuable justification of his words as that given by the flights of Byrd, Kingsford-Smith, and of the Dutch India mail fliers. But his optimism went too far. In giving due credit to the pioneering spirit of the crews who succeeded in landplanes, we should not forget all those who perished in the attempt, mostly because of the insufficient floatability of their planes. We cannot use landplanes for transoceanic commercial flying, as they absolutely depend on the reliability of their engines and are merely flying fuel tanks when traveling directly from one continent to another. And as far as multi-engined landplanes are concerned, it is still a question whether they can really reach land with their speed considerably reduced by the failure of one of the engines.

We must continue along the way indicated by the American NC-boats in 1919, by Franco (Wal) in 1926, by de Pinedo (Savoia) and Beires (Wal) in 1927, and by Mermoz (Latecoere) and von Gronau (Wal) in 1930. Admittedly the range of the seaplane is not yet sufficient; but we know how the Atlantic can be subdivided, and by beginning on smaller routes we must prepare for the service, which will be a fact as soon as, for instance, the Diesel engine has been further developed. Our aim must be to produce seaplanes which will be able to compete with steamships, because they have here the best opportunity to show their superiority in speed. This competition would by no means affect the steamship companies—consider only the number of passengers aboard a *Leviathan* or *Bremen*. And developments show that it is just these companies that do take an active interest in airplane service: the ship-to-shore services of the North German Lloyd and of the French Line are well known, and some weeks ago the Lloyd Sabaudo began to connect its steamers with an express flying-boat service at Gibraltar. The fact that the commander of the DoX, Captain Christiansen, was previously master of a passenger steamer in the Hamburg-Mexico service, is also significant.

One pronounced advantage of the seaplane is the very low cost for the acquisition and maintenance of its "landing fields." So we often find seaplane services running along mountainous coasts where landplane fields are scarce, e.g., in Italy. And flights over unexplored countries have been very successfully made with seaplanes (Mittelholzer, Cobham over Africa, Pinedo over Brazil), relying on rivers and lakes as landing grounds. The natural landing areas offered by the water are normally level and large; it is not by pure chance that the heaviest airplane in the world, the DoX, is a flying boat. Besides, water areas are mostly nearer to the big cities in the world than are landing fields: e.g., New York, London, Chicago, Buenos Aires, Rio de Janeiro, Sidney; in Berlin, however, the existing "Flughafen" has the best possible location.

Political reasons are of course very important; so we have seen during the last years how Great Britain has been anxious to develop efficient seaplanes for improving the connections within the Empire. Similar reasons of a more commercial character have led to the establishment of the wide network of the Pan-American Airways.

Historical

As a part of what follows deals with German seaplanes, a few historical notes are appropriate at this point. German seaplane development was negligible before 1911. The first seaplane competition was held on Lake Constance with a representative of the single-float type. But this was abandoned in favor of the twin-float seaplane, which was the standard type during the World War, equipped with 160- to 220-hp. engines. There were flying boats used in the Mediterranean, but the German Naval Air Forces preferred the twin-float seaplane, which showed a reasonable amount of seaworthiness for that time. Incidentally, it may be stated that Dornier began his work in 1914 with the design and construction of flying boats of 142.5 ft. span and about 20,000 lb. weight. The first one made was a failure, but the second flew well, and the third passed the severe seaworthiness test of the German Navy with success.

Later came the termination of activities which was forced upon the German aircraft industry by the Treaty of Versailles, and it was not until the end of 1925 that the aircraft limitations of that pact were interpreted as applying only to military aircraft. In the meantime German designers had been compelled to sell their abilities in foreign countries to keep alive, so a certain experience and tradition was available when the time for planning efficient commercial airplanes came. During recent years there have been four establishments in Germany which have more or less specialized in seaplanes: namely, the Dornier, Rohrbach, Heinkel, and Junkers.

Efficiency of the Seaplane

As far as aerodynamic efficiency is concerned, in the smaller sizes, the seaplane is inferior to the landplane of equal weight because of the drag of its floats and bracing. The limit depends on the type of construction of course. For example, the Dornier Works succeeded in developing a two-float seaplane giving the same performance as an equivalent landplane (Figs. 1 and 2). This was done by carefully fairing the float bracing by means of "pants."

Special attention must now be given to the question of over-

[1] Aeronautical Engineer. Mr. Schuettel was born in Berlin, Germany, and was graduated from the Technische Hochschule of that city. He was formerly research engineer of the Dornier Metallbauten G.m.b.H., of Friedrichshafen.

Presented at a meeting of the Metropolitan Section of The American Society of Mechanical Engineers, New York, N. Y., Nov. 18, 1930.

loading. A seaplane cannot be overloaded to the same extent as a landplane, but this is obviously no fault from the standpoint of safety, as excessive stresses are thus avoided. All long-distance flights with landplanes have been made possible by overloading, and the machines would never have passed any official approval tests because of reduced load factors as well as because of lack of sufficient take-off and climb performance.

Table 1 gives data on maximum take-off performances as



FIG. 1 DORNIER TWO-FLOAT SEAPLANE

exact as the author could get them. The figure of merit is the familiar

$$f = \frac{W}{cP} \times \sqrt{\frac{W}{S}}$$

only with a coefficient c in order to take into consideration variations of the propeller efficiency. This coefficient was assumed to decrease from 1.0 for 1000 r.p.m. linearly to 0.93 for 2000 r.p.m. It was not necessary to introduce different factors for wooden propellers as only geared engines were equipped with them, so that the difference in efficiency was negligible. The hp. values given are maximum hp. only for this comparison, the actual power at take-off being considerably smaller.

From this table it is seen that the figure of merit for the maximum possible take-off is about 110 for normal landplanes as against 75 for normal boats and 65 for normal 2-float seaplanes without wind or special starting devices. The author has prepared from tests and reports a set of coordinated curves for normal airplane (Figs. 3 and 4). There is of course no fineness considered, no aspect ratio, no tandem efficiency, and no abnormal decrease of power with altitude due to altitude control or blower systems. Looking at these diagrams with the above figures in mind, we find the limiting performances for the different types to be as follows:

	Power reserve, per cent	Initial climb, ft. per min.	Ceiling, ft.
Landplane	7	40	1300
Flying boat	24	280	6000
Two-float plane	32	410	8500

Remember now the approval requirements for landplanes, made for the sake of safety: "Take-off within 1000 ft., initial climb 400 ft. per min. (Department of Commerce), or altitude 66 ft. within 2130 ft. from beginning of take-off, rate of climb 157.5 ft. per min. in air of specific weight 0.0686 lb. per cu. ft. (D.V.L.)." Fig. 5, derived from these data, shows clearly that American and German climb requirements are substantially the same for machines with normal engines. It is thus seen that for reasons of safety landplanes are not permitted to be as overloaded as the technical possibilities admit, and that for commercial purposes the landplane does not possess the superiority which it might be assumed to have from its distance records. The take-off requirements are even more rigid: both are roughly equivalent and correspond to a time of, say, 18 sec. (this depends largely on the wing loading). A further diagram (Fig. 6) connecting take-off time and the figure of merit shows that even the



FIG. 2 DORNIER LANDPLANE

TABLE 1 MAXIMUM TAKE-OFF PERFORMANCE OF VARIOUS AIRPLANES

Type	$W_{max.}$	Hp.	S	W/S	W/Hp.	f	Take-off
Landplanes							
Lindbergh (Ryan)	5,250	225	319	16.5	23.3	100.0	About 2500 ft.
Byrd (Fokker)	15,600	675	730	21.4	23.1	113.0	48 sec., 3268 ft.[a]
Woolaroc (Trav. Air)	5,375	225	308	17.5	23.9	106.0	.....
Junkers Dur. Record	8,830	360	463	19.0	24.5	111.0	25 sec., 2460 ft.[a]
Fairey Long Range	16,000	600	846	18.9	26.7	119.5	3700 ft.
Kingsford (Fokker)	15,400	675	730	21.1	22.8	110.5	3500 ft.
Coste (Breguet)	13,900	700	647	21.5	19.8	92.0	2630 ft.
Seaplanes							
Dornier DoX	113,000	6300	5130	22.0	17.9	84.5	(At 1300 ft. altitude)
Rohrbach Romar	41,890	2340	1830	22.9	17.9	85.5	.....
Dornier DoR	26,500	1400	1570	16.9	18.9	78.0	.....
Rohrbach Rocco	23,400	1400	1010	23.1	16.7	80.0	.....
Latecoere 28	11,070	640	690	16.0	17.3	69.5	.....
Earhart (Fokker)	11,000	675	730	15.1	16.3	67.0	.....

[a] Special slope.

400-ft. per min. limit is not often reached. The comparative take-off times taken from the preceding curves and plotted together are rather interesting (Fig. 7). The curves show the amount of take-off time for the seaplane is longer than for an equivalent landplane.

A certain handicap to the seaplane's efficiency lies in the additional weight necessary to make boats or floats watertight and fit to withstand the stresses encountered in a seaway. Here the weight empty is about 10 to 20 per cent greater, depending of course on the type and purpose—high-sea or river service. But this handicap is counterbalanced in the case of commercial seaplanes by the larger normal gross weight they may have, and

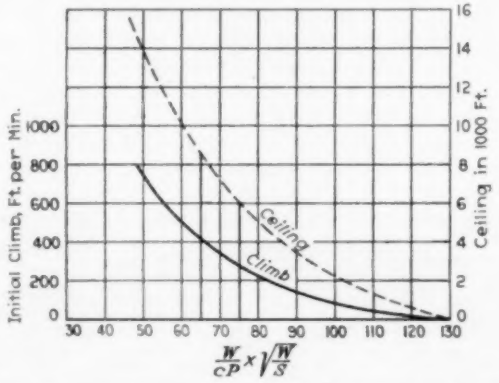


FIG. 3 CEILING AND CLIMB VERSUS FIGURE OF MERIT

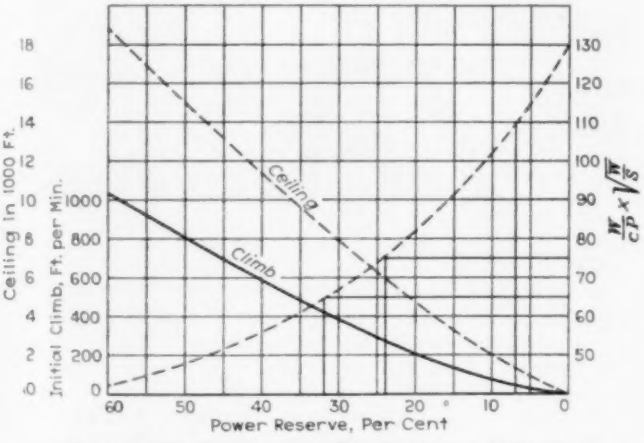


FIG. 4 CEILING AND CLIMB VERSUS POWER RESERVE

which is made possible by ample water areas with no smokestacks in the take-off direction requiring the limitation of take-off performance.

Both the natural take-off limit resulting from water resistance and friction and the increased weight of course govern the maximum ranges of seaplanes, which are of the order of 2000 to 2500 miles as compared to 5000 to 6000 miles for landplanes. But, as already said before, both types are practically equivalent for commercial service by reason of limitations which are necessary from considerations of safety.

CONSTRUCTIONAL FEATURES

Types. The different types are well known; we distinguish between seaplanes with positive and with negative stability. Twin floats, twin boats, sponson boats, and boats with low side floats (called triple floats by Richardson) belong to the group having positive stability, while single-float types and boats with wing-tip floats belong mostly to the group with negative water stability. The influence of stability on the seaworthiness will be discussed later. From the standpoint of weight the boat will in most cases be more efficient than the equivalent float plane, as the weight of the boat with stability devices and engine nacelle

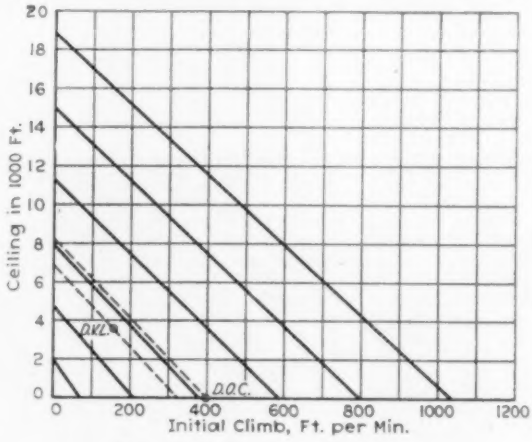


FIG. 5 RATE-OF-CLIMB CURVES

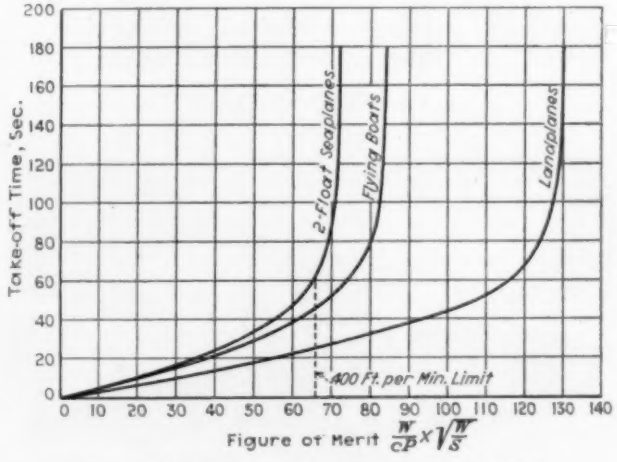


FIG. 6 TAKE-OFF TIME VERSUS FIGURE OF MERIT

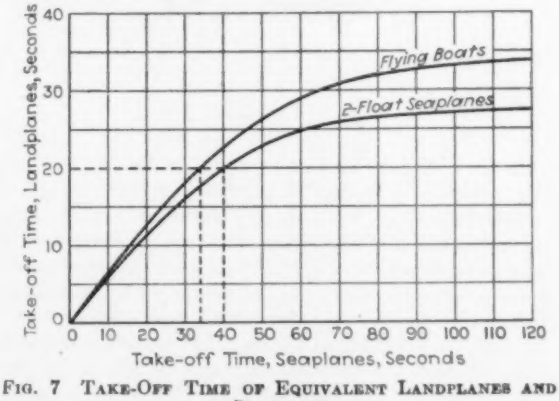


FIG. 7 TAKE-OFF TIME OF EQUIVALENT LANDPLANES AND SEAPLANES

Types. The different types are well known; we distinguish between seaplanes with positive and with negative stability. Twin floats, twin boats, sponson boats, and boats with low side floats (called triple floats by Richardson) belong to the group having positive stability, while single-float types and boats with will be less than that of the fuselage with floats and float bracing. On the other hand, there is the possibility of exchanging floats during service or of substituting a wheel undercarriage on a float plane. The combined landplane and seaplane, the amphibion, should not be overlooked here. This construction lends itself

well to beaching, but it depends on each special case whether its disadvantages shall be taken into account: namely, reduced seaworthiness—a consequence of the special necessity to save weight, increased drag, surplus weight, and the additional maintenance of the landing gear affected by sea water. The published data for the two types of craft make this clear:

	Useful load	Top speed	Initial climb	Cost
Fokker F11 (Amphibion).....	2430	112	700	33,775
Fokker F11a (Flying boat).....	2700	120	800	32,500
Reduction, per cent.....	10	6.6	12.5	4 (additional)

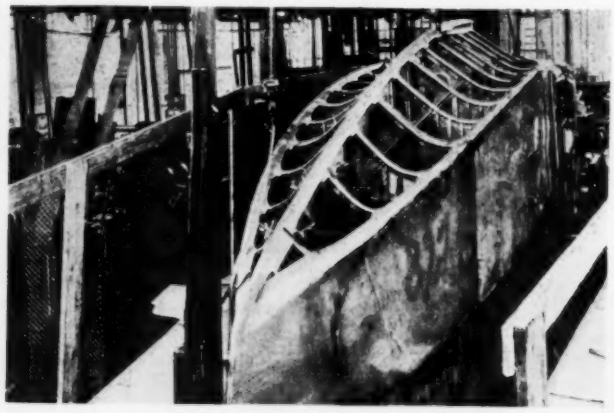


FIG. 8 ROHRBACH HULL

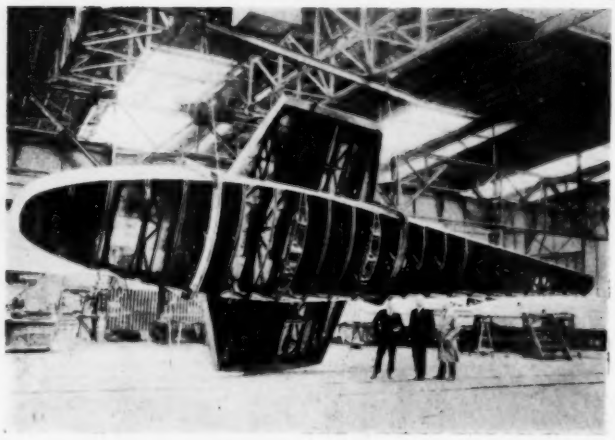


FIG. 9 HULL OF DORNIER WAL

Further, the additional landing gear will probably be so heavy for bigger ships as to make them rather inefficient, especially if one considers that they are subject to the official landplane rules.

The position of the side floats is of course connected with the question of seaworthiness, but apart from this, say, for ships for inland waters, the author would always prefer that position which provides positive stability. This means low side floats and imposes the necessity of having them nearer to the hull in order to avoid high yawing moments from a possible wave impact or from landing with an inclined wing. Thus the location of the side float has wandered from the wing tip to the hull; now it is attached to the hull in the case of sponsons. It is interesting to cite here Dr. Dornier's statement that the relative span of his sponsons decreases with increasing size of the whole boat.

Design of Floats and Boats. The design of floats and boats is a specialized kind of ship design, with much more consideration being given to lightness in weight, and strength is required to withstand landing impact at much higher speeds, whereas in building ships only the bending moments which result from the wave distribution are taken into consideration. There is, also, the twisting and bending influence of the tail forces in the case of a plane. Notwithstanding all this, we find the same constructional features: either strong transverse frames with a few longitudinals or many through-going longitudinals with some frames, resembling the Isherwood system of the shipbuilder. There are also many intermediate possibilities, and they all have been tried by different designers. Strong transverse frames are represented, e.g., by Rohrbach's hull construction (Fig. 8). Dornier employs transverse frames with fairly wide spacing inside the hull and through-going longitudinals outside (Figs. 9 and 10). Small longitudinals running through in narrow spacing are found on the "Consolidated Commodore," small transverse frames being attached to them but not to the hull covering. There is also the "Short Calcutta" with transverse rings of sheet covering (their joints being at the same time riveted to the frames) and numerous longitudinals narrowly spaced and interrupted at the frames (Fig. 11). The Blackburn boat, however, has many longitudinals running through, a number of strong frames being cut out for them and attached to the covering. The spacings of the stiffeners are rather characteristic—they are about 6 in. on British boats as against 1.5 to 2 ft. on German boats. This is to a certain extent also due to the derivation of the British metal design from wooden boats of the flexible type, and to the more complicated shape. The corresponding high number of rivets must obviously have an unfavorable influence on the production costs.

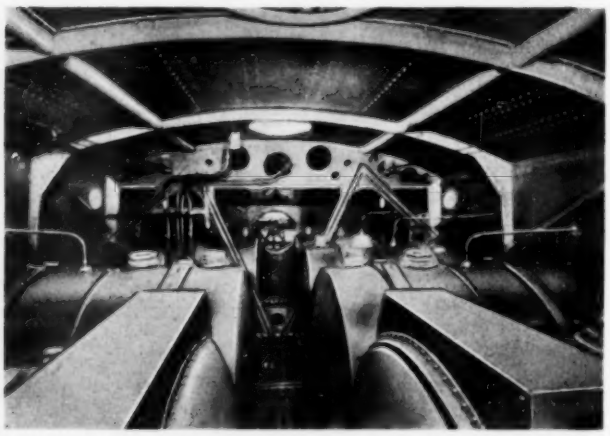


FIG. 10 FUEL ROOM OF DORNIER WAL

As for the floats, the standard type of Heinkel's is a wooden float with correspondingly flat walls, having ash frames and longitudinals and plywood covering (Fig. 12). Junkers, however, uses the reinforcing effect of his corrugated sheets also for the longitudinal stiffening of floats at the side walls (Fig. 13). An example of the use of wide spacing of stiffeners in the United States is the Edo float, which has fairly thick walls, a few longitudinals, and only the bulkheads as transverse members.

We have seen that the longitudinals may be provided at the inside or at the outside of the hull, the latter arrangement avoiding crossing difficulties. Service experiences show, however, that outside longitudinals should be avoided on the bottoms of metal ships as they complicate beaching and are likely to be damaged, thereby causing corrosion.

Engine Arrangement. The engine arrangement is normal for

float seaplanes; one has only to see that the clearance of the propeller from the water line is preferably not less than 2 ft. As the hull cannot be considered for the installation of engines so long as these are directly coupled with the propellers, all boats have separate engine nacelles either under the wing or above, the position of the screw just ahead of the wing resulting in strong interference. The position under the wing, especially in use on biplanes (with reference to their upper wing), has the advantage that the influence of the parasite drag on the efficiency of the wing is small; but the propeller slipstream counteracts the circulation, whereas in the position above the wing it increases the circulation and the lift, but shows a very bad drag with idling propellers and affects the suction side of the wing unless the nacelles are carefully designed. The question of installation is less difficult on bigger ships as the propeller clearance from the water increases with the span. It is clear that the necessity of providing separate

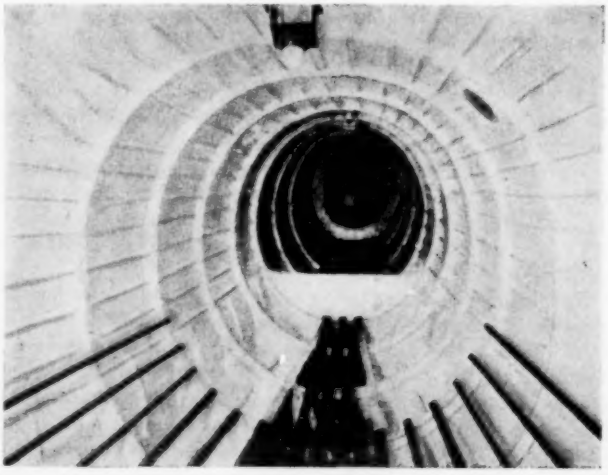


FIG. 11 INTERIOR OF SHORT CALCUTTA

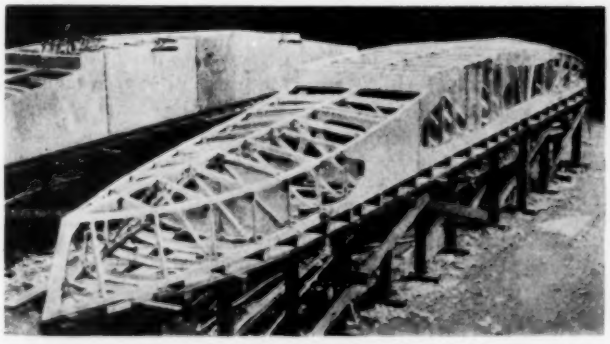


FIG. 12 HEINKEL FLOAT (WOOD)

engine nacelles has furthered the adoption of the tandem arrangement for flying boats; its advantages—small drag, no yawing moments due to engine failure—and its disadvantages—loss of efficiency for quick-running airscrews, cooling difficulties with the present air-cooled engines—are widely known. There is of course the influence of the high line of thrust on the air stability, in so far as it makes it more difficult to obtain the right balance for both cruising and gliding flight conditions.

Maneuvering on the water is more difficult and more important than on land, the importance being accentuated by the fact that seaplanes on the water are regarded as engine-driven ships subject to international navigation regulations. This point of

TABLE 2 VALUES OF BEAM LOAD W/B

Type of plane	W Weight, lb.	$W_{max.}$ lb.	B Beam, ft.	W/B	W/B^2
Dornier DoX	....	113,000	15.40	7340	477
Rohrbach Romar	....	41,890	6.82	6120	900
Dornier DoR	....	33,100	11.15	2970	266
Rohrbach Rocco	....	23,400	6.40	3660	572
Short Calcutta	24,000	.....	10.40	2310	222
Consolidated Commodore	17,600	.....	8.42	2090	248
Dornier Wal	14,330	.....	8.20	1730	211
Supermarine Southampton	14,300	.....	7.40	1935	260
Sikorsky S 38	10,480	.....	6.50	1610	248
Curtiss H-16 (N.A.C.A. Rep. 346)	10,000	.....	10.00	1000	100
Latecoere 28	....	11,070	4.43[1]	1250	282[2]
Junkers G 24 W	14,350	.....	4.59[1]	1565	341
Junkers A 20 W	3,200	.....	2.43[1]	659	271
TS-1 (Rep. 328)	2,150	.....	2.17[1]	495	228

[1] Width of one float (2-float seaplanes).
[2] $W/2B^2$.

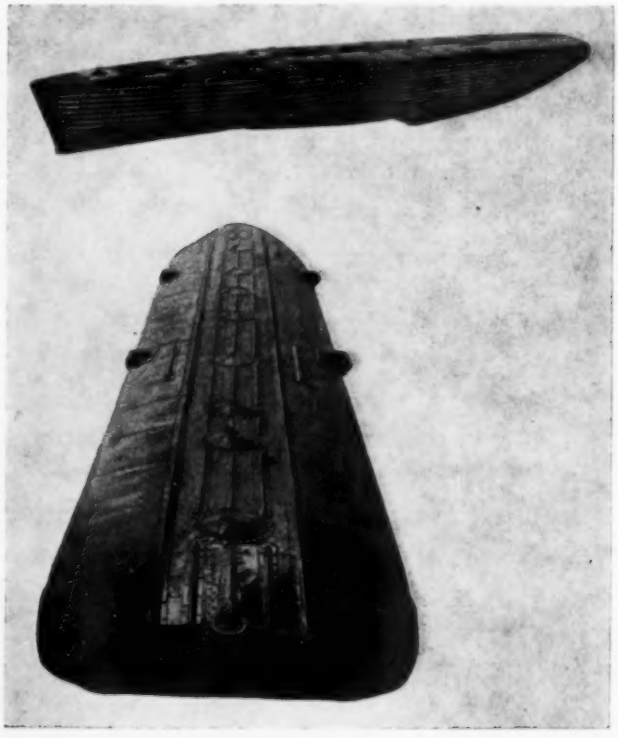


FIG. 13 JUNKERS FLOAT

view will often require the use of an enlarged rudder or ailerons, which mostly means increased control forces during flight.

Beam Load. It is interesting to investigate the beam load, or the gross weight divided by the width of the hull, for the different types and sizes of seaplanes. Values for this are given in Table 2, as well as values of weight divided by the square of the width.

These values are plotted on two diagrams (Figs. 14 and 15), and attention is called to the comparatively low values of the H-16, Calcutta, and DoR, and the high values of the Romar and Rocco. Points of two-float planes are seen on the same curve in Fig. 14. The difference between float seaplanes and boats is only seen when considering the bottom loading calculated as $W/2B^2$ for the two floats. This value of the bottom load of course varies less for normal machines than that of the beam load, but any variation of the type is much more pronounced. The plotting of points for different conditions of gross weight is justified for rough comparison as their alteration does affect but little the character of the curves, the points moving around along straight lines which radiate from zero. The second curve indicates why the take-off

of float seaplanes is normally worse than that of a boat, but it also shows that a narrow hull does obviously not affect the take-off possibility if adequately designed (compare Table 1 of take-offs). But of course the question of space for passengers and load is still more important from the economic point of view.

It is justifiable not to consider floats and sponsons when measuring the width, as these devices either do not assist very much in the take-off, or have such incidence against the hull bottom that any wave impact will probably not meet both with the same force. Thus we can assume the impact pressure to have some relation to the curve of the "bottom load." The actual values

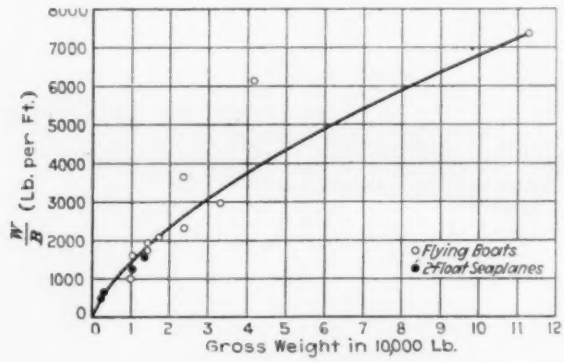


FIG. 14 BEAM LOAD VERSUS GROSS WEIGHT

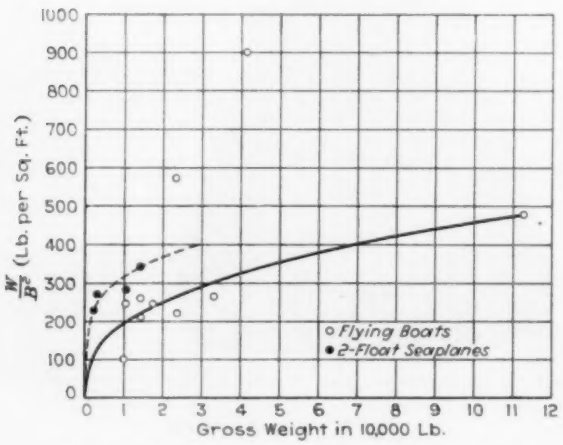


FIG. 15 BOTTOM LOAD VERSUS GROSS WEIGHT

measured by the N.A.C.A. (Reports 328 and 346) are about 10 lb. per sq. in. for the TS-1 and 14 lb. per sq. in. for the H-16, and the author imagines that bottom pressures on Rohrbach boats are not far from being as high as indicated by their proportional position on the diagram.

SEAWORTHINESS

The definition of seaworthiness deals with the maneuverability of the seaplane on the water and the combination of flight and marine performances when taking off and landing. The seaworthiness is sufficient when no injuries or damage occur to those on board the plane or to the plane itself. Going into details, seaworthiness in general comprises behavior during: (1) mooring, (2) drifting, (3) towing, (4) taxying, (5) landing, and (6) taking off.

There have been designers striving after the seaplane with "absolute" seaworthiness, but until now the seaworthiness obtained has always borne a certain relation to the weather conditions and the seaway encountered. It is hardly reasonable to require that absolute seaworthiness which we are unable to obtain, even with considerable sacrifices of useful load. It should be borne in mind that storms often cause great damage on the big liners.

From the standpoint of safety, adequate performance must be required for points (1) to (5), but it seems reasonable to except the take-off in a high seaway. Limiting thus the definition of seaworthiness does not affect the safety; the seaworthy plane is able to land in a high sea, is always maneuverable, and it rides out a storm and takes off again under better weather conditions when repairs have removed the cause of the forced landing. Even such early types as the NC-boats in 1919 and the old H-16 of the above-mentioned report experienced no serious troubles when landing normally.

When discussing seaworthiness, one should divide all seaplanes into two groups as the D.V.L. does for its strength requirements, namely, into planes for rivers and lakes, and planes for the high seas. The formation of a third group might be suggested for the coast and the Great Lakes, but presumably the short waves of the Great Lakes, e.g., are at least as dangerous as those of the ocean. As all difficulties are more or less due to the seaway, the following paragraphs will deal especially with the "high-sea" group.

Investigating details Nos. 1 to 6, we find that they are a function of different seaplane characteristics: of the flight performance as far as landing and taking-off speed and time of the take-off are concerned; and of the water stability, size, and position of the wing for the remainder.

The landing and the taking-off speed should be as low as possible, of course, as the wave impact increases with the square of the speed. But considerations of flight performances of the weight-strength ratio (especially of the wing), and of the influence of a large wing on the water stability have led in the opposite direction. The landplane has gone through the same development even without this third reason, so that it is not fair to blame the seaplane for it.

The lateral water stability of a seaplane seems to resemble that of any floating body. This is true in the case of a calm, but with wind the stability of a seaplane can only be compared to that of a sailing vessel—and this only to a certain extent as the wing area cannot be reduced like the canvas of a sailing ship. This makes clear the danger of a large wing surface, corresponding to a small wing loading. It shows also why the initial stability expressed by the metacentric height is greater for an inherently stable seaplane than for a steamship, e.g., 15 ft. for the DoX. Apart from the size of the wing the heeling wind moment depends largely on the height of the former above the water line (this is often excessive for the upper wing of biplanes), on the projected area of the hull, and on the dihedral of the wing. For example, Rohrbach accepted the tapered wing in order to balance the influence of a pronounced dihedral. Incidentally seaplanes with a high wing loading do not tend to fly away from either beach or buoy in a 40- to 50-mile "breeze."

In discussing the marine qualities of the different types, a new factor must first be considered, namely, the size—this not from reasons of stability but merely because it gives a measure of the distance from the water line to the pilot. It is quite clear that a pilot sitting high and dry in a small-float seaplane is better able to view his surroundings than one sitting low in a small boat and more or less disturbed by spray. For the bigger sizes, however, the boat is obviously better; it should always be possible to build it lighter and with less drag. The average limit for both types according to experience is at about 11,000 lb. gross weight or about 900 to 1000 hp. This is one of the reasons why the lighter two-float seaplanes attained the first three places in the severe test of the German Seaplane Competi-

tion in 1926 (Von Gronau being the winner), and that helped to develop the flying boat as the heavy patrol plane of the U. S. Navy.

Although as said before the high position of the upper wing of a biplane is rather unsatisfactory, the low position of the lower wing is still worse. The possible heeling angle of a ship with low wings is very small, and the wing tips are much exposed to the seaway. A construction cannot be considered as satisfactory when students are advised to cut off its fabric in case of distress, and in view of all this, the monoplane offers the best solution.

The most acceptable take-off and landing qualities seem to have been obtained with the central-body seaplane, i.e., with the single-float seaplane and the central boat. The better maneuvering qualities are of course associated with positive stability, that is, with the two-float system and with the central inherently stable boat. Side wind considerably endangers every seaplane with negative stability due to the increased angle of attack. One logical step has been to lower the wing-tip floats far enough to insure positive stability, as a hull that would itself give this stability would be too broad. Rohrbach, for example, did this and employed side floats instead of tip floats at the same time. But for high-sea service the low side float has not proved its



Fig. 16 Rohrbach Rocco Taking Off

superiority. The stabilizing device of outstanding merit is still the Dornier sponson, as it insures a high degree of positive stability, centralizes the buoyancy and impact forces, does away with the danger of being torn away, which is always possible for floats, and avoids the large impact of a broad, inherently stable boat by the incidence of the sponson. We have seen before that only the single-body plane combines good taking-off and maneuvering qualities, so that we understand now the success of the inherently stable sponson boat, as exemplified by the Dornier, Latecoere, and Supermarine. Planes of the "Wal" type have repeatedly been able to take off and land (of course) with considerable side wind or along a rather heavy swell, which ability is very important for all cases where wind, seaway, and swell cross each other. Take-offs of this kind are also mentioned in Report No. 328, but this same report also shows the unsuitability of the wing-tip float for this maneuver, as it was damaged by waves causing a ground loop of the whole boat.

Longitudinal water stability presents no difficulties whatever, as a certain length of the boat, e.g., is given by the position of the tail and as the length of floats is already partly determined by the required buoyancy. The necessity that the ship shall be stable with any two compartments flooded is of some importance, too, but it affects more the spacing of the bulkheads.

The shape of the bottom has a distinct influence on the seaworthiness, as it determines the wave impact, the ease of take-off, the smoothness of a landing, and the amount of spray water. It has often been stated that the flat bottom gives the shortest take-off on smooth waters, whereas keeling is more agreeable for seaway. A number of characteristic sections are shown in Fig. 17. Interesting is the application of longitudinal steps, the lowest part of the bottom being especially used for planing at high speed. The success of the flat-bottomed Wal is partly due to this longitudinal step which avoids contact of the whole bottom with the water at once and serves as a very efficient stiffener. The straight keeling of American boats compared

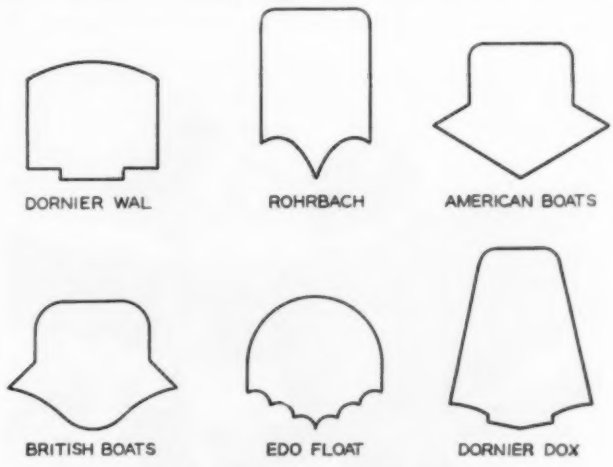


Fig. 17 Characteristic Bottom Shapes

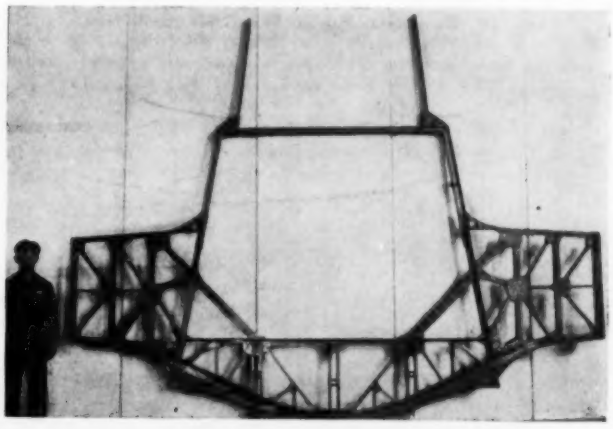


Fig. 18 Main Frame of Modern Dornier Boat

with the wave-binding Rohrbach bottom should be less efficient as regards the time of take-off and maximum tests, but no actual figures are yet known. A combination of longitudinal steps with the Rohrbach shape is represented by the Edo floats. It is interesting to note that without excessive width of the hull the draft of the Dornier DoX is 3.4 ft. fully loaded as compared with 2.7 ft. of the Supermarine Southampton, for example. Economic considerations suggest a shape of bottom lying between the flat one and the type with pronounced keeling, as illustrated by the main frame of a modern Dornier flying boat (Fig. 18).

Service Questions

Embarking. There are a number of questions for flying companies which arise from peculiarities of the seaplane. Taking up embarking first, there are these possibilities:

1 Entering the plane from the bow when it is lying on the beach or on an inclined runway with its front part. There is no question here of seaway as seaplane bases will always be

situated on sheltered water, but wind will often affect the taxying to and from such a place.

2 A pier or a rigid bridge is unsatisfactory when the tide has to be considered, as it should be so low that the wing has sufficient clearance.

3 The use of buoys is logical as in the case of yachts. However, it makes a commuting boat necessary, but this will often be available as it is also required in the case of accidents.

4 The installation of a raft either attached to an anchor or sliding up and down in guides is to be earnestly considered. It



FIG. 19 ROHRBACH ROBBE WITH BEACHING GEAR

may be accessible over a movable bridge and the seaplane will be able to go alongside under all wind conditions if the distance from the land is great enough.

5 The easiest solution is of course given by the amphibion; the floating runway of the Air Ferries in San Francisco deserves to be especially noted.

Beaching. It is the custom on European seaplane lines to leave the machines moored during the season and to take them out from the water only for repairs and overhauls. There are several methods:

a The launching cradle or slip car is well known; it needs a special shape for all non-flat bottoms, and this makes the maneuver rather difficult, say, with side winds.

b Some manufacturers supply special auxiliary wheels for their ships so that they can taxi up the beach or slope with their own power after the gear has been applied by men with rubber suits. (Fig. 19, Rohrbach, and Fig. 20, Dornier beaching gears.)

c Whereas these wheels remain at the ground base in the case of seaplanes, the amphibion incorporates them in an under carriage, thus eliminating beaching difficulties but sacrificing some pay load and speed.

d If an adequate crane is available, hoisting of the seaplane is effected easily and quickly. This method is very suitable for airplane works, where a delay of the maneuver by unsuitable wind conditions is not of the utmost importance.

e Floating docks were used in England during the war and there is one now at the central seaplane base of the German Lufthansa, but it is believed that the initial and maintenance costs are rather high.

f An elevator scaffold or slip could be provided with a second car independent of the rails, so that it might be adjusted to every position of the seaplane due to variations of the wind direction.

g A simple wooden slope lubricated with water is cheap and works perfectly with small machines having a strong keel.

The question of maintenance is rather important, too. Corrosion is still the only serious handicap for the metal seaplane, but the author feels quite sure that there will be some efficient protection developed within the next few years. As a matter of fact, metal construction is imperative for the bigger ships—the DoX would never have been built from wood.

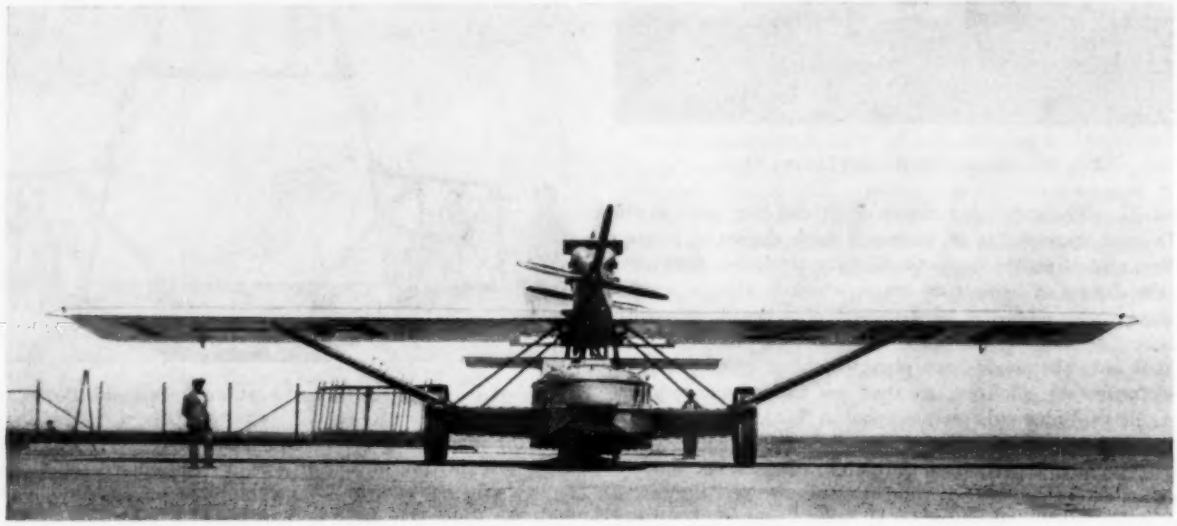


FIG. 20 DORNIER WAL WITH AUXILIARY WHEELS

REQUIREMENTS

As the requirements of the different authorities for the approval of seaplanes for service naturally differ from those for landplanes, a brief glance at them may help to give some idea of the seaplane, the development of which they follow to a certain extent. Further details are given in an article by Mock and Pappel in the *Aero Digest*, September, 1930.

First there are certain requirements for the flight performance. In its regulations now being modified, the Department of Commerce requires a take-off within 1000 ft. This is to hold for seaplanes and landplanes, but considering the higher water resistance

and friction of a seaplane when equivalent to a landplane in the air, it means a serious handicap to the former. So the Department does not actually insist on the regulation. If any requirement of this kind is made, it should limit the time of take-off, as the run is not so easily measured on the water, and this limit should reasonably not be under 45 sec. (see Fig. 7).

The D.V.L., the German Experimental Institute for Aeronautics, has no requirements at all for the take-off of seaplanes, apart from seaworthiness requirements which are now being developed. They acknowledge that the natural limit of seaplanes prevents dangerous overloading, and that the size of the natural landing places does not require limitation of take-off, and they provide for strength by requiring higher load factors for higher landing speeds.[2]

The Department of Commerce wants a certain maneuverability in the air expressed by the radius of figure 8's. The tendency toward equal treatment for landplanes and seaplanes is justified;



FIG. 21 HEINKEL HD 19 ON LAUNCHING CAR

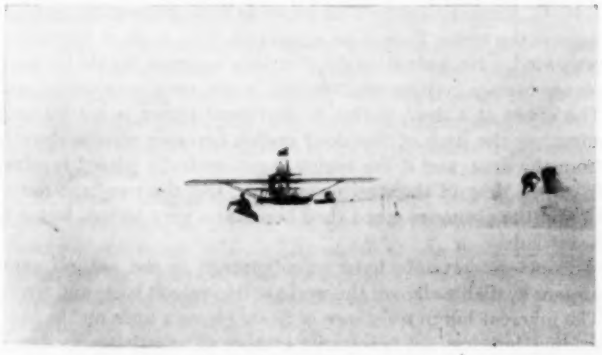


FIG. 22 DORNIER WAL OF AMUNDSEN; PREPARING TAKE-OFF FROM POLAR ICE

once in the air, a plane should be a plane. Certain minor difficulties for the seaplane arise only from special considerations for rudder and ailerons, as already mentioned, or from the influence of floats on the lateral air forces. The author believes that service experiences will also here lead to measuring the time for a circle or figure 8 instead of the radius, this time to be staggered for certain weight groups to take account of increasing inertia.

A third requirement might be new for multi-engined seaplanes: the demand that they shall fly with full load with one engine out of commission. This regulation should be seriously discussed before enforcement. When asking any boat pilot about emergency landings, he will tell you that he feels more comfortable than in other planes as a normal seaplane route is composed of a natural series of "emergency fields" and as an emergency landing of a boat on land will never endanger the lives of those on board. All planes have to face a crash when forced down over woods or mountains. The merits of the flying boat in this direction have been proved, e.g., by the recent landing of a Commodore on Cuba, by emergency landings as reported from amphibions with wheels up, by the take-off of Amundsen's Dornier Wal from the polar ice (Fig. 22), and by the take-offs and landings of a ship of the same type from land covered only with a 2-in. layer of snow when searching for Eielson (Siberia, 1930). Comparing thus the chances of a forced landing, it seems



FIG. 23 HEINKEL HE 8, LONG-RANGE SEAPLANE

to be reasonable to consider the differences by varying the requirements for multi-engined flying boats and for such landplanes.

The reason for the introduction of multi-engined airplanes was to make them more reliable and independent of engine failures. But it would be unreasonable to prevent sound development by barring with restrictions two-engined planes as well,

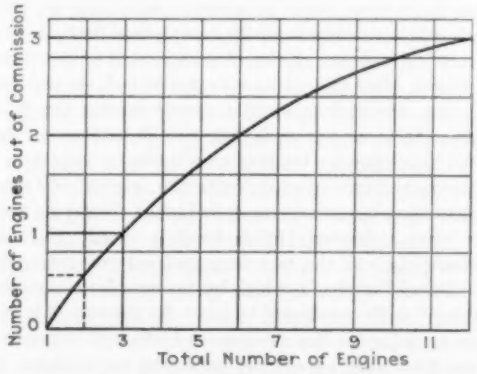


FIG. 24 NUMBER OF ENGINES OUT OF COMMISSION VERSUS TOTAL NUMBER

as one cannot forbid single-engined planes with 0 per cent reserve in the case of engine failure. The complete exclusion of the two-engined seaplane might be a consequence of the regulation that they shall fly with one engine out of commission as its weight empty is already greater than that of the two-engined landplane. Some American airplanes which are at present able to meet this requirement will not be able to do so later, for it is the fate of aircraft to gradually grow heavier and heavier. The author would suggest connecting such a requirement (if any) with a curve (Fig. 24) based on the possible failure of one-third of the engines for a three-engined and a six-engined plane, the ratio decreasing toward zero for the single-engined machine, and again decreasing for more engines as the probability of a failure of so large a number of engines at the same time is very remote. In the case of fractions the corresponding hp. and r.p.m. could be

[2] See reference to later regulations at end of paper.

determined from the propeller load curve of the motor in question.

The different requirements for strength take up much more space in the regulations, but only a few outstanding details will be mentioned here. The D.V.L. has an empirical formula for the landing load factor. It depends on the gross weight, on the 1.5th power of the landing speed, on the angle of keeling, and on the seaway, being 30 per cent higher for seaplanes which will withstand seaway No. 5 instead of No. 3. All important factors are thus combined. The Department of Commerce requires a higher load factor for floats and bracing, whereas the D.V.L. permits a lower one than for the remainder of the plane or for boats. The highest stresses are indeed in the float system, but the D.V.L. thinks that overstresses should first result in bending or deflecting that system without affecting the rest of the plane and the persons.

Generally the D.V.L. derives its requirements from the landing impact. The Department of Commerce, however, has certain requirements referring to floating which are less severe. Buoyancy and floatability are insured by special regulations for the surplus displacement and for the stability with any two compartments flooded according to the D.V.L. wording, the Department aiming at the same goal by requiring subdivision into five compartments of approximately equal size.

There have been no official requirements with respect to seaworthiness until now (they are being developed by the D.V.L.), but there are a number of flying companies with certain regulations, e.g., the German Lufthansa. The corresponding tests comprise drifting, towing on all courses from 30 to 30 deg., taxying of figure 8's, landing and taking off; all this with No. 2 seaway at least and corresponding wind. Such or similar requirements should have the same importance as those for the life-saving devices on steam ships. No private flier should be prevented from drowning if he insists upon doing so, but safety must be the first consideration for commercial flying over sea.

[The new regulations of the Department of Commerce having become known after this paper was preprinted, the author added the following concerning landing speeds during the reading of the paper: It is understood that the Department intends to extend the limitation for landing speeds also to seaplanes. Some details mentioned in connection with the take-off will contribute to the view that landplanes and seaplanes should be treated in different ways, allowing higher landing speeds for seaplanes. The author points to the fact that such an interesting development as that of Rohrbach, which by no means resulted in "dangerous ships," will be excluded by law. By the way, it is wondered how a great many of the modern and efficient American landplanes shall be able to comply with the regulations, as their wing loading indicates landing speeds of 65 to 75 m.p.h.]

Discussion

B. C. Boulton.[3] This most interesting paper covers many phases of a broad field. If one may venture a criticism it is that the author in touching on several subjects just gives enough to excite the reader's interest without satisfying it. For a paper to have the most value to practicing engineers, it should go thoroughly into the matters taken up. Owing to the wide divergence in design practice between this country and Germany and the merit of German flying boats, it is of great importance that designers in this country be acquainted with the methods and formulas used by German engineers in determining the various features of boat design. Because of Mr. Schuettel's familiarity with these, it is to be hoped that in the near future he will contribute a paper or articles giving more detailed information.

[3] Glenn L. Martin Company, Baltimore, Md.

It is a great pity that more basic test data are not available for foreign boats. This information is similar to wind-tunnel data on airfoils, and with it the relative merits of the various bottom shapes and lines shown in Fig. 17 could be determined as regards efficiency. The author's opinion that the sharp V-bottom of the Rohrbach type is more efficient than the American design is questioned.

Figs. 14 and 15, showing the beam and bottom loadings, are interesting and check with the writer's experience. However, it is believed that Captain Richardson's formula for beam in terms of weight and loaded waterline length is more reliable. The Rohrbach boats check closely with this, although they fall far above the curve given for beam loading. Fig. 15 brings out very clearly the necessity for using a much higher unit loading in designing the bottom bracing of large boats than would be satisfactory for smaller ships.

The author touches on a mooted question in discussing the spacing of bottom longitudinals and in pointing out the great difference in British or American and German practice. One at once wonders what gage of bottom plating will permit a stringer spacing of 1.5 to 2 ft. The question also arises of the effect on water resistance of the rather large external longitudinals employed by Dornier.

The suggestion made by the author under the heading of seaworthiness, that boats be divided into two classes depending on the service for which they are designed, is an excellent one. Unquestionably flying boats designed for use on the ocean must be built considerably heavier than those for river or small-lake service. It is believed important enough to consider bringing to the attention of the Department of Commerce so that a differentiation in load factors may be made.

H. C. Richardson.[4] This paper is very interesting, particularly as the writer finds close agreement with most of the matter presented. He feels strongly that long overseas flights by landplanes is only inviting disaster and is not even reasonably safe. The effect of a dead engine in decreased power is not its only effect, as the drag of the dead engine becomes serious directly from the drag, and if the engine is not centrally placed involves increased drag of the steering controls, and the resultant reduction in the optimum speed then becomes a very serious factor in headwinds.

Seaplanes certainly have an advantage in the natural aerodromes available all over the world and in inland lakes and rivers. The inherent hump resistance of floats places a limit on the overloading, but does not necessarily involve an unsatisfactory design commercially. The length of run to the take-off and the time of take-off are relatively unimportant in a harbor, and if any wind is blowing it is always blowing down the runway for a seaplane. In fact, one large landplane specifically built to show high range found that the land-gear tractive effort was a serious factor in limiting the contemplated overload.

Personally it is felt that the triple-float arrangement is superior to the sponson type and that the latter inhibits, to some degree, the employment of sharp V's which are excellent shock absorbers in eliminating the punishment involved in the use of flat bottoms or fairly flat V-bottoms; and, contrary to the generally accepted notion, the writer believes that the intermediate-angle V-bottom does not require power in excess of that required for the flatter bottom types.

For the larger sizes there is no question that the boat type will prevail due to its superior maneuverability and habitability and because twin floats of large size involve such serious wracking forces that the provision for adequate strength involves undesir-

[4] Director of Engineering, Allied Motor Industries, Inc., Chicago, Ill.

able weight. Even for the float type, the central float is superior to the twin float, and particularly with a single engine plane the central float offers protection to the propeller.

The writer is particularly interested in Mr. Schuettel's protest to the Department of Commerce ruling that a twin-engine plane must be capable of flying on one engine. This ruling makes the second power plant with all of its accessories, foundations, controls, etc., a surplus luxury which can ill afford to be carried in commercial types of planes, whereas the presence of the second engine really affords a very measurable improvement in safety in that it very importantly increases the gliding range, and it appears to the writer that the minimum specified rate of descent with one engine out of commission would be a more reasonable requirement for twin-engine airplanes.

Relative to the beam ratio, it is regretted that Mr. Schuettel does not include the length in his table, as the writer has found that, taking account of the waterline length and beam or the beam ratio, the beam required is a function of that ratio and the cube root of the weight, and that this applies to central-float, twin-float, and boat types covering planes from 1000 to 30,000 lb. and for American, German, French, and Italian designs. As to the structure the bottom framing of the Isherwood type will, in the long run, prove to be most satisfactory.

The writer is glad to see a paper of this type as he believes it is time that interest in seaplanes, particularly of the boat type, should be revived in this country.

I. M. Laddon.[5] The author's statement that, in so far as aerodynamic efficiency is concerned, in the smaller sizes the seaplane is inferior to the landplane of equal weight because of the drag of its float and bracing is not agreed with. The fact that the world's high-speed record is held by a seaplane seems to be satisfactory evidence to the contrary. In the larger sizes flying boats have been constructed which, from the standpoint of pay load and performance, exceed comparable landplanes. As a concrete example, the *Commodore*, with two series-B Hornet engines, carries practically the same passenger load that the Fokker F-32 handles, though the latter ship requires four series-B Hornet engines.

The flying boat has the inherent advantage that the hull provides the landing and take-off medium and also functions as the passenger and cargo cabin. The width of the beam necessary for satisfactory take-off assures ample space for comfortable passenger accommodation, a feature which is frequently lacking in the normal land airplane.

The writer does not agree that the maximum range of seaplanes is of the order of 2000 to 2500 miles as compared with 5000 or 6000 miles for landplanes. No one to date has overloaded a modern flying boat to the point where it is just barely able to take off under the most favorable conditions and determined just how far it will cruise. The *Commodore*, loaded to 23,000 lb. gross, would have a cruising range of approximately 4000 miles, which it is believed compares favorably with limiting conditions for the present-day landplane. This type of performance is not advocated, because it does not form the basis of a practicable every-day commercial operation. However, such a parallel should be drawn in the event that flying boats are to be judged on the basis of stunt-performing landplanes.

On the question of constructional features, with particular reference to hull design, our system has been to use a minimum amount of athwartship bracing with the idea that every pound of material used in longitudinal bracing contributes to the overall strength of the hull, whereas athwartship bracing falls in the category of local stiffening, which should be reduced to the minimum consistent with the necessary strength. The hull itself is analyzed in accordance with naval architectural methods, basing external loads on three criteria:

1 Hogging condition with the crest of the wave amidships
2 Sagging condition with the trough of the wave amidships
3 Landing condition with the load concentrated on the bottom forward of the step.

The thickness of plating, spacing of longitudinal stringers, and design of the floor-frame trusses and the keelsons are predicated on this analysis, and while the general assumptions covering external loads can be considered controversial, the structure as a whole has a uniformity of strength which assures reasonable elastic properties and minimum weight.

In general, American flying boats carry a much higher percentage of useful load than those built abroad, and in the writer's opinion the reason for this is that the Continental designers build, first, a rugged, seaworthy hull and then attach thereto sufficient wing area and engine power to permit it to fly. In other words, the average Continental flying boat is primarily a boat and secondarily a flying machine. It is contended that this method of procedure is fundamentally incorrect, i.e., the flying boat should be primarily a flying machine and secondarily a seaworthy boat. The high wing and power loading of foreign designs causes them to be sluggish flying machines, and the great length of run on take-off and high landing speed make it necessary to provide a tremendously strong and correspondingly heavy bottom. If the wing loading and power loading are kept at a reasonable figure, the flying boat will get into the air with a relatively short run in a heavy sea and not be required to take the beating which the more heavily loaded, lower-powered flying boat must undergo. A typical example of foreign flying-boat design consists of two four-engined airplanes which were purchased by an American operating concern each of which had an alleged passenger capacity of 30 persons. Our Department of Commerce would not license these aircraft to carry more than the crew and approximately three passengers. In the writer's opinion European flying-boat development has been greatly overrated as compared with the home product, largely because, with the exception of the *Commodore*, practically all American flying-boat development has been for the Navy and has not received a commensurate amount of advertising and publicity.

Reference seaworthiness, rough-water tests, and thousands of hours of satisfactory operation have proved that the low side float, i.e., one that provides positive lateral stability on the water, is at least the equal of the raised side float for heavy sea duty. Further, this float arrangement assures a degree of taxying control when on the step that is not provided with any other float arrangement.

The reference diagrams in Fig. 17 show characteristic bottom shapes. The *Commodore* bottom does not conform to any of these, but is, in general, similar to that shown for British boats, with the exception of the keel being sharp instead of rounded. Also, the *Commodore* does not utilize sponsons, which are said to be typical for both American and British boats.

The writer is in accord with the author's statement that the monoplane is the desirable type for flying boats. It is believed that the high center of thrust location of the average flying boat is an advantage in so far as stability, power on and off, is concerned, experience indicating that the flying boat is better than the average land machine in this respect.

Disagreement is taken with the author's paragraph on relative engine-wing location. In multi-engined airplanes, where flight with one motor is a performance requisite, the location of the power plants above the wing has an exceedingly detrimental effect. The drag which occurs with an idling propeller is usually

[5] Designer of the Consolidated *Commodore*, Consolidated Aircraft Corporation, Buffalo, N. Y.

of sufficient magnitude to actually prevent horizontal flight with one motor out. A recent series of tests conducted by the National Advisory Committee indicates that the engine location below the wing is preferable.

G. A. Luburg.[6] The writer finds the paper very interesting. It was, however, an article very much on the same order as one that he wrote for the A.S.M.E. last year on "Seaplane Design," in that it was very general in its statements and contained comparatively no new information of a technical nature upon which an engineering discussion could be based.

Author's Closure

Mr. Boulton is perfectly right in his criticism, in so far as it is impossible to crowd experiences on such a wide field into one paper. Besides this, the publication of data will, as long as there is competition, be subject to mutuality, and in this respect Dr. Dornier seems to have been one of the most liberal.

The remark about spacing should be understood generally. The gages on European boats are not excessive, so that the difference seems to lie in a larger allowance for elastic deformation on the unstiffened parts, combined with the use of less but stronger stiffeners. There is some initial water resistance for the Dornier type of sponsons; it disappears quickly, however, due to the rise on the step. Tank tests in this direction by Supermarine should not be generalized; Mr. Richardson is perfectly right when intimating the importance of considering the combined water work of hull bottom and sponson.

The author cannot agree with some of Mr. Laddon's statements. He feels that the development of speedy seaplanes was stimulated by the Schneider Cup, whereas no such stimulant existed for landplanes. The four-engined F-32 and the two-engined Commodore should not be mentioned so closely together as they belong in distinct, different classes, characterized by low power loading and high speed versus medium power loading and low speed. The seating capacity as such does not give a good picture if not quoted together with the corresponding range. A large cabin is of slight value to the operating company if the actual capacity must be considerably reduced for a longer run.

The maximum range for seaplanes has indeed been increased by a Latecoere plane to about 3000 miles. It should, however, be kept in mind that this type of plane was extraordinarily light in weight and that its high stress-to-weight ratio was fatal to several pilots. With the indicated gross weight the Commodore should not cruise much farther than 2500 miles. The limit of overload has been determined for many flying boats, some data being given in Table 1. It is hoped that the range of 4000 miles will follow the introduction of Diesel engines.

The author agrees with Mr. Laddon in rating the transverse stiffening as secondary. Any neglect of it, however, might lead to weakness in respect to torsional stresses. The generalization about European seaplanes is not justified, the principle that flying boats should be airplanes first and then boats is adhered to by most European designers. The four-engined boats referred to had been ordered without any reference to American requirements, which partly did not exist at the time of their construction. It should have been mentioned that their failure was mostly due to an engine type which, in geared condition, had been said to be satisfactory for the tandem arrangement, but which did not prove to be so, according to the author's information.

The author regrets that he did not know Mr. Luberg's paper at the time. The duplication of some statements may give the satisfaction of knowing that seaplanes are working just the same on the other side of the ocean as in this country. It is believed that the data concerning maximum tests and comparing landplane and seaplane take-offs are valuable enough to have justified a paper for them.

---

[6] Assistant Chief Engineer, Curtiss Aeroplane and Motor Company, Garden City, N. Y.

Sheet-Metal Airplane Construction

By PROF. DR. ING. HERBERT WAGNER,[1] DANZIG-LANGFUHR, PRUSSIA

ONE of the first successful attempts to manufacture all-metal planes in Germany was made by the airplane branch of the Zeppelin Works at Lindau, later known as the "Dornier-Metallbauten." This firm had the advantage of the experience of the Zeppelin Works in the use of the light-weight metal duralumin. Entirely new principles of construction were applied in Professor Junkers' "Iron-Monoplane" (Eiseneindecker) built in 1915. Up to that year the covering of airplane wings served essentially the sole purpose of giving them a certain shape and of providing a surface for maintaining the supporting air pressure. The distribution of the stresses was assigned to a special supporting structure which also included the external wiring and struts. Junkers made the external fairing of the wings of metal and, at the same time, combined it with the internal supporting structure so as to form a strong integral supporting body.

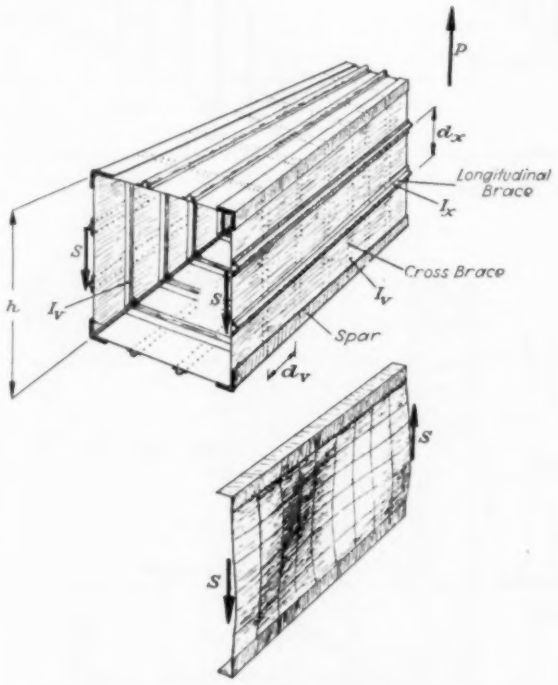


FIG. 1 BOX FUSELAGE
(The lower part of the picture shows the side wall buckled due to shear S.)

Thus the increase of weight of the fairing as compared with a cloth covering was at least partially saved, in the internal structure.

These principles prepared the way for the development of all-metal planes in Germany, and in the course of years various forms and types of such supporting fairings have been developed.

Inasmuch as the use of sheet-metal fairing for support is the outstanding constructional characteristic of such structures, it is the object of this paper to discuss the possibilities of building such sheet-metal walls.

Fig. 1 shows a hollow fuselage consisting of spars and sheet-metal walls. These walls are stiffened by means of braces, made of rolled shapes and riveted to the walls. For convenience the transversely arranged braces will be called "cross braces" and those running in a longitudinal direction "longitudinal braces." Such a fuselage will be subjected to bending stresses by forces exerted by air pressure on the tail plane or by the tail skid in the course of landing. The side wall will therefore be subjected to the shear S, and the top and bottom walls, together with the spars, serve as tension and compression members.

Consider first the strength of such a sheet-metal wall subjected to a shearing stress. With the shear S increasing, the sheet-metal wall finally breaks and buckles diagonally. This, like any other collapse or buckling, happens suddenly with the load increasing only slightly.

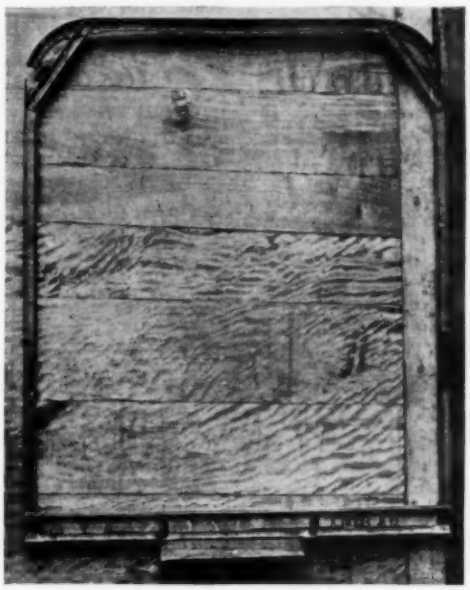


FIG. 2 FUSELAGE FRAME OF A ROHRBACH LAND PLANE
(The bottom beam is fitted with a web securely braced against shear.

In cooperation with Dr. Schmieden[2] the author has made calculations of this problem of strength, the results of which give the basis for the following considerations.

When the walls buckle out the braces are bent, the load S depending on the bending strength of these braces. The numerical value of S may be calculated as follows:

$$S = \frac{33E}{h}\sqrt[4]{\left(\frac{I_v}{d_v}\right)^3 \frac{I_x}{d_x}}$$

[1] Department of Aeronautics, Technical University of Danzig.

Presented at the Fourth National Aeronautic Meeting, Dayton, Ohio, May 19 to 22, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Paper read by J. Otto Scherer, Chief Engineer, Junkers Corporation of America.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] "Das Ausknicken versteifter Bleche unter Schubbeanspruchung," by C. Schmieden, in the *Zeitschrift für Flugtechnik und Motorluftschiffahrt*, vol. 21 (1930), no. 3. A supplementary treatise by Dr. Schmieden and the author will soon be published in the same magazine.

In this formula E is the modulus of elasticity of the material; h is the height of the fuselage or beam, I_v is the cross-sectional moment of inertia of a longitudinal brace, and d_z is the distance between these braces. At a given external load S and for a certain height h of the beam, the moments of inertia of the bracing, therefore, have to be chosen so that

$$\left(\frac{I_v}{d_v}\right)^3 \frac{I_z}{d_z} \geqq \left(\frac{Sh}{33E}\right)^4$$

Assume an external shear S of 10,000 kg. (22,000 lb.), and using the above formula, the problem is to determine which type of construction may be used to best advantage with different beam heights.

First, if this beam is to be constructed without bracing, how small must this height h be? Such a beam may be represented by the floor-beam of a Rohrbach land plane, as shown in Fig. 2. The value of the shearing stress τ at which such an unbraced sheet collapses and buckles diagonally has been determined by theoretical calculations as well as by experiments. This value (see Fig. 3) depends on the ratio of the distance d between the bracing members to the thickness s of the metal sheet. For all values of d/s higher than about 50, τ can be calculated from the following formula given by Southwell-Skan:

$$\tau = \frac{5E}{(d/s)^2}$$

For $d/s = 50$ the buckling stress of duralumin is about equal to the yield point (approximately 1500 kg. per sq. cm. = 21,330 lb. per sq. in. shearing stress) and consequently its increase is not worth considering even with smaller distances between the bracing members. To get a light structure, i.e., to subject the sheet to the highest practically obtainable stress, the distance between the bracing members must not be made greater than 50 times the thickness of the sheet.

If, now, the beam with a shear load of 10,000 kg. (22,000 lb.) has a total height of 22 cm. (8.66 in.) and a height of about 15 cm. (5.9 in.) between the spars, then a desired shearing stress of 1500 kg. per sq. cm. (21,330 lb. per sq. in.) will call for a thickness of about 3 mm. (0.118 in.) so that the sheet is sufficiently braced to resist buckling. For such beams, and of course for beams of smaller height as well, special bracing can be eliminated.

Considering, however, a beam with a height of 33 cm. (approx. 13 in.) with the same shearing stress of 1500 kg. per sq. cm. (21,330 lb. per sq. in.), the sheet need only be 2 mm. (0.079 in.) thick, while the distance between the bracing members must not exceed 110 mm. (3.95 in.) to give the sheet sufficient resistance. As the spars are already too far apart in this case, bracing must

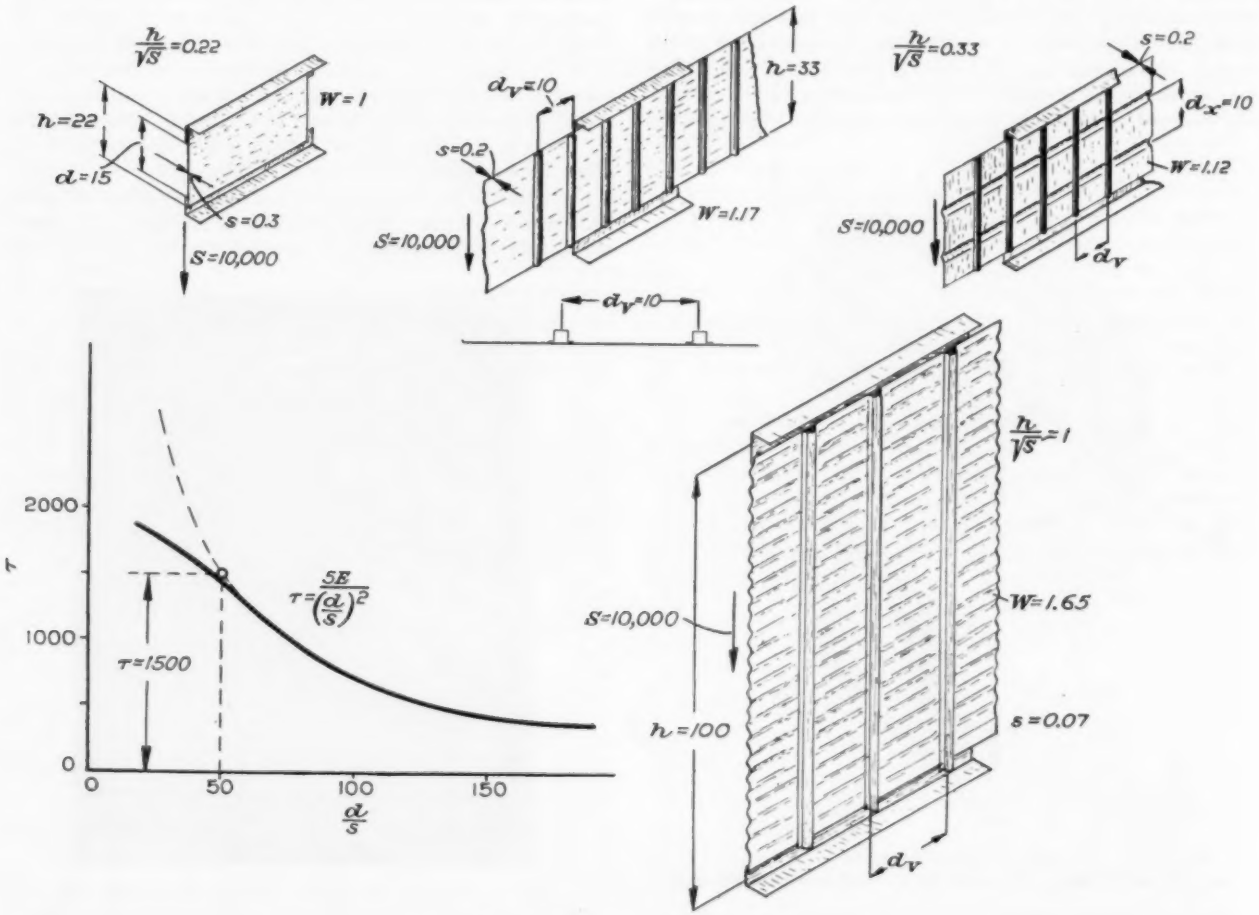


FIG. 3 DETAILS OF SIDE WALL

(In the lower left-hand corner the shearing stress τ is given, under which plane sheets with a wall thickness s buckle, if the distance between the bracing members is d. The other illustrations show the most favorable forms of design of sheet-metal walls securely braced against shear, dependent on the value of $h/\sqrt{S}$. W indicates the ratio of the weight of the sheet wall (exclusive of the spars) to the weight of a sheet secure against shear (exclusive of the weight of the bracing), so that the figures after the decimal point give approximately the comparative weight of the bracing. The weight of the sheet only is, at a given shear load, approximately the same for all heights of the beam.)

be provided. If, for example, cross-braces are chosen, the moment of inertia of these braces has to be determined from the formula previously given, in which the resistance to bending of the sheet proper, namely, the value $s^3/12$, must be inserted for I_z/d_z.

$$\left(\frac{I_v}{d_v}\right)^3 \frac{s^3}{12} = \left(\frac{Sh}{33E}\right)^4$$

If such cross-braces are chosen as are customarily used for airplanes, it will be found that they can be very light, their weight amounting to about 17 per cent of the weight of the metal sheet. Bracing the sheet also in a horizontal direction, most easily and simply accomplished by corrugations or by the use of corrugated sheet metal, would result in an additional reduction of the weight of these profiles. By these means the moment of inertia in horizontal direction is materially increased so that now the stiffness of the cross-braces may be decreased. Then the weight of the cross-braces amounts to only about 12 per cent of the weight of the metal sheet.

At the given shear load of 10,000 kg. (22,000 lb.) the thickness s of the wall is decreasing more and more with increasing height of the beam. If no special longitudinal braces are provided the moment of inertia I_v of the cross-braces would, according to the equation, have to increase disproportionately fast in order to provide sufficient resistance to buckling. For beams of such a height only corrugated sheet is used. Then the weight of the

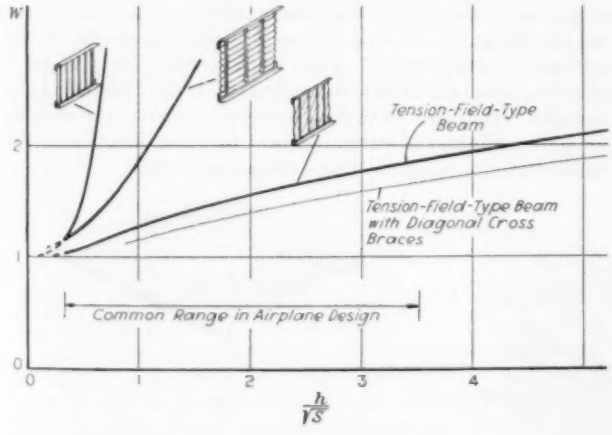


FIG. 4 RELATION BETWEEN W AND $h/\sqrt{S}$

(The ratio W of the weight of the sheet wall, without spars, to the weight of a sheet secure against shear, exclusive of the weight of the bracing, is plotted against the factor $h/\sqrt{S}$ for various designs of sheet-metal beams (plane sheet metal securely braced against shear by cross-braces; corrugated sheet with longitudinal corrugations and braced by cross-braces; tension-field-type beams with vertical cross-braces; tension-field-type beams with cross braces arranged at an angle of 120 deg. to the spars). In this sheet, which is secure against shear, the allowable shearing stress τ is equal to 1500 kg. per sq. cm. (21,300 lb. per sq. in.). See also Fig. 3.)

cross-braces rises only slowly with increasing height of the beam, amounting to about 65 per cent of the weight of the metal sheet for a beam 100 cm. (39.37 in.) high, which should be 0.7 mm. (0.0276 in.) thick.[3]

These examples show how the design of beams with various heights may be determined when they are subjected to a shear of 10,000 kg. (22,000 lb.). These most favorable types of design also hold true for any other values of shear as long as the value $h/\sqrt{S}$ is the same as in these examples. This can easily be understood from the law of similarity[4] as applied to the science of the strength of materials.

In Fig. 4 W, the ratio of the weight of the sheet wall, exclusive of spars, to the weight of the sheet secure against shear, exclusive of bracing, is shown to rise with increasing value of $h/\sqrt{S}$.

The values of $h/\sqrt{S}$, which are common in airplane design, vary between about 0.3 and 4 cm. kg.$^{-1/2}$. For the side wall of a fuselage having a height of 200 cm. (78.74 in.) and which has to resist a shear of 3600 kg. (7920 lb.) the value of $h/\sqrt{S}$ will be 3.3. As, for such high values of $h/\sqrt{S}$, the added weight of the bracing would be quite considerable, the construction is modified by making some of the cross-braces especially stiff (see Fig. 5).[5] Such strong frames are also often necessarily employed for other

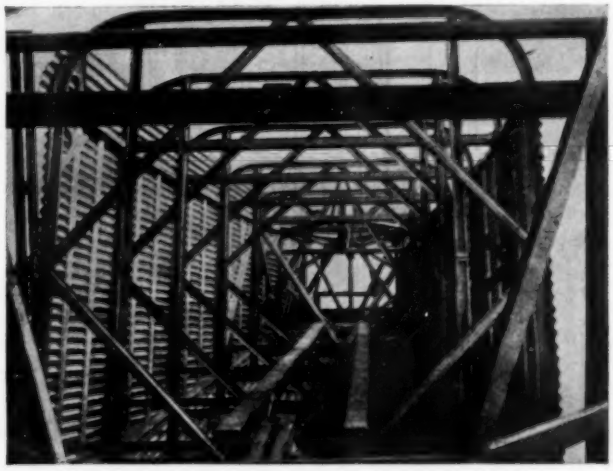


FIG. 5 INTERIOR OF THE FUSELAGE OF A JUNKERS AIRPLANE, SHOWING THE STRONG FRAMES, BETWEEN WHICH THE CORRUGATED SHEET IS STIFFENED BY BRACES

purposes, as, for instance, in the passenger cabin for taking up the loads.

The strength of such a construction can likewise easily be determined. (See Fig. 6.) According to well-known principles a shearing stress in a vertical direction is always accompanied by an exactly equal shearing stress in a horizontal direction. The sheet-metal wall between two strong frames, therefore, has to be considered as a beam with the height d, which has to transfer the horizontal shear $\left(S\frac{d}{h}\right)$ from the bottom spar to the top spar; and the corrugated sheet with its bracing must be dimensioned accordingly.

Thus far the resistance of braced sheet-metal walls to shearing forces has been considered. There follows now a discussion of the resistance to compression forces in the direction of the spars.

Consider, for instance (Fig. 7), the top of a square fuselage subjected to the stress set up by an elevator reaction, which produces compression stresses in the upper sheet-metal wall and the upper spars. If the upper sheet-metal wall were not sufficiently braced it would buckle as indicated in Fig. 7. The pressure X, at which this occurs, may be comparatively easily calculated as follows:

[3] The weight of the sheet wall is, of course, also dependent on the kind of corrugations. In the case under consideration, corrugations having a depth of 15 mm. (0.59 in.) have been assumed. For the same reason the curve for corrugated sheet in Fig. 4 is true only for certain dimensions customarily used for airplane design. The curve of the value W for tension members, however, is only slightly dependent on the cross-sectional form of the cross-braces.

[4] For details see Professor Wagner's treatise, "Einige Bemerkungen über Knickstäbe und Biegungsträger," in the *Zeitschrift für Flugtechnik und Motorluftschiffahrt*, vol. 19 (1928), no. 11.

[5] The illustrations of the Junkers designs have been taken from Langsdorff's book "Fortschritte der Luftfahrt," by courtesy of H. Bechhold, Frankfurt am Main.

$$X = \frac{20E}{h}\sqrt{\frac{I_v}{d_v}\frac{I_z}{d_z}} + \frac{10G}{h}\frac{T_v}{d_v}$$

From this equation the relations of the weight of the material and the height and pressure, i.e., $h/\sqrt{X}$ may be derived; these relations being quite similar to those assumed when the effect of shear was studied. A brief discussion of the equation will suffice. The first half of the equation contains the moments

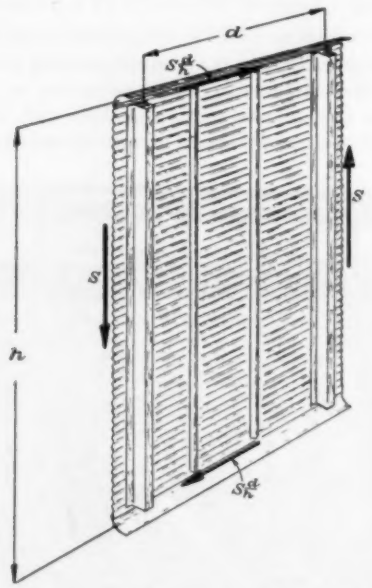


FIG. 6 THE DIRECTION OF FORCES IN THE SHEET-METAL BEAM OF FIG. 5

of inertia of both bracing members, both factors being of the same power. In the second half of the equation GT_v is the resistance of a cross-brace to torsion. This half of the equation may be neglected if the cross-braces have so-called "open sections" (see Fig. 7), which have little or no strength in torsion. Cross-braces, however, built up of closed (tube-shaped) profiles or which together with the corrugated sheet form a channel-like cross-section, as shown in Fig. 7, have considerable resistance to torsional stress. A numerical evaluation makes it apparent that the compression strength of the wall is thereby approximately doubled. At all events, closed profiles are to be preferred for corrugated sheet-metal structures.

An examination of the equation furthermore shows that even when using comparatively heavy cross-braces and coarsely corrugated sheet the compression strain in the corrugated sheet metal lies, in most cases, well below the yield point (about 2600 kg. per sq. cm. = 36,972 lb. per sq. in. for duralumin), which means that the material may not be used to its full advantage. These difficulties can be evaded by curving the surfaces as in Fig. 8. Although numerical data on this type are not available it is certain that the resistance to compression forces is considerably increased.

So far sheet-metal walls which are braced against forces of shear and compression have been discussed. The following discussion covers another type of design developed years ago by the author when he was with the Rohrbach Metallflugzeugbau and described in 1929.[6] The present discussion is limited to the most essential points.

Fig. 9 illustrates a sheet-metal wall, consisting of upper flange, lower flange, a thin sheeting, and cross-braces subjected to a shear S. At a very small load the thin sheet will buckle diagonally.

It has been shown that the sheet may be prevented from buckling by arranging cross-braces very close to each other. What will happen if the distance between the bracing members is so large that the sheet may buckle almost unhindered?

After the first buckling the load may be considerably increased, for example, a hundred or thousand times, without collapsing the beam or rendering the surface excessively uneven. The wrinkles in the surface may attain a depth of perhaps 3 mm. (0.118 in.) and a width of perhaps 100 mm. (3.94 in.), thus representing only very slight corrugations.

Fig. 10*a* shows a square sheet of very thin metal, which is assumed to be as thin as paper. This may be wrinkled (Fig. 10*b*) without applying much force, thereby somewhat diminishing the edge-to-edge distance a. Tensile stresses acting at the upper and lower ends of the sheet in the direction of the wrinkles (Fig. 10c) will not change the direction of these wrinkles, though their depth will be somewhat reduced by the lateral contraction. There is no force acting in the sheet perpendicularly to the direction of the wrinkles. The direction of the greatest elongation of the sheet is, for such conditions of stress, identical with the direction of the tensile strain, thus representing a single-axial strain.

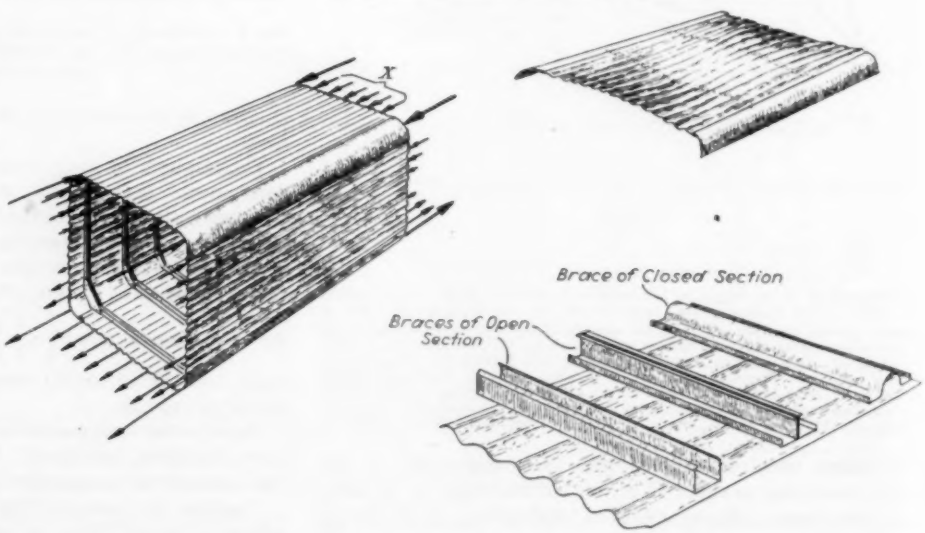


FIG. 7 FUSELAGE UNDER A STRAIN OF FLEXURE
(The upper right-hand corner illustrates the buckling of the upper sheet-metal wall due to the shear S. The lower right-hand corner shows braces of open and closed sections.)

Attention is called to a phenomenon of disturbance caused by the fact that the edges are not free, but are riveted to braces. It will be noted from Fig. 11 that the edges themselves cannot

[6] "Ueber ebene Blechwandträger mit sehr dünnem Stegblech," by Herbert Wagner. *Zeitschrift für Flugtechnik und Motorluftschiffahrt*, vol. 20 (1929), nos. 8 to 11. See also the Jahrbuch der Wissenschaftlichen Gesellschaft für Luftfahrt, 1928, p. 113.

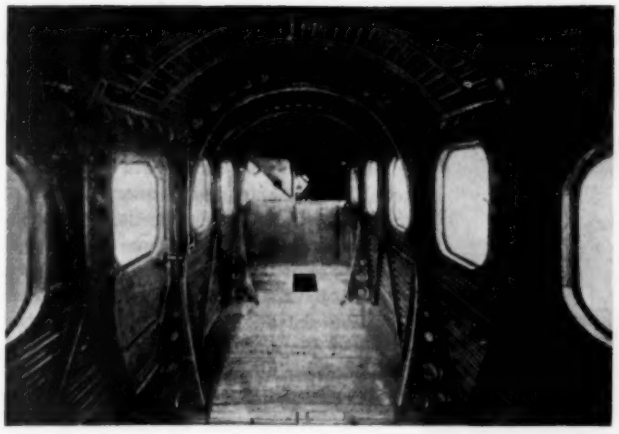


FIG. 8 INTERIOR OF THE FUSELAGE OF A JUNKERS AIRPLANE
(The upper surface is curved, which increases the resistive strength against forces of compression. The braces can be clearly distinguished.)

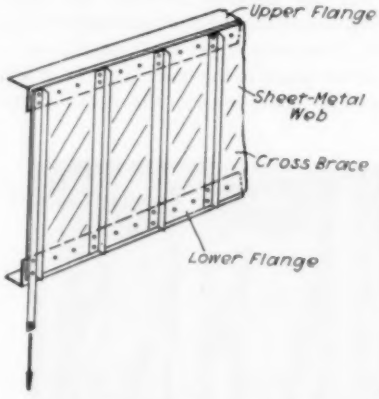


FIG. 9 TENSION-FIELD-TYPE BEAM

not change the direction of the greatest elongation of the sheet or the direction of the wrinkles, which remains 45 deg. This means that the diagonals are not running from I to II (see Fig. 12*d*), but at an angle of 45 deg., their direction being, in the case of rigid rods, independent of the distance between the rods.

Fig. 13 et seq. illustrate a few experiments with "tension-field-type" beams made several years ago at the Rohrbach Works.[7] Fig. 13*a* shows a sheet-metal beam supported at the right end and carrying a load of shot-bags by means of levers. Upper flange, lower flange, and the thin sheet-metal wall can easily be recognized. The position of the cross-braces on the back of the beam is recognizable from the riveting. All dimensions of the experimental model are to scale.

The tension wrinkles are distinctly visible and appear rather uniformly over the whole sheet. The shear in the beam increasing to the right, because of the load on the different rods, the wrinkles in the right part of the sheet are more strongly formed than those in the left one. The difference, however, obviously is not great. The load shown on this picture is far beyond that at which the first buckling of the sheet occurred. This initial buckling could be caused by slightly pressing one of the levers by hand, i.e., by a shear of very few pounds. The

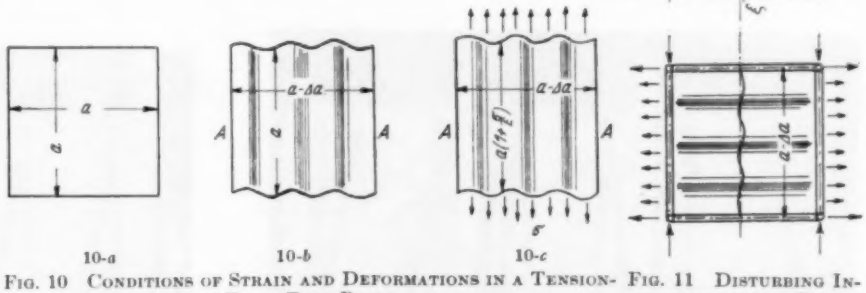


FIG. 10 CONDITIONS OF STRAIN AND DEFORMATIONS IN A TENSION-FIELD-TYPE BEAM

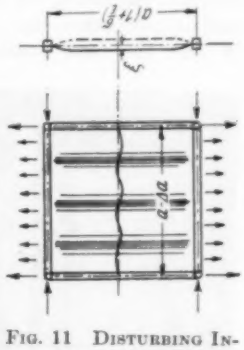


FIG. 11 DISTURBING INFLUENCE OF THE EDGES

wrinkle. It can, however, easily be shown that this disturbance is of no importance to further considerations.

A simple but important practical example may be illustrated by Fig. 12. Fig. 12*a* shows a frame formed by four rods which are to be pin-jointed to each other and are assumed to be perfectly rigid. A thin sheet is to take up the shear. Because of this shear S, shearing stresses τ will become effective within the sheet metal. The direction of the main stresses is at an angle of 45 deg. to the direction of these shearing stresses. The main stress σ_1 sets up a tensile strain, its direction being identical with the direction of the greatest elongation of the sheet metal. The other main stress σ_2 causes a compression strain. If the sheet is very thin it will soon buckle under the effect of the compression strain and wrinkle in the direction of the tensile strain, σ_1 (Fig. 12*b*). With further increase of the shear S the tensile strain σ_1 will rise rapidly until, at very high loads, it will take up nearly all the shear. Thereby the field has buckled uniformly. Such a field is called a "tension field."

The value of the tensile strain σ_1 can easily be calculated as follows:

$$\sigma_1 = \frac{2S}{as} = 2\tau$$

The equation shows that the tensile strain is twice as great as would be the shearing strain, if the sheet would withstand shear.

Adding (see Fig. 12*c*) another equally rigid rod to the four rods and pin-jointing the new one to the rods O and U naturally would

wrinkles shown in this illustration do not represent permanent deformations. They disappeared entirely after the load had been

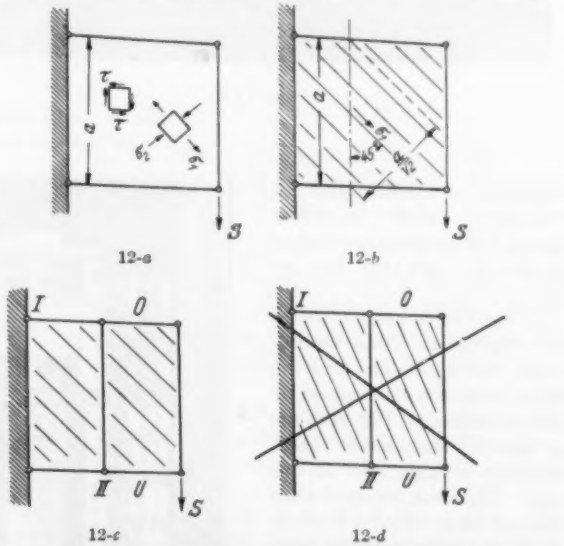


FIG. 12 THIN SHEET WITH RIGID RODS AT THE EDGES AND LOADED BY A SHEAR S

[7] On the occasion of a lecture delivered in Danzig, Dr. Rohrbach permitted the author to publish the pictures of these tests.

removed. Permanent deformations appear only when the tensile strain in the sheet has nearly reached the yield point. This is hardly influenced by the local transverse strains which are caused by the formation of the wrinkles.

Fig. 13*b* shows the beam from the back, under an appreciably higher load. The tension field, indicated by the wrinkles, stands out very distinctly. The angle between the spars and the direction of the wrinkles, which, as stated, is 45 deg. in the case of perfectly rigid rods, is in this case somewhat smaller on account of the flexibility of the cross-braces. It is about 40 to 42 deg., which is in accordance with the calculation.

At the load applied in Fig. 13*b* the yield point of the sheet has already been somewhat exceeded. It is of great importance to the simplicity of calculation that the conditions of the stress in the sheet and the braces are not essentially changed even when the yield point in the web has been exceeded. On the contrary, the assumption of negligibly small transverse strength of the sheet now proves to be even more true than before.

In Fig. 13*c* the beam is shown after the break of the vertical rods, the allowable load on which was being determined by this experiment. Fig. 14 shows a similar sheet-metal beam carrying a single load on one end. The break occurred when the web tore. Such a tension-field-type beam may be compared with a framework having cross-braces placed in the same direction and diagonals at the same angle of 40 deg. as that of the tensile strains in the web of the beam (see Fig. 15). If, for the diagonals in a framework, straps are used of such a width that two adjacent ones just touch each other, and provided that they are of the same thickness as the sheet of the beam, these straps will be stressed to the same extent as the sheet.

In a tension-field-type beam the spar of each field between two cross-braces (see Fig. 16) is bent toward the sheet metal by the tensile strains in the web acting at the spars. The cross-braces which support both spars against these forces are subjected to compression loads which tend to cause buckling. Any of these stresses can be calculated very easily. The results of such calculations are, in accordance with the results of the tests, the following:

If the distance between the cross-braces is chosen within reasonable limits, such as being equal to about $1/6$ to $1/3$ of the height of the beam (i.e., the length of the cross-braces), the bending moments in the spar caused by the pull of the sheet have practically no effect upon the strength of the spars. On the contrary, the resistance of the spars to breaking strain is very high because of the comparatively close arrangement of the cross-braces which support the spars.

Furthermore, the deflection of the spars between two cross-braces on account of this pull of the sheet is so small that the spars may be assumed to be rigid. If great deflections of the spars were to occur the tensile strains in the web would become unequal and the direc-

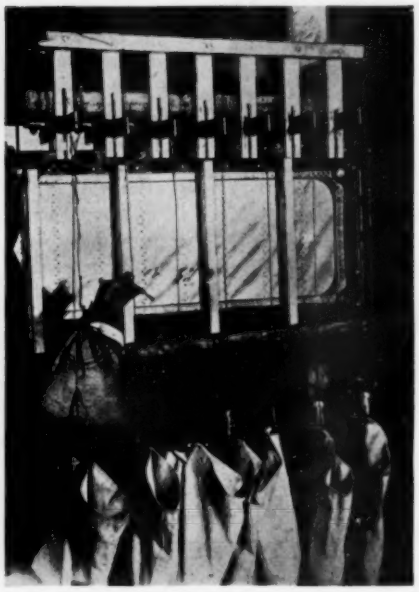


Fig. 13-*a*

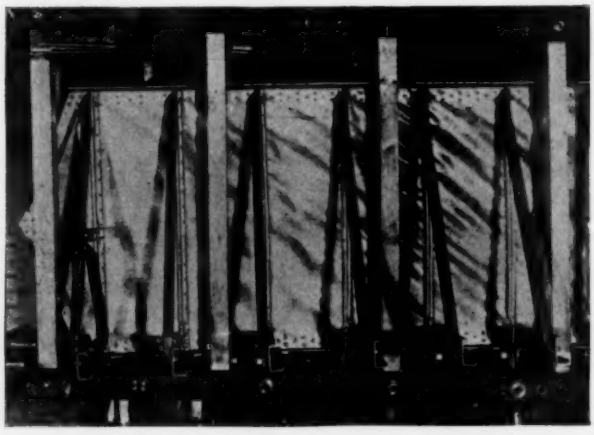


Fig. 13-*b*

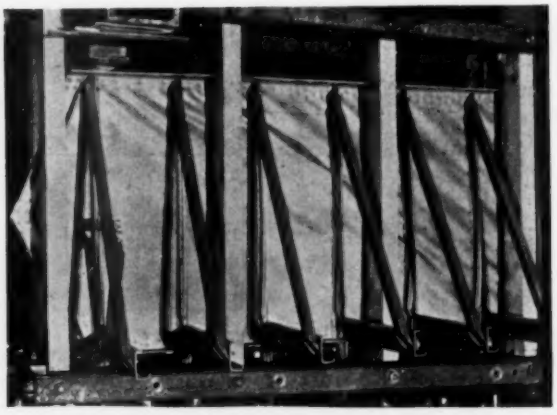


Fig. 13-*c*

FIG. 13 A TENSION-FIELD-TYPE BEAM UNDER LOAD
(The break occurs after buckling of the cross braces.)

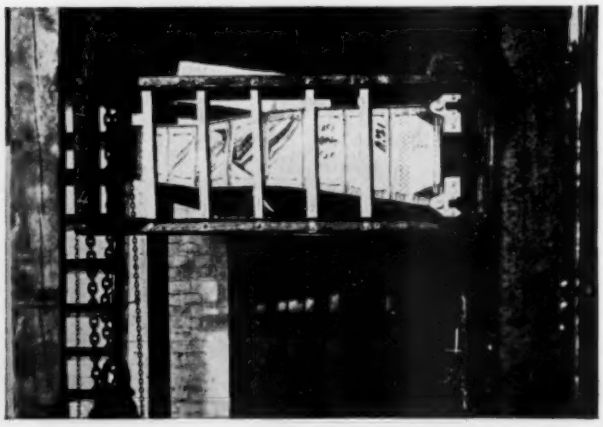


FIG. 14 A TENSION FIELD UNDER LOAD
(The break occurs when the web tears.)

tions of the wrinkles and tensile stresses respectively would no longer be parallel. In accordance with the calculation, such things, however, do not happen. Yet should this happen to a certain degree, due to the weakness of the spars, this problem could be checked comparatively easily by calculation, at least as far as this is of practical importance.

This unexpectedly high stiffness and resistance of the spars to bending forces is an essential fact. Thus, for instance, as the author subsequently heard, a Mr. Rode, in referring to deliberations of American engineers, called attention, in the Austrian magazine, *Der Eisenbau*, in 1917, to the observation that a web, after buckling, could still transmit forces by resisting tension. But, as Mr. Rode fears the insufficient transverse strength of the

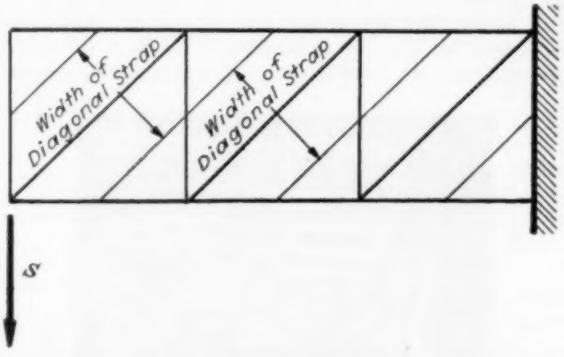


FIG. 15 COMPARING A TENSION-FIELD-TYPE BEAM WITH A FRAMEWORK BEAM

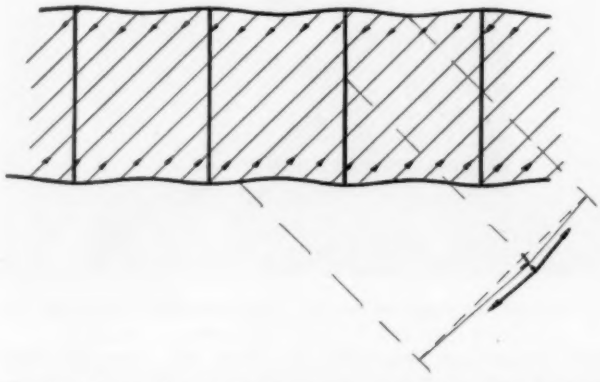


FIG. 16 SPARS AND CROSS-BRACES OF A TENSION-FIELD-TYPE BEAM UNDER STRAIN
(The strained sheet complicates buckling of the cross-braces.)

spars, he consequently does not conclude that such a design can actually be carried out to advantage.

It has been shown that the cross-braces are under compression strains. (See Fig. 16.) They must, therefore, be stiffened to prevent buckling. However, this stiffness can be comparatively small since the strained sheet metal holds them in place. For when a cross-brace buckles and moves out of the plane of the sheet metal it must also move the sheet to which it is riveted. Thereby the tension lines of the sheet suffer a displacement at the cross-brace with the result that the reaction of the stress lines against the cross-brace exert a force which is opposed to the buckling force. This essentially increases the resistance of the cross-braces to buckling stresses. This resistance has been calculated by the author. Depending on the distance between the cross-braces, it is about 4 to 7 times as great as Euler's buckling load for unsupported columns.

The cross-braces may also be arranged at an angle to the spars (see Fig. 17), in which case, under certain conditions, the member may be built lighter. Arranging the cross-braces at an angle of 120 deg. to the spars will have the least weight. The total weight of the web plus the cross-braces is thereby decreased by 15 per cent as compared with a sheet-metal wall having cross-braces arranged perpendicularly to the spars. However, this advantage can be used to the full only when the external shear S_1 acts essentially in one direction only. For this beam with diagonal cross-braces is considerably less resistant to the shear S_2 in the opposite direction. The shear S_2 in the direction shown must not exceed a value equal to about $^1/_3$ of the shear S_1.

In Fig. 4 a curve has been plotted which represents the weight required for such tension members. It will be noticed that

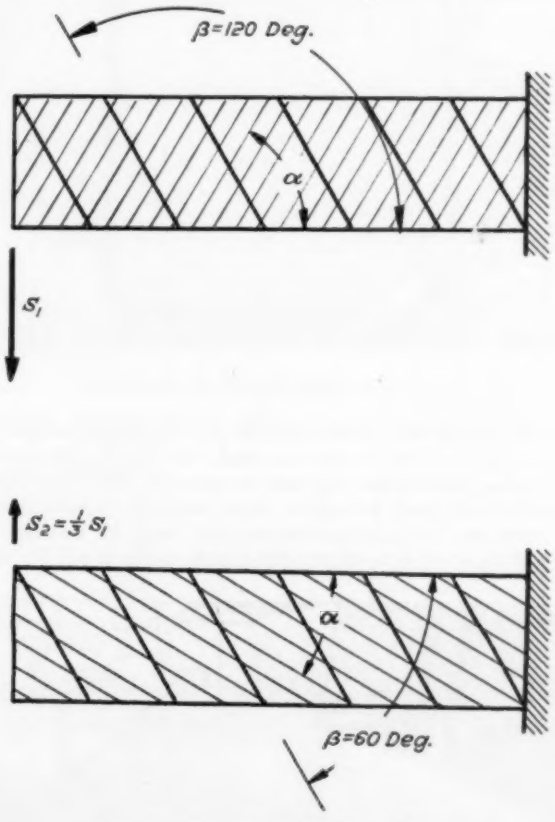


FIG. 17 EFFECT OF REVERSING LOAD
(The direction of the wrinkles [tensile strains] is changed if the external load S is reversed. The direction of the wrinkles approximately bisects the angle between the spars and the cross-braces, $\alpha = {}^1/_2\beta$.)

beams of great height and comparatively small shear, i.e., with high values of $h/\sqrt{S}$, are essentially lighter, for example, than corrugated sheet metal amply stiffened against buckling. Since these high values of $h/\sqrt{S}$ are frequently encountered in airplane design these tension-field-type beams are particularly suitable for airplane design.

Fig. 18 shows a frame of a Rohrbach seaplane, whose bottom beam has been designed as a tension-field-type beam. Figs. 19 and 20 show the hull, the walls of which represent tension-field-type beams.

The weight requirements for sheet-metal beams (Fig. 21), which must resist, in particular, longitudinal forces in addition to the shearing forces, will now be discussed.

In a fuselage made up of sheet-metal walls which are braced to resist buckling not only the spars but also the corrugated sheet-metal sides resist longitudinal forces. It should be observed

that as the spars used at the edges of the fuselage are directly restrained against buckling in both directions, they can resist a much higher compression stress than can the sheet-metal wall which is held against buckling only by comparatively easily bending beams, namely the cross-braces. Naturally the permissible compression strain depends on the weaker member of the structure. Thus, on the one hand, the force in the spars of this type of construction decreases because the sheet metal takes up part of this force, and on the other hand, the cross-section of the spars has to be made comparatively large for this smaller force, if the sheet is to be prevented from buckling out and sustaining greater and permanent deformations before the calculated breaking load is reached.



FIG. 18 FRAME OF A ROHRBACH SEAPLANE

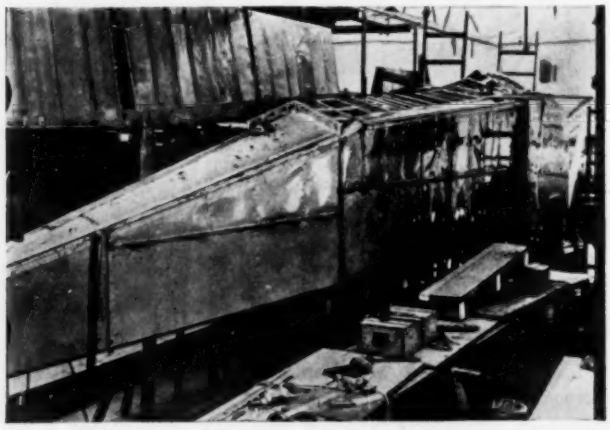


FIG. 19 HULL OF A ROHRBACH SEAPLANE
(The surface walls are designed as tension-field-type beams.)

In the tension-field-type beam the sheet metal buckles under a very small compression load and therefore does not resist compression loads to any appreciable extent. But as this buckling of the thin sheet metal takes place without appreciable strain and no permanent deformations remain after removing the load, the spars may be loaded as highly as warranted by the fact that they are supported by the adjacent walls of the fuselage. Though they must resist all of the longitudinal force in the tension-field-type beam they may, nevertheless, under certain conditions, be lighter than the spars of the sheet-metal beam which is securely braced against shear and compression.

Consider now the stresses in a wing of the Junkers type such as is shown in Fig. 22. The corrugated sheet metal, the corrugations of which are in the direction of flight, is fastened to strong spars that resist the bending moment. The torsional moment appearing during a vertical dive puts the corrugated sheet under a shear load. The sheet, therefore, has to be braced between the spars by longitudinal braces. These braces, running parallel to the spars, can be noticed in Fig. 22. How do these longitudinal braces behave under compression forces when the wing is bent?

The longitudinal braces are held by the corrugated sheet metal against buckling out of the plane of the sheet wall and, the cross-sectional area of the longitudinal braces and consequently their compression load being very small, the corrugated sheet metal is such an effective support that the permissible compression strain in the longitudinal braces is hardly lower than that in the spars. This can easily be proved with the help of the equation previously given for the compression forces in such sheet-metal walls.

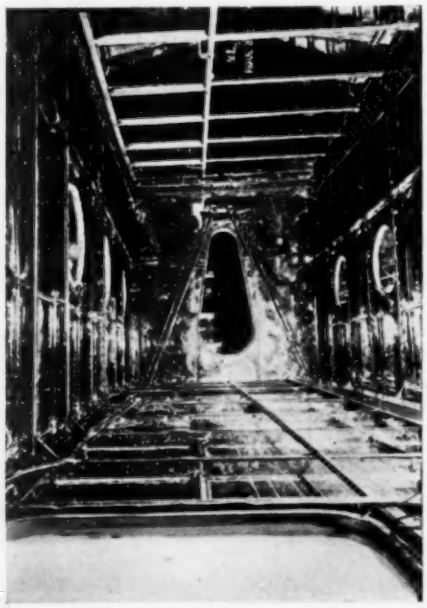


FIG. 20 INTERIOR OF THE HULL OF A ROHRBACH "ROMAR"

We have thus arrived at a result quite similar to that obtained for the tension-field-type beam, i.e., the sheet-metal wall serves only to resist the shearing forces. The spars resist nearly all the longitudinal forces, but, on the other hand, may be loaded as highly as conforms to their being well supported in two planes.

Fig. 23 shows a Junkers wing which has the spars spaced so closely that the corrugated sheet need no longer be stiffened by special cross-braces.

It has been shown that sheet-metal beams may either be fitted with sheet metal which is securely braced against shear or be designed as tension-field-type beams. A warning should be issued against the middle course. If a web is made with corrugations, as shown in Fig. 24, and if the distance between these corrugations is made larger than would conform to the optimum strength of the plain, smooth part of the sheet between the corrugations against shear, then this part of the sheet will buckle

and exert tensile forces on the corrugations. This stretches out the corrugations, and ugly bumps or protrusions will remain in the sheet when the load is removed. Even in the case where special diagonal bracing is used to resist the shearing forces, the distance between the corrugations must be chosen cautiously.

Thus far the structural weights of different sheet metal constructions have been compared with each other. There remains the comparison between the weight of these constructions and that of fabric-covered structures. This comparison will give different results, depending on the size of the airplane. Since the external forces exerted by the air on the plane and, consequently, the shearing stresses in the sheet-metal beams are rising at least with the square of the lineal dimension, the thickness of the sheet required for taking up the shear increases at least in direct proportion (linear) to the size of the plane.

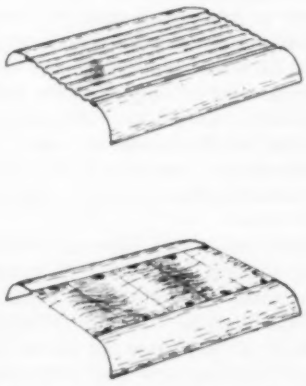


FIG. 21 SHEET-METAL BEAMS AND TENSION-FIELD-TYPE BEAMS SECURELY BRACED AGAINST SHEAR AND COMPRESSION AND UNDER A LOAD OF COMPRESSION FORCES ACTING IN DIRECTION OF THE SPARS

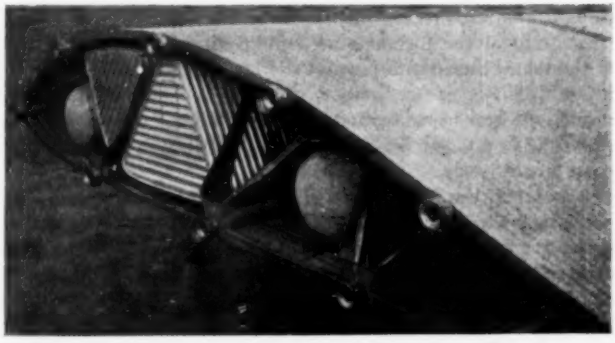


FIG. 22 INTERIOR OF THE WING OF THE "BREMEN" (JUNKERS W 33)
(Note the longitudinal bracing running parallel to the spars.)

For small airplanes the required thickness of the sheet is so small that the sheets must be made thicker than this calculation would demand in order to make them secure against accidental local stresses. Moreover, sheets less than 0.3 mm. (0.0118 in.) thick are difficult to manufacture in sufficiently large sizes and such a sheet-metal construction will, under these conditions, of course, be heavier than a small fabric-covered plane. It should be added that for small and medium-size airplanes, corrugated sheet metal appears to be preferable to smooth sheet metal because it has a higher resistance to local stress and because very thin plane sheets are easily deformed in riveting.

However, for larger airplanes the strength in shear is the deciding factor as far as the thickness of the sheet metal is concerned. Such fully utilized sheet beams are, at all events, lighter than a framework, if they are, for example, built as tension-field-type beams. For heavily loaded, very large planes, therefore, the metal-covered type may be the most favorable solution in so far as weight is concerned. However, those parts of the wings, even of large planes, which are under little strain, particularly in the neighborhood of the trailing edge, may be covered with fabric, as it has been done, for example, on the huge Dornier seaplanes "Superwal" and "DoX."

Fabric-covered airplanes have been compared in this paper with metal-covered ones only with regard to their respective weights. The other advantages and disadvantages are so well known that only brief mention is necessary.

Metal-covered planes require a greater amount of work in the design room and the shop and are a little more difficult to repair. The construction of the wings of very small planes is particularly difficult.

Metal-covered wings have a greater air resistance than those

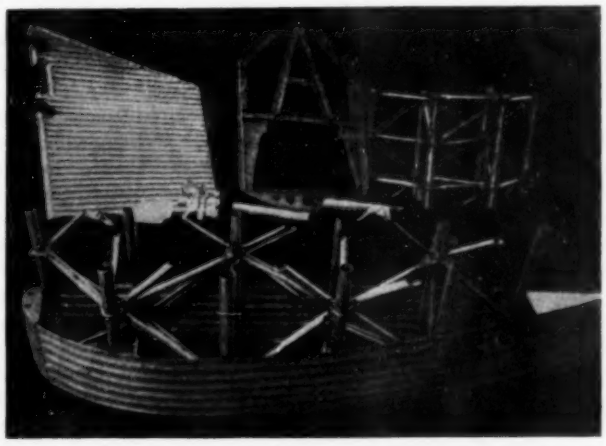


FIG. 23 JUNKERS WING
(The spars are arranged so closely that the corrugated sheet is secure against shear even without special bracing.)

covered with fabric. This holds true not only for corrugated-sheet-metal but also for smooth-sheet-metal covering, as this deforms under the strain of the air pressure and because the frictional resistance is increased on account of the rivets and especially bracing placed on the outside. The advantages of metal-covered airplanes are their durability and lower fire hazard. Very large all-metal airplanes are lighter than fabric-covered ones. The use of wood for pontoons as compared with the use of metal has the disadvantage of water-logging and thereby increasing the gross weight.

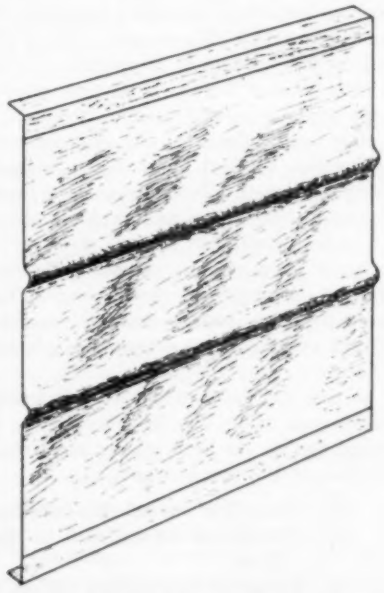


FIG. 24 SHEET-METAL BEAM WITH LONGITUDINAL CORRUGATION
(If the distance between the corrugations in a plane sheet is too great the sheet will buckle between the corrugations and exert tensile stress on the corrugations in a manner similar to the tension field. These are thereby stretched out.)

An attempt has been made in this short paper to survey the most important problems of strength encountered in the design of metal-covered airplanes. The discussion has been restricted to plane sheet-walls, because the problems of strength of curved sheet-walls have not yet been settled entirely.

Discussion

COL. V. E. CLARK.[3] The paper is most interesting and valuable at this time because of the rapidly increasing use of duralumin and "Alclad" monocoque construction in this country.

It is believed that there is at present more need for experimental data on the sheer strength of thin plate girders and reinforced shells than on any other phase of airplane structure. In a way the use of the term "sheer" is an unfortunate habit among engineers as it tends to obscure the fact that shear in the web of a beam is a combination of diagonal tensile and compressive forces acting at right-angles to each other. A thin web without vertical stiffeners is buckled in compression at a comparatively small stress.

In the case of a girder employing vertical stiffeners refitted to a thin web, the compressive stresses must be confined largely to the stiffeners and the chords, since the web is effective only for tensile stresses. There is no sharp division between the case where the web will support the compressive component and where it will not. The allowable compressive stress in the web is gradually reduced as the d/s ratio is increased. (d = minimum spacing of reinforcing members.) The addition of vertical stiffeners to a thin web not only provides additional compression-resisting cross-sectional area, but also undoubtedly increases the ability of the web to support diagonal compression.

Credit is due Dr. Wagner for inviting attention to the fact that the shear strength of thin webs is a function, not only of the spacing of stiffeners, but of their compressive strength as well.

There is an important difference between the plate girder and the side wall of a stiffened monocoque shell of curved cross-section. The shear strength of the former is reduced by the bending inward of the chord members, while the continuity of the latter avoids this marginal effect. The flat-sided fuselage with square corners is comparable to the plate girder, but is of little interest because of its poor aerodynamic qualities, awkward appearance, and tendency toward "drum-head" vibration.

It is very difficult to obtain experimental results applicable to fuselage or float construction without testing a complete monocoque shell. Tests demand the provision of a plate girder simulating continuity and avoiding the usual marginal effect.

The Southwell-Skan formula given by the author is applicable to beam webs without vertical stiffeners where the web must support diagonal compressive loads. The fact needs emphasizing that it is not applicable to panels bounded on all sides by stiff margins or to continuous skin stiffened by frames and stringers.

Testing and development work in metal monocoque construction is well under way in this country, but the application to design has so far been wholly empirical. A few of the complexities of the problem are:

(1) Strength of curved skin in compression due to bending
(2) Strength of curved skin in diagonal tension due to shear
(3) Design of longitudinal stiffeners to withstand inward component of tension in skin
(4) Strength of main bulkheads
(5) Loads on and strength of curved frames.

The writer would like to put a few questions regarding details and uses of the formula presented by the author that do not seem clear.

In the formula giving the shear force S required to buckle walls, I_z and d_v are not explained. Is the assumption correct that I_z is the moment of inertia of the section of the longitudinal reinforcements, d_z the distance between them, and I_v the moment of inertia of the transverse reinforcements, and d_v the distance between them? If this is true, how would the value of I_z/d_z be calculated in the tail portion of a monocoque fuselage where there are no longitudinal reinforcements and the skin is corrugated? Could each corrugation be treated as a reinforcement and d_z be the pitch of the corrugations? It is interesting to note that, should this formula be applied to smooth shell structures, the maximum shear load is governed entirely by the properties and location of the reinforcements and is independent of the skin thickness. It would have been interesting had the author explained how this formula could be applied to the tail section of a monocoque fuselage when a smooth skin and a large number of equal size longitudinal reinforcements are used.

The article does not state whether the constants in the formulas are calculated for the metric or English system of measurement.

The subject of allowable shear stress in a thin reinforced sheet is very interesting at the present time. It is regretted that Dr. Wagner did not present the actual test data or explain the deviation of the Southwell-Skan formula. Using the English system of units in this formula, it is found that for values of $d/s = 50$, where d = diameter of tube and s = wall thickness, it gives values comparable with those we use. For values in tubes with d/s less than 50, it gives values proportionally higher than those now recommended in this country. For values in thin web beams and monocoque shells with cross-bracing, where $d/s = 100$ to 350, it gives values far below those based upon our test data. It is believed that in reinforced smooth skin structures d/s is seldom less than 100.

A study of the formulas used for computing the properties of the reinforcements to prevent the buckling of the web is very interesting. It is understood that I_z/d_z = the moment of inertia of a transverse section of a longitudinally corrugated web of width d_z divided by the width or the height, as in a wing spar.

Apparently the deeper the corrugations, the greater will be the shear load capacity of the web. It is regretted that the author did not give this optimum depth for the best strength-weight ratio—the depth governed by the sheet thickness and d_z. It would be interesting to know if the same formula may be used for the so-called "wandering web" used sometimes in wing spar webs where the corrugations are large in proportion to the sheet thickness and lie normal to the plane of horizontal shear. If it is applicable, then d_v must be the pitch of the corrugations and I_v the moment of inertia of a cross-section of a complete corrugation, about its neutral axis.

The author's formula giving the allowable compressive load presents a very interesting question—whether the sectional properties and spacing of the longitudinal and transverse reinforcements have an equal effect in calculating the allowable compressive load. It appears that the material in the longitudinal reinforcements, if any, has a dual purpose—that of preventing the skin from buckling and that of carrying some of the compressive load directly while the transverse reinforcements simply prevent the skin and longitudinals from buckling and do not carry any of the compressive load which is acting normal to the transverse members. The same formula contains another very interesting factor—that of the effect of the torsional strength of a transverse reinforcement. Is it correct to calculate T_v as the torsional moment that the section of the transverse member will carry at the ultimate or yield point and G as the torsional modulus of elasticity? It is not clear in the paper how this strength affects the allowable compressive load normal to the reinforcement. It is noted that the author did not compare the diaphragm or bulkhead type of transverse reinforcement with the ring type. In the design of a bulkhead

[3] Buffalo, N. Y. Mem. A.S.M.E.

is I_s calculated for a section from the center to the outer fiber at a point in question?

In applying this formula to a monocoque fuselage covered with corrugated sheet and not employing any longitudinal reinforcements, is it correct to calculate I_x as the moment of inertia of a cross-section through one corrugation of the skin and d_x as pitch of corrugation? Again, it is interesting to note that in applying this formula to a smooth-skin monocoque structure the thickness of the skin does not enter into the calculations for the maximum compressive load when I_s and I_x apply to the transverse and longitudinal reinforcements, respectively.

The writer agrees with Dr. Wagner that the radius of curvature has an important influence on the allowable compressive stress in the region of small radii.

It has been our experience that the effect of the radius of curvature on the allowable compressive stress in a smooth metal monocoque shell cannot be plotted as a linear function. From test data available it appears that the effect of an increase in the radius of curvature is practically negligible when the radius of curvature is greater than 1300 times the skin thickness, while the reduction of the radius of curvature from $300t$ to $200t$ increases the allowable compressive stress about 9 per cent, where t = sheet thickness, and for smaller radii the rate of increase is greater.

It has been found that failure in compression in metal monocoque structures results in an inward buckling of the skin and reinforcements. This is apparently due to the eccentric loading of the reinforcements.

The author has made detailed refererce to square-section bodies where four main longerons are used to carry the principal stresses. It would have been interesting had he treated a round or elliptical section with a number of longitudinal reinforcements all having the same section. In such structures there are regions of the skin that lie between the regions supporting maximum shear and those supporting maximum compression that are subjected to both shearing and compressive forces. It is believed that in these regions a careful study should be made to select the proper amount of skin or shell reinforcement.

It is interesting to note that the author gives an angle of 120 deg. between the longitudinal and transverse reinforcements as the best angle to take shear in one direction. Since in the opposite direction we require in the case of wing spars about one-half the main strength and this angle reduces the strength to one-third, this rule might not be applicable to wing-spar design.

The author's reference to "tension field type" beams apparently applies to all reinforced thin sheets where d/s is large and the sheet wrinkles under a light shear load which is actually carried by tension lines. The author states that beams built as the "tension field type" are "lighter than a framework." This might open a discussion for those who are fostering the truss type of construction for very deep units. It is assumed that the author means "truss" when he uses the word "framework."

It would have been interesting had the author given a general idea regarding the amount of reduction in performance when corrugated covering is used on fuselages. The increase in drag may be partly due to the angle existing between the line of air flow and the axis of the corrugation.

Closure by J. Otto Scherer[9]

Colonel Clark's discussion of the paper brings up a number of interesting points. The writer is sorry that he cannot answer the questions as to the derivation of the formulas, etc., but believes such should be answered by Dr. Wagner himself.

Dr. Wagner, it appears, has rather slighted two important forms of all-metal construction, forms which are steadily increasing in popularity in this country, while making a case for the "tension field" beam, with which he has had extensive experience and which has given very good results.

With regard to the increased drag caused by the wrinkles appearing in the "tension fields" of Dr. Wagner's beams, it might be pointed out that about two years ago the D.V.L. published the results of rather extensive full-scale tests conducted to determine the effect of skin smoothness on drag.

These tests showed that an absolutely smooth surface gave a considerably lower drag. However, as soon as this surface was marred, if even only to the extent necessitated by countersunk rivet heads, the drag increased to a "normal" figure, which varied but little for all the usual types of covering.

By "usual types of covering" are meant a fabric covering, as normally applied and finished, a plywood covering with its ever-present small unevennesses, a smooth metal skin with either countersunk or round-head rivets, and the corrugated Dural skin. The "tension field" construction would probably present surfaces about the same as a normal plywood wing covering.

With regard to the corrugated-skin construction, it may be of interest to mention that the Junkers Works use two depths of corrugation, depending on whether the structural stresses carried by the skin are relatively high or low. On such parts as the fuselage and the portions of the wing near the fuselage, the corrugations have a depth equal to $^1/_3$ the pitch, while on the outer parts of the wing, etc., the depth of the corrugations is only $^1/_6$ pitch.

The metal monocoque fuselage employing a corrugated skin is usually strong enough to carry all the flying loads, even though no longitudinals are incorporated. Experience, however, has shown that it is desirable to run a rather stiff longitudinal along the bottom center of fuselages of oval section or along the lower corners of those of rectangular section. Such longitudinals are of considerable value in protecting the fuselage against damage incident to accidental mishandling on the ground.

In conclusion may one take the liberty to remark that, though designers in this country are, on the whole, inclined toward types of all-metal construction in which the skin is designed to carry the compression as well as the tension components of the shear stresses, Dr. Wagner's type of construction is well worth consideration and well applicable to good advantage on a number of types built in the United States today.

The "tension field" construction could nicely be applied to almost all jobs which make use of a rectangular welded steel-tube fuselage with fabric covering. The fabric, with its dope, weighs just about as much as a thin Dural sheet, but does not contribute anything to the strength of the structure. The Dural sheet, on the other hand, can safely be included in the structure, making possible a reduction in the weight of the trussing without any sacrifice of strength.

[9] Chief Engineer, Junkers Corporation of America, New York, N. Y. Jun. A.S.M.E.

APM-53-1

The Stresses in Rotating Disks

By THOS. H. FROST,[1] CAMBRIDGE, MASS., and K. F. WHITCOMB,[2] WORCESTER, MASS.

This paper comprises the results of a photoelastic investigation, sponsored by The Norton Company, of the stresses in rotating disks. The classical theory of Chree with a few additions is presented, and the results of the experimental determinations are compared with the theory as well as actual breakages of vitrified wheels.

THE object of the investigations described in the present article has been to determine, by photoelastic experimentation, the stress distribution in rotating disks of constant external but varying internal diameter, and to compare the results obtained with the existing classical theory and actual practice.

The practical consideration which suggested the work was the modern demand for higher grinding-wheel speed as an effective means of increasing production. Since engineers are not agreed on relative speeds, with equal safety, as wheel dimensions are varied, it was felt that a more complete understanding of stresses and their causes would be of great value in the design of wheels as well as in the specification of higher, but safe, operating speeds.

Theory of Stresses in Rotating Disks

In order to obtain some idea as to what streases exist in a rotating disk, the solution on rotating isotropic disks by Chree[3] is given in the Appendix. This solution, based on the theory of elasticity, is most generally accepted as being the correct one. A few additions to Chree's paper, such as diagrams showing the stress forces and the strains, have been made.

The theoretical stress equations, as derived by Chree, giving the relationships between stress, speed, mass, and dimensions are as follows:

$$\widehat{rr} = \frac{\omega^2\rho}{8}(3+\eta)\left\{a^2 + a'^2 - r^2 - \frac{a^2a'^2}{r^2}\right\} \quad \ldots\ldots [22]$$

$$\widehat{\varphi\varphi} = \widehat{rr} + \frac{\omega^2\rho}{4}\left\{(1-\eta)r^2 + (3+\eta)\frac{a^2a'^2}{r^2}\right\} \quad \ldots\ldots [23]$$

Where:

$\widehat{rr}$ = radial stress at any point distant r feet from the center of the disk, in pounds per square foot

$\widehat{\varphi\varphi}$ = tangential stress at any point distant r feet from the center of the disk, in pounds per square foot

ω = angular velocity of disk in radians per second

ρ = mass per unit volume of disk

η = Poisson's ratio, assumed equal to 0.27 here

a = radius of disk in feet

a' = radius of central hole of disk in feet.

These equations were used in obtaining the theoretical values of stress presented later in this paper.

The Photoelastic Method

Photoelasticity has been explained in detail in various journals, hence it is felt that a very brief résumé will suffice to make clear the method of experimentation used in the present investigation.

It is well known from the theory of elasticity that at any point in a stressed body there exist three orthogonal planes across which the traction is purely normal. These normal tractions are known as the principal stresses, and when they can be determined both in magnitude and direction, the elastic state is completely defined. When one of these stresses is zero, we have a two-dimensional stress problem, and it is in such cases that the photoelastic method is used to determine the direction and magnitude of the principal stresses. A model of the structure which it is wished to examine is made of some transparent material, usually some form of celluloid, and placed in a beam of polarized light. When the model is stressed it becomes temporarily double-refracting, and upon analysis of the light the image of the model is seen to be highly colored. These colors are a measure of the difference of the two principal stresses at any point, and this difference can be measured with a high degree of accuracy by introducing a compensating device in such a way that the color effect can be destroyed. The directions of the principal stresses can be also determined by a simple manipulation of the plane of polarization of the light.

The Apparatus Used

The set-up of the apparatus as used for the determination of the stresses in rotating disks was necessarily different from that conventionally used for the examination of statically loaded specimens. A photograph of the arrangement as used in these experiments is shown in Fig. 1.

In order to secure an intermittent source of light which would give a stroboscopic effect, two electrodes connected to a high-tension transformer set were employed. These electrodes are contained in the box 2 and are connected in series with a bakelite disk 1 stripped with copper. Two copper-wire brushes properly placed made contact at a certain time during each revolution of the shaft with the ends of the copper strip on the Bakelite disk, thus completing the circuit to the electrodes and forcing a spark inside the box at the same position of the disk during each revolution of the shaft. By controlling the input to the high-tension transformer the synchronization of spark and disk could be easily controlled.

The light from the box 2 was polarized by the Nicol prism 3 and passed through the rotating celluloid disk 4 in a parallel beam. The compensating device for measuring the magnitudes of the principal stresses is shown at 5, and the analyzing Nicol prism at 6. A potential of 70,000 volts was found to be sufficient to allow easy optical observations.

The design of the disk used, as well as the device employed for fitting it to the shaft, is shown in Fig. 2. The original celluloid disk had an external diameter of 20 in. and an internal diameter of 6 in., two drive-pin holes $^1/_2$ in. $\times$ $^5/_8$ in. being placed 180 deg. apart. As the experiment progressed the internal

---

[1] Assistant Professor, Department of Physics, Massachusetts Institute of Technology. Mr. Frost was graduated from The Citadel in 1914. After the War he attended M.I.T., receiving the degree of Master of Science in 1922. Since graduation, he has been on the staff of the Institute.

[2] Research Laboratory, The Norton Company. Mr. Whitcomb was graduated from the University of Wisconsin in 1918. After the war he spent two years with the Western Electric Co. and later was for a year and a half engaged in graduate work at the University of Chicago and Northwestern University. Since 1927 he has been doing research work on the mechanics of ceramic materials.

[3] Proc. Camb. Phil. Soc., 1891, pp. 201–215.

Contributed by the Applied Mechanics Division and presented at the Semi-Annual Meeting, Detroit, Mich., June 9 to 12, 1930, of The American Society of Mechanical Engineers.

diameter of the disk was changed successively to 9 in. and 11.29 in., this last diameter being the largest that could be used with the holding device as originally designed.

The holding device itself was of simple construction, consisting of a duralumin disk $^3/_8$ in. thick with two shoulders on one side. The larger of these shoulders received the celluloid disk which was to be examined, and this shoulder was made $^1/_{64}$ in. thicker than the sheet celluloid used for the models to insure against any friction across the surface of the disk. The smaller

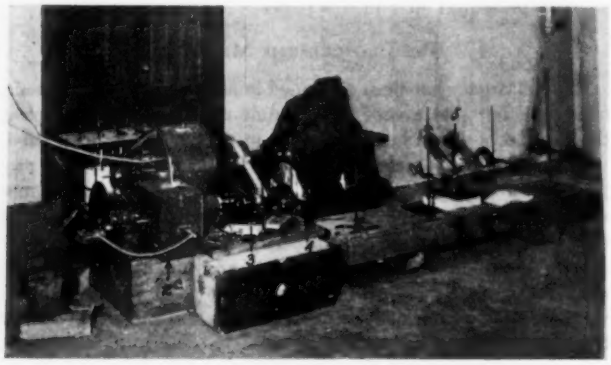


FIG. 1 PHOTOELASTIC SET-UP

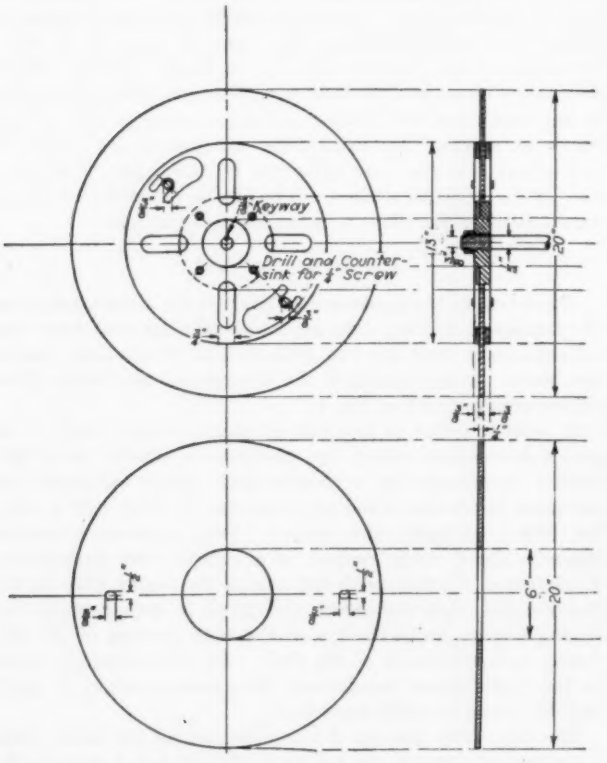


FIG. 2 DESIGN OF DISK AND OF DEVICE EMPLOYED FOR FITTING IT TO THE SHAFT

shoulder received a duralumin disk which served as a guard on the outside of the rotating model. The duralumin disks and the model were secured together by the drive pins mentioned above. These pins had a shoulder $^1/_2$ in. in diameter between the two metal disks to prevent the celluloid from being squeezed by the holding device in assembling.

In order to observe the stresses in that part of the model covered by the holding device, four slots were cut through the duralumin disks at 90 deg. apart and at 45 deg. from the drive pins.

Fig. 3 shows the model and holding device demounted. The spacer disk in the foreground is one of those used as a bushing between the disks having inside diameters greater than 6 in. and the shoulder of the holding device which received the 6-in. disk directly. It can be seen from the photographs that the

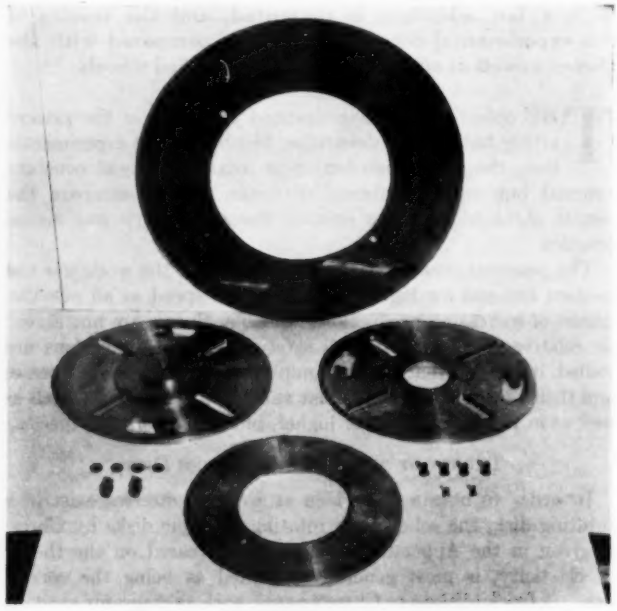


FIG. 3 ROTATING DEVICE DEMOUNTED

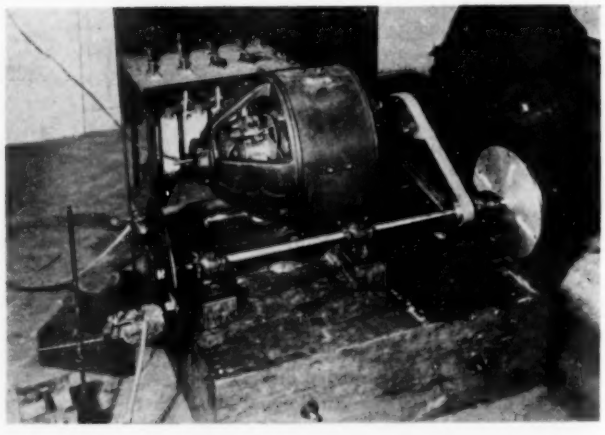


FIG. 4 METHOD OF ROTATING DISK AND SYNCHRONIZING DEVICE

drive-pin holes in the duralumin disks have been placed with open space on all possible sides in order to permit direct photoelastic observation of the stresses around the pins when the disk was in motion.

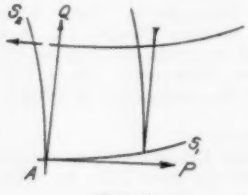


FIG. 5

Fig. 4 illustrates the apparatus used for rotating the disks. The model to be examined and the synchronizing apparatus are shown on opposite ends of a 1-in. shaft supported by three ball bearings.

A 10-hp. d.c. motor furnished the motive power, and by introducing resistances in the armature and field circuits, any desired speed could be obtained.

Experimentation

The apparatus being arranged as shown in Fig. 1, the model was rotated to the desired speed, all measurements of r.p.m. being taken on a Hassler tachometer, which was frequently checked during the runs. The operator placed himself in such a position that he could observe the model directly through the analyzing Nicol, and at the same time manipulate the compensator. The high tension being applied to the electrodes and synchronized by an assistant, the image as seen through the apparatus appeared still and continuous, so that the observations of stress magnitude could be made at leisure. By moving the photoelastic equipment bodily outward, any desired radius could be completely explored.

The measurement of stress magnitude at any point on a radius gives the difference of the principal stresses unless it be at the inside or outside periphery, both of which are of course free. The separation of these values of difference of principal stress into the two existing stresses is usually accomplished by means of Filon's equations.[4] In the case of the rotating disk, however, it became necessary to consider the body forces which are not taken care of in the above-mentioned equations, and also to use the equation of equilibrium given in polar coordinates. An adaptation of the Filon method to care for such cases is given below.

Adaptation of Filon's Method

In order to obtain the principal stresses, Filon developed a method based upon equations given by A. Mesnager for the space rates of change of the principal stresses, taken along the lines of principal stress. In this problem, due to the symmetry of the figure the directions of the principal stresses are along radii and along concentric circles. Filon's method proceeds as follows:

If we denote by S_1, S_2 arcs taken along the lines of principal stress corresponding to stresses P, Q, respectively (Fig. 5), dS_2 being obtained from dS_1, by a counterclockwise rotation of 90 deg., and if ρ_1, ρ_2 are the radii of curvature of the two lines of principal stress, being measured positive when the tangents to the curves rotate counterclockwise as the arcs S_1, increase, we have the equations

$$\frac{\partial P}{\partial S_1} + \frac{P - Q}{\rho_2} = 0$$

$$\frac{\partial Q}{\partial S_2} + \frac{P - Q}{\rho_1} = 0$$

These expressions are nothing more than the equations of equilibrium at one corner of a curved elementary rectangle. If we integrate them and proceed along the corresponding line of principal stress, we obtain

$$P = P_0 - \int_0^A (P - Q)dS_1/\rho_2$$

$$Q = Q_0 - \int_0^B (P - Q)dS_2/\rho_1$$

We see that the integrands of these equations contain the differences of the two principal stresses which we obtain by photoelasticity.

In our particular problem the equation of equilibrium of an element taken from the disk takes on a somewhat different form. Let us consider this equation of equilibrium [2] in the Appendix, namely,

[4] *Engineering*, October 19, 1923.

$$\frac{d\widehat{rr}}{dr} + \frac{d\widehat{rz}}{dz} + \frac{\widehat{rr} - \widehat{\varphi\varphi}}{r} + \omega^2\rho r = 0 \ldots\ldots\ldots [2]$$

Chree has found as shown in the Appendix that, with thin disks, the shear stress $\widehat{rz} = 0$. Therefore Equation [2] takes the following form:

$$\frac{d\widehat{rr}}{dr} + \frac{\widehat{rr} - \widehat{\varphi\varphi}}{r} + \omega^2\rho r = 0 \ldots\ldots\ldots\ldots [2a]$$

Equation [3], the second equation of equilibrium given in the Appendix, becomes zero since Chree found that $\widehat{rz}$ and $\widehat{zz}$ were equal to zero.

Integrating Equation [2a] gives

$$\widehat{rr} = \int \frac{(\widehat{\varphi\varphi} - \widehat{rr}) - \omega^2\rho r^2}{r}\, dr + K$$

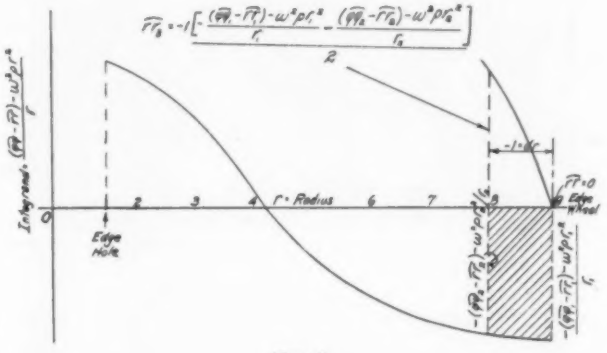


Fig. 6

where K is a constant of integration of such a value that at the free surface a value of $\widehat{rr} = 0$ is obtained.

It will be seen from this equation that it contains a term $(\widehat{\varphi\varphi} - \widehat{rr})$ in the integrand which is the difference between the tangential and the radial stresses at any point that is being considered. But this corresponds to what we obtain by photoelasticity.

Thus if we substitute these values of $(\widehat{\varphi\varphi} - \widehat{rr})$ in this equation for given values of r and also obtain $\omega^2\rho r^2$ for the same values of r and finally take the area under the $\frac{(\widehat{\varphi\varphi} - \widehat{rr}) - \omega^2\rho r^2}{r}$ — r curve by multiplying

$$\frac{(\widehat{\varphi\varphi}_1 - \widehat{rr}_1) - \omega^2\rho r_1^2}{r_1} + \frac{(\widehat{\varphi\varphi}_2 - \widehat{rr}_2) - \omega^2\rho r_2^2}{r_2}/2$$

by dr, which will be taken as unity, we shall obtain the radial stress at the point considered—see Fig. 6.

By adding the values obtained above by integration to the photoelastic value obtained for the particular point considered we obtain the tangential stress for the point considered. This method was followed in determining the tangential- and radial-stress-distribution curves from the photoelastic values obtained in tests of disks with different-sized holes, as is shown under Results.

In a general problem we should not have a straight line along which to integrate as we have in this case with the radial stresses. In such cases this adaptation of Filon's method would be inapplicable.

Results

In Fig. 7 are shown the results of two runs at 3400 and 3700

r.p.m. for a disk having an internal diameter of 6 in. The values plotted are those obtained by photoelastic analysis, viz., the difference between the tangential and radial stress at any point. The separation of these values into the separate values of the principal stresses has been accomplished by the adaptation of Filon's method explained above.

In Fig. 8 the experimental results given in the preceding figure are shown after the separate values of the radial and tangential stresses have been determined by the aid of the adaptation of Filon's method. Also the theoretical values for the same conditions have been calculated and are shown plotted in the same figure.

It is interesting to note that the difference between theory and experiment at the slower speed is approximately 20 per cent at the inside edge of the disk. This is the maximum variation found in the series of experiments when considering the edges of the holes. At the higher speed the theoretical and experimental curves lie much closer together. The agreement between theory and experiment for the radial stresses is very good, but the magnitude of these stresses is small as compared to the tangential ones.

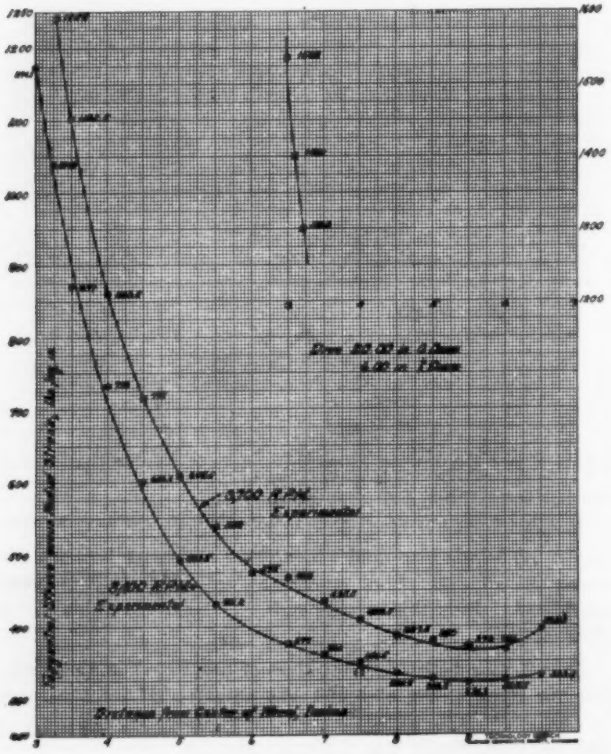


FIG. 7

In Fig. 9 the experimental results of runs at approximately the same speeds on disks having inside diameters of 9 and 11.29 in. are given. The separate values of the radial and tangential stresses are given in Figs. 10 and 11, respectively. In the case of the disk of 9 in. inside diameter the agreement between theory and experiment is very close for the tangential stresses; the experimental values, however, are somewhat lower than the theoretical ones. In the case of the disk of larger inside diameter it is interesting to note that for the first time the experimental values at the inside edge of the disk are higher than the calculated ones.

Brief runs were made on a disk having very small internal diameters. In Fig. 12 the difference between the radial and tangential stresses as experimentally determined is shown in comparison with the theory. The agreement between theory and experiment is so close that only the theoretical separate values are shown graphically. The curves of difference of stresses are radically different in shape from those taken on disks of large internal diameter. The downward slope of these curves from the outer periphery to the center of the radius, as well as the very abrupt increase in stress as the inner edge of the disk is approached, is very clearly reflected in the separate

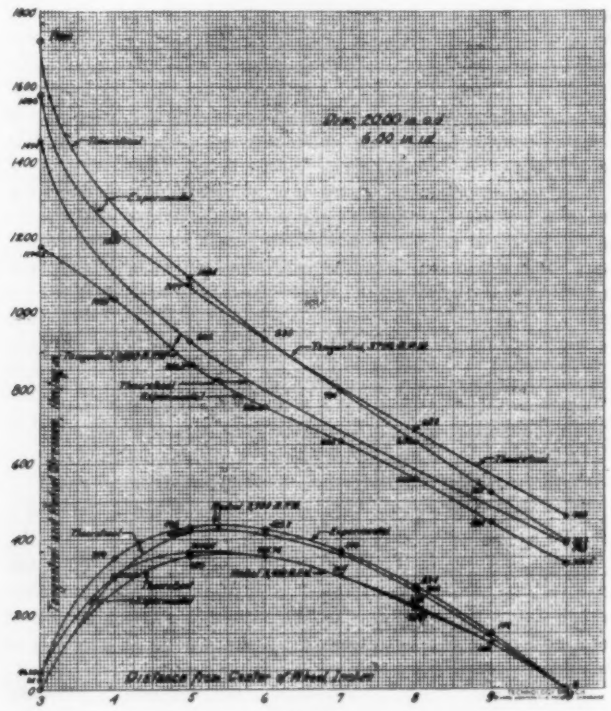


FIG. 8

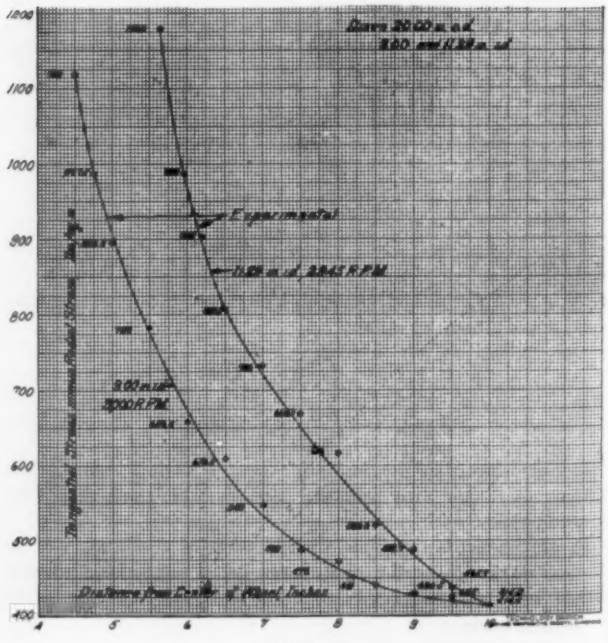


FIG. 9

theoretical curves for the tangential and radial stresses. At the middle of the radius the difference in stress is very small, but the actual values of the radial stresses are larger than those found in the disks having larger internal diameters. It is also of interest that at certain points the absolute values of the radial stresses are very close to those of the tangential stresses.

In Fig. 13 the results of the tests on a disk having an internal diameter of 0.25 in. are shown. These curves are very similar in shape to those of the preceding figure. However, the maximum radial stresses in this case are higher than those in Fig. 12, and approach more closely the values of the tangential stresses. The maximum stress in the disk as determined experimentally is considerably lower than in the case of the disk of 1.09 in. internal diameter.

In order to determine the effect of varying speeds on the maximum tangential stresses, runs were made on the disks having internal diameters of 6, 9, and 11.29 in., the results being shown graphically in Fig. 14. The curves show very clearly the effect of hole size at given speeds. It is also interesting to see that the stress at the edge of the hole varies approximately as the square of the rotational speed.

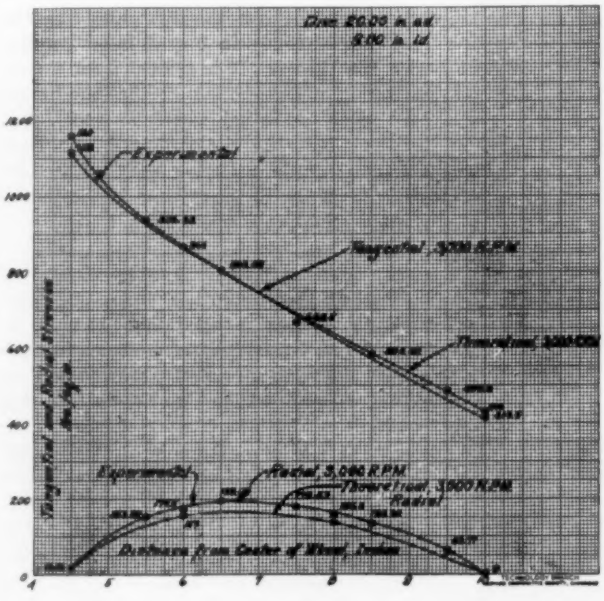


Fig. 10

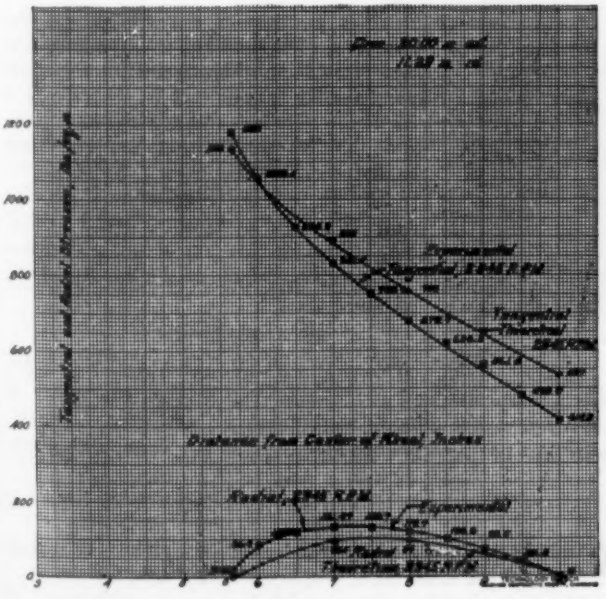


Fig. 11

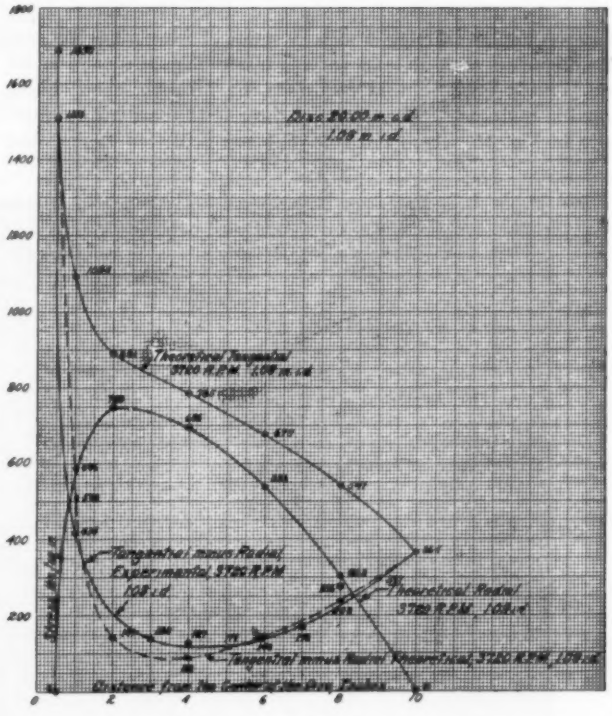


Fig. 12

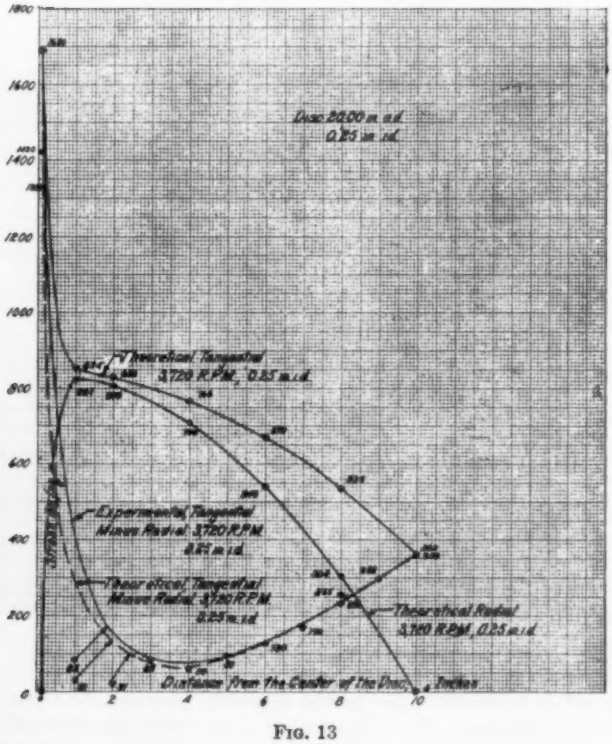


Fig. 13

Fig. 11

In order to compare the results of these experiments with those on actual grinding wheels, a series of wheels having outside diameters of 20 in. and varying inside diameters were made and rotated until rupture occurred, the breaking speed being noted in each case. The results of these tests are shown graphically in Fig. 15, where the hole diameter is plotted against the

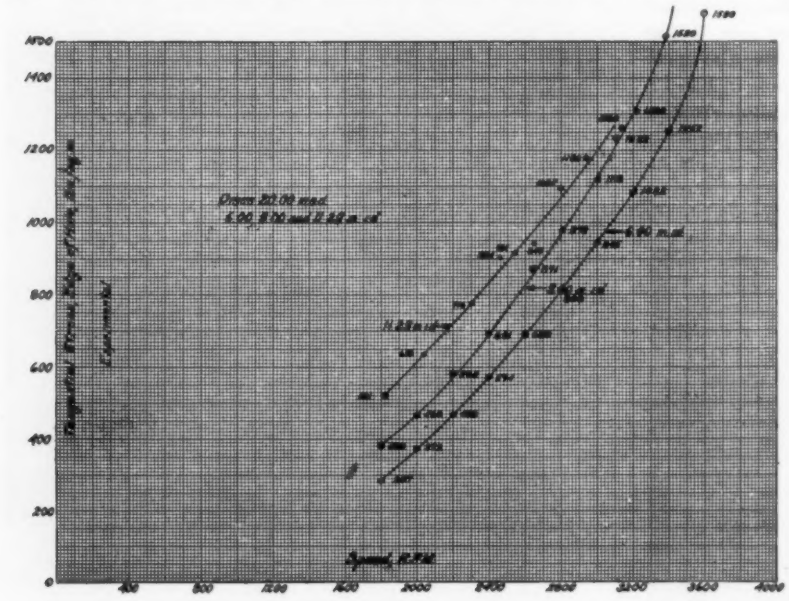


FIG. 14

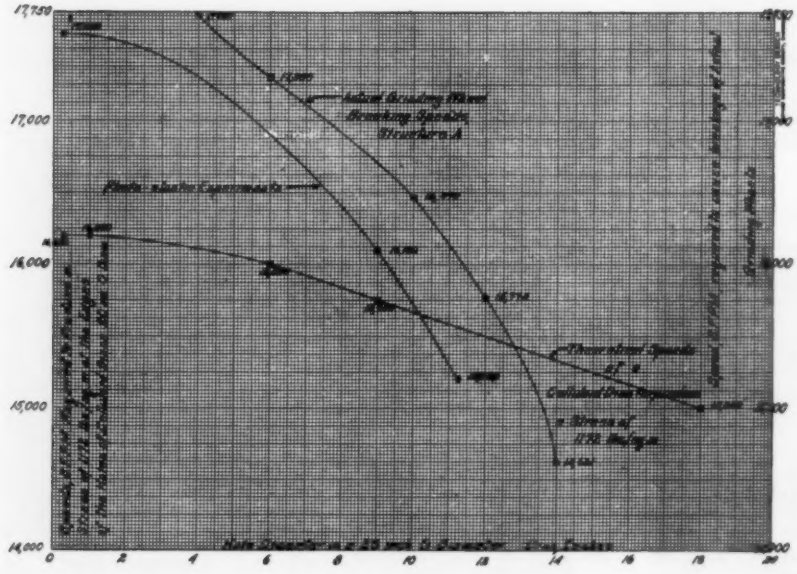


FIG. 15

speed in surface feet per minute necessary to cause the actual wheel to rupture. Since the structure of the wheels was identical, it was assumed that the breaking stress would be very nearly the same in all cases. The celluloid disks used for the photoelastic experiments were then rotated successively until the maximum tangential stress at the edge of the hole rose to an arbitrarily chosen value of 1172 lb. per sq. in. The curve of speed necessary to produce this magnitude of stress plotted against hole diameter is shown also in Fig. 15. It can be seen that the two curves are very nearly parallel, which indicates that the photoelastic experiments approximate very closely the elastic behavior of the actual wheels used for this comparison. The curve which results from theoretical calculations is also shown in the same figure, and it will be noticed that it crosses both of the experimental curves. The discrepancy, however, is not so great as would appear at first glance since the scale of ordinates is unusually large.

It was felt of course that the structure of the actual wheels would greatly influence the shape of the curve obtained in Fig. 15. The first structure employed was designated as "Structure A." A second structure of wheel, designated as "Structure B," was then chosen, and the experiments outlined above were repeated with this new material, the results being shown in Fig. 16 in comparison with the theory. It is evident from a consideration of Figs. 15 and 16 that the breaking speeds for actual wheels are controlled between limits by the structure employed, and that for the materials used in these investigations one of these limits is the theory and the other is the photoelastic analysis.

Summary of Results

Photoelastic experiments on rotating disks having a common external diameter of 20 in. and varying internal diameters have, in comparison with the theory of Chree as well as actual wheel breakages, given the following results:

a The maximum stress in a rotating disk is a tangential tension at the edge of the central hole.

b This maximum stress is proportional to the square of the rotational speed, within the limits of experimental error.

c Increasing the diameter of the central hole results in an increase in the maximum stress.

d The discrepancy between theory and experiment when considering the edges of the holes of rotating disks ranges from zero to approximately 20 per cent in the case of a disk with a 6-in. hole rotating at 3400 r.p.m., where the photoelastic results are lower than the theoretical results.

e By controlling the structure of grinding wheels the curve of breaking speed against hole size can be varied to accord with either the photoelastic results or the theoretical calculations.

Conclusion

Taking all data into consideration it is felt that a reasonable and usable agreement between the theory of elasticity, actual grinding wheel breakage, and photoelastic experiments has been pointed out.

Appendix

CHREE'S solution, with several additions, giving the derivation of the stress equations in a rotating disk, follows.

The term "disk" is here restricted to mean a thin plate, of

which a section parallel to the faces is bounded by two concentric circles. The disk is assumed to be of uniform density ρ and of an isotropic material, for which m is the bulk modulus and n the modulus of rigidity—see Thomson and Tait's "Natural Philosophy," Part II.

Take the axis of the disk for the axis of Z with the origin 0 in the central plane or plane bisecting the thickness of the disk, and let the disk rotate with a uniform angular velocity ω.

The internal-stress equations of equilibrium for an element of this disk are derived as follows:

Fig. 17 shows the distribution of the stress forces on an element enclosed between two cylindrical surfaces of radius r and $r + dr$, two planes through the axis at angles φ and $\varphi + d\varphi$, and two transverse planes distance dz from each other in general. This element has been removed from the interior of the rotating disk.

We may suppose the disk at rest, acted on by a centrifugal force $\omega^2\rho r$ per unit of volume $= R$. The other body forces in the φ and z directions are equal to zero.

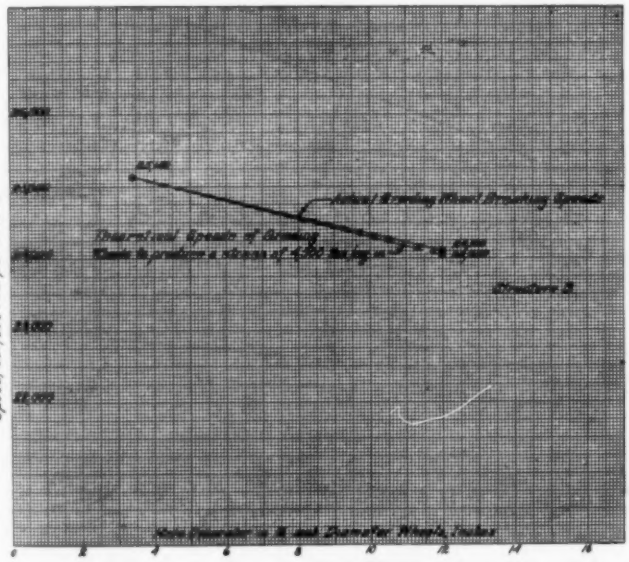


FIG. 16

The moment equations about the axes through the middle point of the element give the usual result

$$\widehat{r\varphi} = \widehat{\varphi r}, \qquad \widehat{rz} = \widehat{zr}, \qquad \text{and} \qquad \widehat{z\varphi} = \widehat{\varphi z}$$

Resolving along r, we find for $\widehat{rr}$:

$$\left(\widehat{rr} + \frac{\partial \widehat{rr}}{\partial r}\,dr\right)\cdot(r + dr)\,d\varphi dz + \omega^2\rho r \times rd\varphi drdz - \widehat{rr}rd\varphi dz$$

Disregarding infinitesimals of the second degree, we obtain:

$$\widehat{rr}drd\varphi dz + \frac{\partial \widehat{rr}}{\partial r}\,dr \times rd\varphi dz + \omega^2\rho r^2 d\varphi drdz$$

For the stresses $\widehat{\varphi\varphi}$, which act inward at an angle of $\frac{1}{2}d\varphi$ to the middle radius, we have:

$$-\left(\widehat{\varphi\varphi} + \widehat{\varphi\varphi} + \frac{\partial \widehat{\varphi\varphi}}{\partial \varphi}\,d\varphi\right) drdz \times \tfrac{1}{2}d\varphi$$

since $\sin\frac{d\varphi}{2} = \frac{d\varphi}{2}$ for a small angle.

Again disregarding infinitesimals of the second degree, we have

$$-\widehat{\varphi\varphi}drdzd\varphi$$

The shear stresses give

$$\frac{\partial \widehat{\varphi r}}{\partial \varphi}\,d\varphi drdz + \frac{\partial \widehat{zr}}{\partial z}\,dzrd\varphi dr$$

The body force $\omega^2\rho r \times rd\varphi drdz$ has been given above.

Adding the above forces and dividing by $rdrd\varphi dz$, we obtain the first of the three equations of equilibrium:

$$\frac{\widehat{rr}}{r} + \frac{\partial \widehat{rr}}{\partial r} + \omega^2\rho r - \frac{\widehat{\varphi\varphi}}{r} + \frac{\partial \widehat{\varphi r}}{\partial \varphi}\Big/r + \frac{\partial \widehat{zr}}{\partial z} = 0$$

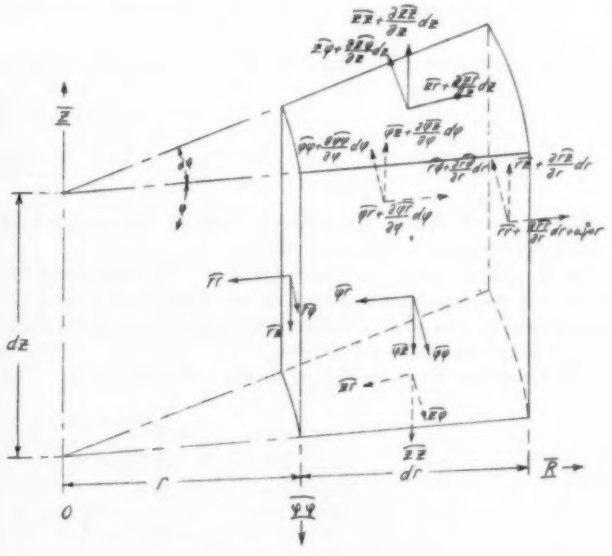


FIG. 17

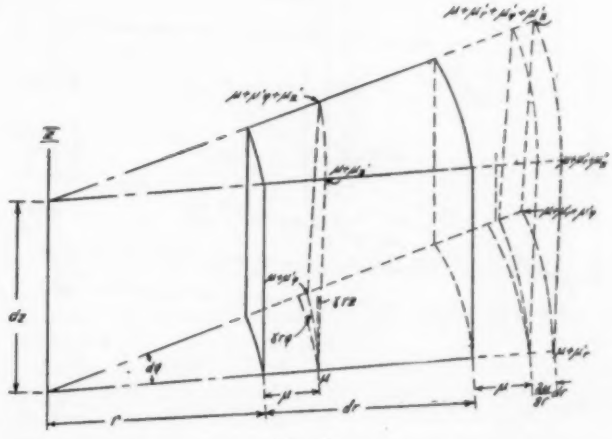


FIG. 18

which reduces to

$$\frac{d\widehat{rr}}{dr} + \frac{d\widehat{rz}}{dz} + \frac{d\widehat{\varphi r}}{d\varphi}\Big/r + \frac{\widehat{rr} - \widehat{\varphi\varphi}}{r} + \omega^2\rho r = 0 \ldots . [1]$$

Due to the symmetry of the disk the shear $\widehat{\varphi r} = 0$; therefore we have

$$\frac{d\widehat{rr}}{dr} + \frac{d\widehat{rz}}{dz} + \frac{\widehat{rr} - \widehat{\varphi\varphi}}{r} + \omega^2\rho r = 0 \ldots\ldots\ldots [2]$$

for the first equation of equilibrium.

The other two stress equations may be obtained by resolving the forces along the tangent and along the Z-axis, respectively. They are as follows:

$$\frac{d\widehat{\varphi\varphi}}{rd\varphi} + \frac{d\widehat{\varphi z}}{dz} + \frac{d\widehat{r\varphi}}{dr} + 2\frac{\widehat{r\varphi}}{r} = 0 \ldots\ldots\ldots [3a]$$

$$\frac{d\widehat{zz}}{dz} + \frac{d\widehat{zr}}{dr} + \frac{\widehat{zr}}{r} + \frac{d\widehat{\varphi z}}{rd\varphi} = 0 \ldots\ldots\ldots [3b]$$

Due to the symmetry of the disk and the loading, the shear $\widehat{\varphi r} = 0$, as mentioned above, and for the same reason the shear $\widehat{\varphi z} = 0$. Therefore [3a] reduces to $\frac{d\widehat{\varphi\varphi}}{rd\varphi} = 0$, and [3$b$] reduces to

$$\frac{d\widehat{rz}}{dr} + \frac{\widehat{rz}}{r} + \frac{d\widehat{zz}}{dz} = 0 \ldots\ldots\ldots [3]$$

The equations giving the relations between the strains and the displacements are derived as follows:

Let u, v, and w be the components of the displacements, where w is the component parallel to the Z-axis and v the component normal to a plane through the Z-axis or tangential component and u the radial component.

Fig. 18 shows the deformation in the r-direction only. The

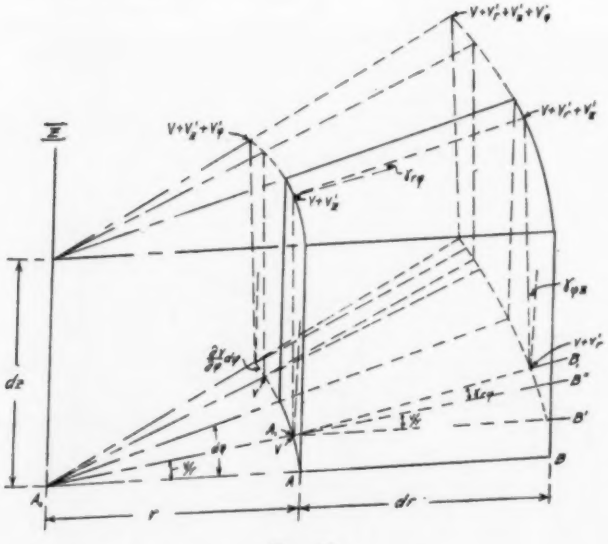


FIG. 19

increment $\frac{\partial u}{\partial r} dr$ is denoted by u_r', $\frac{\partial u}{\partial \varphi} d\varphi$ by u_φ', $\frac{\partial u}{\partial z} dz$ by u_z'.

All four edges have the same extension u in the r-direction. The outside edges of the element all have the extension $u + \frac{\partial u}{\partial r} dr$. Thus the strain $\epsilon_r = \frac{\partial u}{\partial r} dr/dr = \frac{\partial u}{\partial r}$, $rd\varphi$ becomes now $(r + u)d\varphi$, and therefore strain $\epsilon_\varphi = \frac{ud\varphi}{rd\varphi} = \frac{u}{r}$.

In this case we have $\epsilon_z = 0$.

The right angle in the $r\varphi$-plane is changed by the amount

$$\gamma_{r\varphi} = \frac{\partial u}{\partial \varphi} d\varphi/(r + u)d\varphi = \frac{\partial u}{\partial \varphi} d\varphi/rd\varphi = \frac{1}{r}\frac{\partial u}{\partial \varphi}$$

Omitting $ud\varphi$ as it is small compared with $rd\varphi$, we have in the $\overline{rz}$-plane

$$\gamma_{rz} = \frac{\partial u}{\partial z} dz/dz = \frac{\partial u}{\partial z}$$

In the $\overline{\varphi z}$-plane $\gamma_{z\varphi} = 0$ since no change can occur in u or r and remain in this plane.

Let us now consider the deformations in a tangential direction. Fig. 19 shows the deformation in the φ-direction only. The

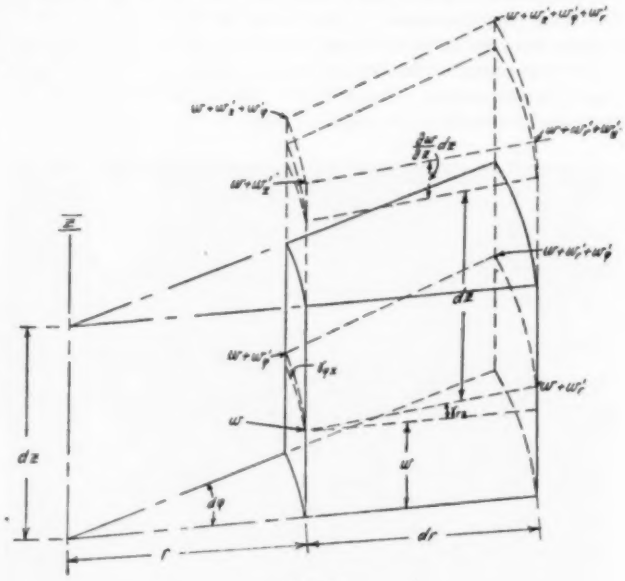


FIG. 20

increment $\frac{\partial v}{\partial r} dr$ is denoted by v_r', $\frac{\partial v}{\partial \varphi} d\varphi$ by v_φ', and $\frac{\partial v}{\partial z} dz$ by v_z'.

Upon undergoing deformation, A goes to A_1 and B to B_1, and A_1B' is parallel to AB; therefore $\tan B'A_1B_1 = \frac{\partial v}{\partial r} dr/dr = \frac{\partial v}{\partial r}$. But $\gamma_{r\varphi}$ is the angle between A_1B_1 and the radius A_0A_1 produced to B'', or in other words, the angle B_1A_1B''. Therefore $\gamma_{r\varphi} = \frac{\partial v}{\partial r} - \frac{v}{r}$, since the angle between $B''A_1$ and $B'A_1$ is equal to $\frac{v}{r}$.

There can be no ϵ_r strain since v is at right angles to r. Therefore $\epsilon_r = 0$.

The extension in the φ-direction will be $v + \frac{\partial v}{\partial \varphi} d\varphi$; thus strain $\epsilon_\varphi = \frac{\partial v}{\partial \varphi} d\varphi/rd\varphi = \frac{\partial v}{\partial \varphi}\frac{1}{r}$.

ϵ_z, the strain in the Z-direction, $= 0$ since $\overline{Z}$ is perpendicular to v-deformation.

In the $\overline{RZ}$-plane $\gamma_{rz} = 0$ since no change can occur in V or φ and remain in this plane.

In the $\overline{\varphi Z}$-plane $\gamma_{\varphi z} = \frac{\partial v}{\partial z} dz/dz = \frac{\partial v}{\partial z}$.

Let us now consider the deformations in the Z-direction only. Fig. 20 shows the deformations in the Z-direction only. The

increment $\frac{\partial w}{\partial r}\,dr$ is denoted by wr', $\frac{\partial w}{\partial \varphi}\,d\varphi$ by w_φ', and $\frac{\partial w}{\partial z}\,dz$ by w_z'.

There can be no ϵ_r or ϵ_φ strain since w is at right angles to φ and r. Therefore $\epsilon_r = 0$, and $\epsilon_\varphi = 0$; $\gamma_{r\varphi} = 0$ since it is perpendicular to w, and $\epsilon_z = \frac{\partial w}{\partial z}\,dz/dz = \frac{\partial w}{\partial z}$.

In the $\overline{ZR}$ plane $\gamma_{rz} = \frac{\partial w}{\partial r}\,dr/dr = \frac{\partial w}{\partial r}$

In the $\overline{\varphi Z}$ plane $\gamma_{\varphi z} = \frac{\partial w}{\partial \varphi}\,d\varphi/rd\varphi = \frac{\partial w}{\partial \varphi}\,\frac{1}{r}$

By addition we obtain ϵ_r total $= \frac{\partial u}{\partial r}$

$$\epsilon_\varphi \text{ total} = \frac{u}{r} + \frac{\partial v}{\partial \varphi}\,\frac{1}{r}, \text{ and}$$

$$\epsilon_z \text{ total} = \frac{\partial w}{\partial z}$$

also by addition

$$\gamma_{r\varphi} = \frac{\partial u}{\partial \varphi}\,\frac{1}{r} + \frac{\partial v}{\partial r} - \frac{v}{r}$$

$$\gamma_{\varphi z} = \frac{\partial v}{\partial z} + \frac{\partial w}{\partial \varphi}\,\frac{1}{r}, \text{ and}$$

$$\gamma_{rz} = \frac{\partial u}{\partial z} + \frac{\partial w}{\partial r}$$

From symmetry the shear strains $\gamma_{r\varphi}$ and $\gamma_{\varphi z}$ are zero; thus all we have left for shear strain is $\gamma_{rz} = \frac{\partial u}{\partial z} + \frac{\partial w}{\partial r}$, also

$$\frac{\partial v}{\partial \varphi} = 0;\ \therefore\ \epsilon_\varphi = \frac{u}{r} \text{ for tangential strain}$$

$$\epsilon_r = \frac{\partial u}{\partial r} \text{ and } \epsilon_z = \frac{\partial w}{\partial z}$$

$$\delta = \text{dilatation} = \epsilon_r + \epsilon_\varphi + \epsilon_z$$

$$= \frac{du}{dr} + \frac{u}{r} + \frac{dw}{dz} \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

The stress equations in terms of the strains and modulus of rigidity and bulk modulus are:

Normal Stresses

$$\widehat{zz} = (m - n)\,\delta + 2n\,\frac{dw}{dz} \text{ parallel to } OZ$$

$$\widehat{rr} = (m - n)\,\delta + 2n\,\frac{du}{dr} \text{ along } r$$

$$\widehat{\varphi\varphi} = (m - n)\,\delta + 2n\,\frac{u}{r}, \text{ perpendicular to } OZ \text{ and to } r$$

Shearing Stress: $\widehat{rz} = n\left(\frac{du}{dz} + \frac{dw}{dr}\right)$ in plane $\overline{ZR}$

As mentioned when deriving the stress equations of equilibrium, the shear stresses $\widehat{\varphi r}$ and $\widehat{\varphi z}$ for this problem are equal to zero.

Let 21 denote the thickness of the disk, a the outer radius, and a' that of its inner cylindrical surface or edge. Then assuming the disk is exposed to no surface forces, the solution should satisfy the following surface conditions:

Over the flat faces, $z = \pm$ $\widehat{lzz} = 0 \ldots\ldots\ldots\ldots [4]$

$\widehat{rz} = 0 \ldots\ldots\ldots\ldots [5]$

Over the edges, $r = a$ and $r = a'$ $\begin{cases} \widehat{rz} = 0 \ \ldots\ldots [6] \\ \widehat{rr} = 0 \ \ldots\ldots [7] \end{cases}$

Substituting for the stresses their expressions in terms of the strains and using [1], [2], and [3] become, respectively:

$$(m + n)r\,\frac{d\delta}{dr} + n\,\frac{d}{dz}\left\{r\left(\frac{du}{dz} - \frac{dw}{dr}\right)\right\} = -\,\omega^2\rho r^2 \ldots [8]$$

$$(m + n)r\,\frac{d\delta}{dz} - n\,\frac{d}{dr}\left\{r\left(\frac{du}{dz} - \frac{dw}{dr}\right)\right\} = 0 \ \ldots\ldots [9]$$

By making use of Equations [8], [9], and [1], differential equations were obtained which when integrated gave the following solutions for δ, w, and u:

$$\left.\begin{aligned} \delta &= A + C(z^2 - {}^1/_2 r^2) - {}^1/_3\omega^2\rho r^2/(m + n) \\ w &= (A - 2\alpha)z - \frac{m}{3n}\,Cz^3 + {}^1/_6\left(\frac{m + n}{n}\,C - 8\xi\right)(2z^3 - 3zr^2) \\ u &= \frac{D}{r} + \alpha r + \xi\,(4z^2 r - r^3) + {}^1/_8\left(\frac{m}{n}\,C - \frac{\omega^2\rho}{m + n}\right)r^3 \end{aligned}\right\} \ \ldots\ldots [10]$$

where all the constants are independent.

By making use of Equations [4] to [7] and equations for stresses and strains, Chree determined values for the constants α, C, and ξ, and then proved that the surface conditions [4], [5], and [6] were satisfied. The only surface condition left is [7], but this he could not exactly satisfy, unless $m = n$, by means of the present solution. For, substituting the above values of the arbitrary constants, he obtained from the expression for $\widehat{rr}$ in terms of the strains

$$\widehat{rr}_{r\,=\,a} = {}^1/_2(3m - n)A - 2na^{-2}\,D - \frac{7m - n}{16m}\,\omega^2\rho a^2 - \frac{(m - n)\,(3m - n)}{4m(m + n)}\,\omega^2\rho z^2 \ldots\ldots [11]$$

and

$$\widehat{rr}_{r\,=\,a'} \ [= \text{a similar expression, replacing } a \text{ by } a'] \ldots [12]$$

It is obvious he could not make these stresses vanish for all values of z.

If then, according to Chree, the thickness 21 of the disk be of the same order of magnitude as the radius a, this failure renders the present method inapplicable; but when $1/a$ is small it is easy to obtain a solution which according to Saint-Venant and other eminent authorities must be very approximately exact except in the immediate neighborhood of the edges.

The principle this solution is based on is that of statically equivalent systems of loading. According to this principle, when a surface of an elastic solid has a small dimension—such as the thickness of a thin disk—all systems of surface forces which in their distribution along the small dimension are statically equivalent produce, except in the immediate neighbor-

hood of their points of application, practically identical strains and stresses. We may thus for practical purposes replace any system of surface forces over the small dimension by any statically equivalent system. Chree proceeded with the following method of solution, which he believed was in accordance with the view that would have been taken by Saint-Venant:

A and D were determined from the equations

$$\int_{-1}^{+1} \widehat{rr}_r = a \qquad dz = 0 \ldots\ldots\ldots\ldots [13]$$

$$\int_{-1}^{+1} \widehat{rr}_r = a' \qquad dz = 0 \ldots\ldots\ldots\ldots [14]$$

This still left normal stresses of the order of the square of the thickness over the edges, but the forces along each generator of an edge form a system in statical equilibrium.

For the annular disk he found from [13] and [14]

$$\left.\begin{aligned} A &= \frac{7m-n}{8m\,(3m-n)}\,\omega^2\rho(a^2+a'^2) + \frac{m-n}{6m(m+n)}\,\omega^2\rho 1^2 \\ D &= \frac{7m-n}{32mn}\,\omega^2\rho a^2 a'^2 \end{aligned}\right\} \ldots [15]$$

Also he determined C and ξ explicitly, and found α in terms of A. Thus all the constants of the solution were determined.

The physical results attainable from the solution he rendered more practically serviceable by replacing the m and n of this work by Young's modulus E and Poisson's ratio η. To express the values obtained above for the arbitrary constants in terms of E and η he used the following relations:

$$\left.\begin{aligned} n &= \tfrac{1}{2}\,E/(1+\eta) \\ m &= \tfrac{1}{2}\,E/(1-2\eta)\,(1+\eta) \end{aligned}\right\} \ldots\ldots\ldots [16]$$

Substituting the expressions found for the arbitrary constants in terms of E and η in [10], he found for the strains in an annular disk

$$\delta = \frac{\omega^2\rho(1-2\eta)}{4E}\left\{(3+\eta)(a^2+a'^2) - 2(1+\eta)r^2\right\} + \frac{\omega^2\rho}{3E}\,\frac{\eta(1-2\eta)\,(1+\eta)}{1-\eta}\,(1^2-3z^2) \ldots\ldots [17]$$

$$w = -\frac{\omega^2\rho}{4E}\,\eta\left\{(3+\eta)(a^2+a'^2)\,z - 2(1+\eta)r^2 z\right\} - \frac{\omega^2\rho}{3E}\,\eta^2\,\frac{1+\eta}{1-\eta}\,z(1^2-z^2) \ldots\ldots [18]$$

$$u = \frac{\omega^2\rho}{8E}\left\{(1-\eta)\,(3+\eta)\,(a^2+a'^2)r - (1-\eta^2)r^3 + \frac{a^2a'^2}{r} (1+\eta)\,(3+\eta)\right\} + \frac{\omega^2\rho}{6E}\,\eta\,(1+\eta)r\,(1^2-3z^2) \ldots [19]$$

From these strains he found for the stresses:

$$\widehat{rr} = \frac{\omega^2\rho}{8}\,(3+\eta)\left\{a^2+a'^2-r^2-\frac{a^2a'^2}{r^2}\right\} + \frac{\omega^2\rho}{6}\,\eta\,\frac{1+\eta}{1-\eta}\,(1^2-3z^2) \ldots\ldots [20]$$

$$\widehat{\varphi\varphi} = \widehat{rr} + \frac{\omega^2\rho}{4}\left\{(1-\eta)r^2 + (3+\eta)\,\frac{a^2a'^2}{r^2}\right\} \ldots [21]$$

According to the principle of statically equivalent surface forces, this solution does not strictly apply for values of r that differ from a or from a' by quantities which do not exceed several multiples of 1. In other words, it may possibly give values for the strains and stresses over the edges differing from the true values by terms of the order 1^2. Thus in determining the greatest values of the stresses, which occur at or near the inner edge in an annular disk, we are not warranted in retaining terms of this order of small quantities. Chree proposed in determining these quantities to neglect terms of 1^2 and z^2. This has been done when determining the stresses given on the curve sheets of this paper. The following equations were thus used:

$$\widehat{rr} = \frac{\omega^2\rho}{8}\,(3+\eta)\left\{a^2+a'^2-r^2-\frac{a^2a'^2}{r^2}\right\} \ldots [22]$$

$$\widehat{\varphi\varphi} = \widehat{rr} + \frac{\omega^2\rho}{4}\left\{(1-\eta)r^2 + (3+\eta)\,\frac{a^2a'^2}{r^2}\right\} \ldots [23]$$

H. E. Jenks in 1919 derived stress equations which agree with those by Chree for rotating disks. Simplified forms of the one for tangential tension at the edge of the hole were published in *Grits and Grinds*, June, 1923.

Discussion

H. W. Wagner.[5] There are some individuals connected with the grinding industry who are not in full agreement with all the predictions made by the classical theory. One idea encountered was that if two wheels of different diameters are operated at the same surface velocity, the smaller one running at the higher r.p.m. would be stressed higher than the larger one. Another was that maximum stress is at the periphery of the wheel rather than at the edge of the hole. There were also conflicting ideas on the effect of hole diameter upon stresses and breaking speed.

Existence of contradictory conceptions was considered sufficient reason for experimental verification by the photoelastic method. It is gratifying to note by the "Summary of Results" that photoelasticity as well as actual wheel breakages show, in so far as comparisons can be made, the same trends as does Chree's equation. Thus we have a more firmly established basis on which to work for more economical and safer use of grinding wheels and other rotating members.

A. M. Wahl.[6] The writer was rather surprised to find that there was a 20 per cent discrepancy in some cases between the results for rotating disks as given by the theory of elasticity and the results given by the photoelastic method. It would seem that it should be possible to obtain a closer agreement. The writer would like to suggest that a slight curvature of the stress-strain diagram of celluloid may possibly be responsible for part of the discrepancy.

It will be noted that the test results for grinding wheels having relatively small holes indicated that the computed breaking stresses were higher than in the case of wheels having relatively large holes. In the writer's opinion, this may be due to a slight curvature of the stress-strain diagram of the material itself. If the material obeys Hooke's law exactly, as assumed in elastic theory, then the computed breaking stress as given in Fig. 15 would be the *true* stress at failure; however, if the stress-strain curve of the material has a slight curvature, then the computed stresses at failure would be higher than the true stresses. Where

[5] Research Laboratories, Norton Company, Worcester, Mass. Mem. A.S.M.E.

[6] Research Engineer, Westinghouse Research Laboratories, East Pittsburgh, Pa. Jun. A.S.M.E.

the hole is relatively large, this effect will not occur to any great extent, and in such cases there should be better agreement between the breaking stress as computed and as obtained from test. It seems to the writer that the test results thus indicated that the grinding-wheel material does have a curved stress-strain characteristic.

APM-53-2

Vibrations of Railway Bridges

By J. N. GOODIER,[1] ANN ARBOR, MICH.

The paper contains a discussion of the nature of the approximately resonant vibrations of bridges due to the pulsating forces generated by unbalanced drivers, the influence of the motion of the train, and possible action of the springs; a group of proposed approximate formulas for predicting the natural frequencies of bridges, whether unloaded or supporting trains; a description of the methods of measuring natural frequencies; a list of the sources of damping; a theoretical demonstration of the effects of a small amount of distributed damping in restraining the forced vibration, and an appendix in which the frequency formulas are derived.

Resonant Vibrations

IT IS POSSIBLE for resonant vibrations to occur in a railway bridge through the coincidence of the frequency of the pulsating forces set up by the balance weights of the driving wheels, with the natural fundamental period of vertical oscillation of the whole structure, bridge and train together. It is unfortunate that this should happen within the range of practicable locomotive speeds, but such is the fact, and on account of its practical importance the subject has received much attention in recent years. In some cases another type of oscillation, which may be termed "quasi-torsional," occurs in which the two main girders do not oscillate in phase, and a section of the bridge is subject to a periodic shearing action. This is due to the vibrating forces being eccentrically applied, as must be the case for double-track bridges loaded on one track only. The shortest bridges—say, of 40 ft. span or less—do not vibrate in resonance, being so light and stiff that their natural frequencies are too high to be approached in any practicable running speed; and even for the longer bridges where the natural frequencies are attained in ordinary running, the resonance is never of a straightforward cumulative character—as it would be if the frequency of the pulsating forces could coincide exactly for a time with the natural frequency of the structure—for the train is moving over the bridge during the building-up of the oscillation, and thus continuously modifying the mass distribution, and with it the natural frequency of the vibrating system. Provided, therefore, that the train maintains a constant speed, exact resonance cannot be more than instantaneous. But in spite of this there may be such a close approach to resonance during the passage of the train that serious vibration does ensue.

This variation of natural frequency is clearly greater in bridges of moderate size, whose mass is not so great that the mass of the locomotive is negligible in comparison. However, the smallest bridge on which resonance can occur is precisely the one which is subject to the most intense vibrating forces, for its natural frequency will be so high as to correspond to the highest speed of the locomotive.

The bigger bridges have the lower natural frequencies, and consequently resonance will occur at lower locomotive speeds, and so at smaller pulsating forces, requiring a longer time of action to build up vibrations of serious magnitude. But, in fact, they are allowed this longer time of action under conditions appropriate for resonance, since the mass of the bridge is so much greater than the mass of the locomotive that the motion of the latter cannot disturb very much the agreement between the frequencies of the structure and the pulsating forces, and also because the speed is less and the bridge is longer.

In bridges of medium length the critical conditions are reached at speeds such as those attained by express locomotives, and in longer spans, by freight trains and slow passenger trains.

It may sometimes happen that the oscillations are so violent that the springs of the locomotive begin to act. Before this occurs the locomotive moves up and down solidly with the bridge, and the structure has one fundamental natural frequency at which there may be resonance. When the accelerations in the vibration become sufficiently high, the friction resisting spring movement is insufficient to communicate the acclerations to the spring-borne mass, and the latter no longer vibrates solidly with the bridge. The action of the springs introduces an additional degree of freedom, and the original fundamental natural frequency is replaced by two natural frequencies, both of which may fall within the practicable speed range. Fig. 1[2] shows a typical amplitude-frequency diagram for a bridge of a span of the order of 200 ft. The solid-line curve corresponds to locked springs, and the broken-line curve shows the effect of spring movement. The lowering of the original resonance peak in the latter case is due to the additional damping caused by spring friction. Fig. 2 is a typical amplitude-frequency diagram for a bridge of 120 ft. span. Here there is no intermediate range of frequencies for which the springs are locked.

The existence of two points of critical acceleration as the frequency changes is to be expected from the fact that the maximum acceleration in simple harmonic motion is proportional to the square of the frequency multiplied by the amplitude. As the frequency comes near to synchronism with the natural frequency of the structure, the amplitude grows until this product reaches the critical value, and the first resonance peak with spring movement is attained. After this, as the frequency rises the amplitude decreases, but later, since it is the square of the frequency which is involved, the acceleration again rises to the critical value.

Formulas for Natural Frequencies

It is of course important that there should be some means of predicting the natural frequencies of bridges, whether loaded or not, and whether spring movement occurs or not.

---

[1] University of Michigan. Mr. Goodier was educated in England at Downing College, Cambridge University, where he studied mathematics and mechanical sciences. He took the degree of B.A. through the Mechanical Sciences Tripos, with the Rex Moir Prize for general distinction and the Ricardo Prize for thermodynamics. For two years he worked as a research student under Prof. C. E. Inglis, F.R.S., on problems of elasticity, and applied mechanics, stress distributions, bridge vibrations, and the phenomenon of rail corrugation, and together with H. S. Sayles, B.A., was awarded the John Steddy Winbolt Prize of Cambridge University for a dissertation on "Some Problems of Plane Stress." He was elected a Commonwealth Fellow in 1929. He then went to Ann Arbor to continue work under the supervision of Professor Timoshenko.

Contributed by the Applied Mechanics Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] Figs. 1 to 6, with accompanying formulas, are reproduced by courtesy of Prof. C. E. Inglis, F.R.S., from his paper "Oscillations in a Bridge Caused by the Passage of a Locomotive," in Proc. Roy. Soc. (London), vol. 118, series A, 1928.

For girders, or bridges in which mass and rigidity are uniformly distributed, we may apply the simple beam formula:

$$(\text{natural frequency})^2 = n^2 = \frac{\pi^2 gEI}{4ml^4} \quad \ldots\ldots [1]$$

m being the mass per unit length, l the span, and EI having the usual significance.

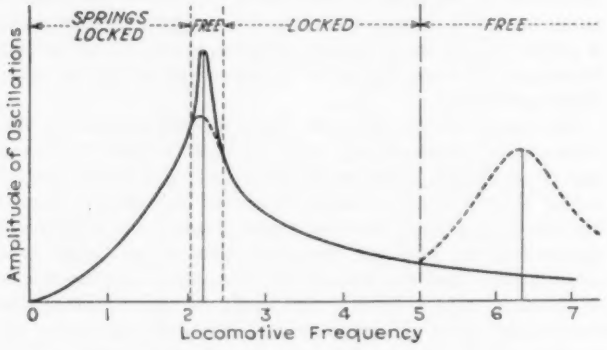


FIG. 1 TYPICAL AMPLITUDE-FREQUENCY DIAGRAM FOR A BRIDGE OF AROUND 200 FT. SPAN

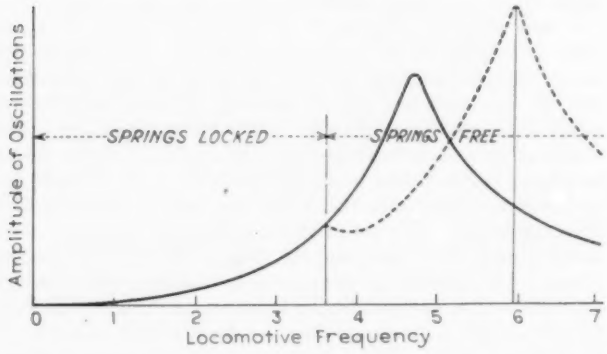


FIG. 2 TYPICAL AMPLITUDE-FREQUENCY DIAGRAM FOR A BRIDGE OF 120 FT. SPAN

Another simple formula of somewhat less restricted application, not requiring an estimate of EI, is

$$n^2 = \frac{gF}{2\pi^2 m_B} \quad \ldots\ldots [2]$$

where F is the central deflection per unit central load, and m_B is the total mass, assumed uniformly distributed. The result of applying this to three bridges for which data were available, together with the observed frequencies for comparison, are given below. The values of F were taken from actual deflection records, but they are characteristics easily obtained for any projected design.

Span in ft.	Mass in tons	F tons per ft.	Calculated frequency	Observed frequency
169	325	3690	4.23	4.00
210	626	5160	3.78	3.70
263	460	2340	2.85	2.86

Three examples are hardly sufficient to test the validity of the formula, but actual measurements of bridge frequencies do not appear to be very numerous.

The corresponding formula for a bridge whose mass cannot be regarded as uniformly distributed is

$$n^2 = \frac{gF}{4\pi^2 \int_0^l m \sin^2 \frac{\pi x}{l}\, dx} \quad \ldots\ldots [3]$$

When the mass of the locomotive is taken into account as a concentrated mass m_1 at a distance a from one end,

$$n^2 = \frac{gF}{2\pi^2 \left[m_B + 2m_1 \sin^2 \frac{\pi a}{l} \right]} \quad \ldots\ldots [4]$$

for a bridge of uniform mass distribution, and otherwise

$$n^2 = \frac{gF}{4\pi^2 \left[\int_0^l m \sin^2 \frac{\pi x}{l}\, dx + m_1 \sin^2 \frac{\pi a}{l} \right]} \quad \ldots\ldots [5]$$

The corresponding formulas when there are several loads on the bridge, or when it is desired to take into account individually the axle loads of the locomotive, say, m_1 at a distance a_1 from the end of the bridge, m_2 at a distance a_2, m_r at a distance a_r, etc., are

$$n^2 = \frac{gF}{2\pi^2 \left[m_B + 2 \Sigma\, m_r \sin^2 \frac{\pi a_r}{l} \right]} \quad \ldots\ldots [6]$$

and

$$n^2 = \frac{gF}{4\pi^2 \left[\int_0^l m \sin^2 \frac{\pi x}{l}\, dx + \Sigma\, m_r \sin^2 \frac{\pi a_r}{l} \right]} \quad \ldots\ldots [7]$$

The indications of formula [4] for the same three bridges loaded at the middle with a locomotive weighing 108 tons and having a wheelbase of 44 ft. 7½ in., together with the measured frequencies, are as follows:

Calculated	3.33	3.15	2.38
Observed	3.10	3.10	2.40

The ratio (loaded frequency)/(unloaded frequency) under the conditions of formulas [2] and [4] appears as

$$\frac{\text{Loaded frequency}}{\text{Unloaded frequency}} = \sqrt{\frac{m_B}{m_B + 2m_1 \sin^2 \frac{\pi a}{l}}} \quad \ldots\ldots [8]$$

This formula was obtained by C. E. Inglis, F.R.S., by a method different from that given in the Appendix.

The two natural frequencies which replace the above fundamental frequencies when the locomotive springs are in action are n_1 and n_2, n_1^2 and n_2^2 being the roots of the quadratic in n^2,

$$n^4 \left[1 + \frac{2m_0}{m_B} \sin^2 \frac{\pi a}{l} \right] - n^2 \left[n_0^2 + 2n_s^2 \frac{m_L}{m_B} \sin^2 \frac{\pi a}{l} \right] + n_s^2 n_0^2 = 0 \quad \ldots\ldots [9]$$

where m_0 = non-spring-borne mass of the locomotive
m_L = total mass of the locomotive
n_0 = fundamental natural frequency of the unloaded bridge
n_s = frequency of the spring mass of the locomotive on its springs, the non-spring-borne mass being fixed
m_B = mass of the bridge, as before
a = distance of the locomotive from one end.

For a single-track bridge with the following characteristics: span = 120 ft., m_B = 200 tons, n_0 = 6, and a locomotive with

the characteristics $m_e = 20$ tons, $m_L = 70$ tons, $n_s = 3$, situated at the middle of the bridge, the quadratic becomes

$$1.2n^4 - 42.6n^2 + 324 = 0$$

and the two frequencies obtained by solving this are 3.32 and 4.94.

The frequency if the locomotive springs remained locked would be 4.60, so that the two above frequencies lie on either side of this. For the lower frequency the spring mass is in phase with the bridge, and for the higher frequency it is in antiphase, its effect being the same as a reduction of bridge mass.

Measurement of Natural Frequencies

Two methods are available. The first is to use a recording deflectometer to make a diagram of the vibration after the bridge has been disturbed by the passage of the locomotive. The frequency of the residual oscillations can then be counted on the diagram. The second is the method of the "bridge oscillator." This is a machine capable of vibrating the bridge without altering appreciably its natural frequency.[3] In the experiments of the British Bridge Stress Committee, it took the form of a truck upon which were mounted two wheels geared together to rotate in opposite directions. Heavy masses are bolted eccentrically to the wheels. When rotated by an electric motor these provide vertical pulsating forces on the bridge similar to those due to the balance weights of locomotives, except that in the case of the oscillator they remain in a fixed position. Arrangements are provided to regulate the speed and to measure the power supplied to the motor, the latter for investigating the damping. The oscillator is run at varying speeds, and the natural frequency of the bridge and oscillator together is located by means of the resonance peak on the amplitude-frequency diagram.

Damping

In bridges where it is possible for resonance to be approximately attained, damping of various kinds is responsible for restricting the amplitudes of vibration to safe limits. The sources of damping may be enumerated as

1. Lack of perfect elasticity in the structure
2. Track ballast
3. Friction at the end bearings
4. Lack of rigidity in the abutments and consequent communication of energy to the ground
5. Friction of oscillating springs of locomotive and other units
6. Trains.

The relative importance of the first four appears to be still uncertain. The report of the British Bridge Stress Committee described the fifth as perhaps the most important of all the single sources of damping. This of course has reference to the British railways.

The damping effect of a train following a locomotive is appreciable, but except on the longer spans the locomotive approaches the middle of the bridge and a large oscillation is built up before the train is sufficiently far on the bridge to have any considerable damping effect. Its influence in such cases is mainly on the residual oscillations, particularly when the train is drawn by two coupled locomotives.

The undamped oscillations produced by a concentrated pulsating force $P \sin 2\pi nt$ moving at uniform speed along a girder are given approximately by

$$y = \frac{D}{2}\left[\frac{\cos 2\pi(N-n)t - \cos 2\pi n_0 t}{1-\left(\frac{N-n}{n_0}\right)^2} - \frac{\cos 2\pi(N+n)t - \cos 2\pi n_0 t}{1-\left(\frac{N+n}{n_0}\right)^2}\right]\sin\frac{\pi x}{l} \quad \ldots\ldots [10]$$

where y is the deflection at a point distant x from one end, $n = v/2l$, v is the speed, l the span, n_0 the natural frequency of the girder, and D the central deflection due to a statical force P at the center. In addition to this there is an oscillation due to the movement of the gravity load of the locomotive. This is approximately defined by

$$y = \delta\left[\sin\frac{\pi vt}{l} - \frac{v}{2ln_0}\sin 2\pi n_0 t\right]\sin\frac{\pi x}{l}$$

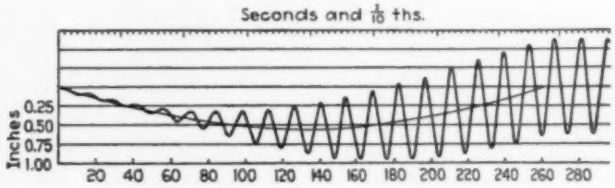


Fig. 3 Undamped Synchronous Oscillation

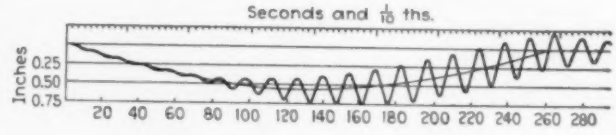


Fig. 4 Damped Synchronous Oscillation

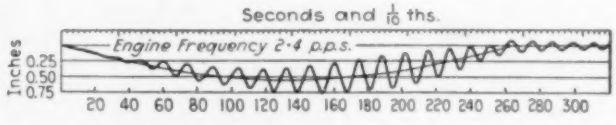


Fig. 5 Damped Synchronous Oscillation, Including Effect of Locomotive Mass

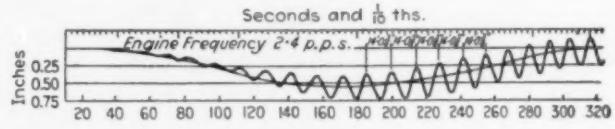


Fig. 6 Observed Oscillation

Figs. 3–6 Oscillations of Midpoint of Bridge Traversed by a Locomotive

(Span of bridge, 262.5 ft.; total mass, 480 tons; unloaded frequency, 2.88 oscillations per sec. Locomotive generates a pulsating force of 0.576 ton at 1 rev. per sec.; driving wheels, 14.0625 ft. in circumference; total weight, 107.65 tons.)

where δ is the central deflection due to the gravity load stationary at the center. This latter component is small and at worst can only contribute about 10 per cent of the whole vibration.

Fig. 3 shows the oscillation given by these formulas for the midpoint of a bridge of span 262.5 ft., total mass 460 tons, and unloaded frequency 2.88 oscillations per second, traversed by a locomotive generating a pulsating force of 0.576 ton at one revolution per second, with driving wheels 14.0625 ft. in circumference, and a total weight of 107.65 tons. The central deflection produced by the locomotive standing at the middle of the bridge is 0.55 in. The natural frequency of bridge and locomotive together is then 2.4, and the engine speed is 2.4 r.p.s.,

[3] See R. Bernhard and W. Späth, "Rein dynamische Verfahren zur Untersuchung der Beanspruchungen von Bauwerken," "Der Stahlbau," 1929, Heft 6.

about 23 m.p.h. The oscillation is building up as long as the locomotive is on the bridge, and finally the maximum deflection exceeds the crawl deflection by 75 per cent. The "crawl deflection" is the deflection due to the locomotive moving slowly over the bridge, being thus equivalent to a static load in various positions.

The effect of damping can be taken into account in such calculations if it is assumed to be equivalent to a distribution of force along the bridge proportional at any point to the velocity. For analytical convenience it may be represented as a force per unit length $4\pi n_b m \frac{\partial y}{\partial t}$, where m is the mass per unit length and n_b a damping coefficient. Then the forced oscillation becomes

$$y = \frac{D}{2}\left[\frac{\cos\{2\pi(N-n)t-\phi\}}{\left[\left\{1-\left(\frac{N-n}{n_0}\right)^2\right\}^2+\left\{\frac{n_b(N-n)}{n_0}\right\}^2\right]^{\frac{1}{2}}} - \frac{\cos\{2\pi(N+n)t-\psi\}}{\left[\left\{1-\left(\frac{N+n}{n_0}\right)^2\right\}^2+\left\{\frac{n_b(N+n)}{n_0}\right\}^2\right]^{\frac{1}{2}}}\right]\sin\frac{\pi x}{l}$$

where

$$\tan\phi = \frac{n_b(N-n)}{n_0^2-(N-n)^2} \text{ and } \tan\psi = \frac{n_b(N+n)}{n_0^2-(N+n)^2}$$

The complete oscillation is obtained by superimposing a free oscillation of the approximate form

$$y = e^{-2\pi n_b t}\,[A\sin 2\pi n_0 t + B\cos 2\pi n_0 t]\sin\frac{\pi x}{e},$$

with A and B adjusted to make the total displacement and velocity zero when the locomotive comes upon the bridge. There is also the small oscillation set up by the movement of the gravity load.

A value $n_b = 0.075$ causes the free oscillations to die out in a geometric progression with the common ratio 0.82. With this amount of damping the theoretical oscillation in Fig. 3 is modified to that shown in Fig. 4.

No account is taken in these formulas of the influence of the locomotive mass in destroying perfect resonance. The introduction of this, which causes much increased complication in the analysis, gives an oscillation represented in Fig. 5.

Fig. 6 is the oscillation actually observed.

Appendix No. 1

FREQUENCY FORMULAS

THE curve of the elastic line of the bridge at any instant during vibration may be expressed, whatever its form, by a Fourier series of the type

$$y = A_1\sin\frac{\pi x}{l} + A_2\sin\frac{2\pi x}{l} + \dots$$

where y is the deflection at a distance x from one end.

Usually it will be sufficient to take the first harmonic and thereby assume that the curve of deflection is a sine curve. This is the fact for a uniform unloaded beam with hinged ends. The deflection at any point on the bridge will be simple harmonic in time. We may therefore write

$$y = \delta\sin\frac{\pi x}{l}\sin 2\pi n t$$

and n will then be the frequency of vibration, the quantity we wish to evaluate; δ will be the amplitude of the middle of the bridge.

The maximum velocity of a point at a distance x from one end will be

$$2\pi n\delta\sin\frac{\pi x}{l}$$

occurring when the bridge is passing through its equilibrium configuration. The whole of the energy of vibration at that instant is kinetic. For the whole bridge it will be

$$\int_0^l \frac{1}{2}\frac{m\,dx}{g}\left[2\pi n\delta\sin\frac{\pi x}{l}\right]^2$$

In addition to this we have contributions from loads on the bridge. Suppose these are m_1 at a distance a_1 from the end, m_2 at a_2, and so on.

The maximum velocity of m_1 is

$$2\pi n\delta\sin\frac{\pi a_1}{l}$$

and its kinetic energy then is

$$\frac{1}{2}\frac{m_1}{g}\left[2\pi n\delta\sin\frac{\pi a_1}{l}\right]^2$$

The kinetic energy for all the loads will thus be

$$\frac{2\pi^2 n^2\delta^2}{g}\left[m_1\sin^2\frac{\pi a_1}{l} + m_2\sin^2\frac{\pi a_2}{l} + \dots\right.$$

or

$$\frac{2\pi^2 n^2\delta^2}{g}\sum m_r\sin^2\frac{\pi a_r}{l}$$

The total energy of the whole vibrating system, expressed as maximum kinetic energy, is accordingly

$$\frac{2\pi^2 n^2\delta^2}{g}\int_0^l m\sin^2\frac{\pi x}{l}\,dx + \frac{2\pi^2 n^2\delta^2}{g}\sum m_r\sin^2\frac{\pi a_r}{l} \dots [11]$$

When the bridge has swung to an extreme position in the vibration, its velocity anywhere is momentarily zero, and all the energy of vibration is stored as strain energy. We require to estimate the energy in this form.

Suppose a central force P is required to produce the central deflection δ. The deflection curve would of course not be exactly the same as the deflection curve in a vibration, but the difference is not very serious. The strain energy of the bridge in static deflection will be $\frac{1}{2}P\delta$. Taking this as an estimate of the maximum strain energy in vibration—that is, the total energy of vibration—we may equate it to [11], or

$$\frac{1}{2}P\delta = \frac{2\pi^2 n^2\delta^2}{g}\left[\int_0^l m\sin^2\frac{\pi x}{l}\,dx + \sum m_r\sin^2\frac{\pi a_r}{l}\right]$$

so that

$$n^2 = \frac{gF}{4\pi^2\left[\int_0^l m\sin^2\frac{\pi x}{l}\,dx + \sum m_r\sin^2\frac{\pi a_r}{l}\right]}$$

where $F = P/\delta$, the central force required for unit central deflection.

This is formula [7], the most general of the formulas quoted in the paper. Formulas [2] to [6] are particular cases of it

and are very simply derived. For the uniform bridge the first term in the denominator is

$$4\pi^2 m \int_0^l \sin^2 \frac{\pi x}{l} = 4\pi^2 \frac{1}{2} ml = 4\pi^2 \frac{1}{2} m_B$$

and for the unloaded bridge the second term in the denominator vanishes.

Sprung Locomotive. It may be shown that the oscillation caused in a uniform beam by a concentrated force P, which may be varying with time, at a distance a from one end of the bridge is approximately defined by the differential equation

$$EI \frac{\partial^4 y}{\partial x^4} + m \frac{\partial^2 y}{\partial t^2} = \frac{2P}{l} \sin \frac{\pi a}{l} \sin \frac{\pi x}{l} \quad \ldots\ldots [12]$$

neglecting damping, which has only a very small effect upon frequencies.

Consider the system represented in Fig. 7, a mass m_0 rigidly

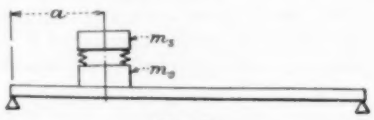


FIG. 7

attached to the beam, with a mass m_s attached to it by springs, both masses being at a distance a from one end of the bridge. Let h be the deflection of the beam under m_0 and m_s at any instant t, and z the deflection of m_s from the position of equilibrium. If μ is the spring constant, the compressive force of the spring is then $\mu(z - h)$. If P is the force between m_0 and the bridge,

$$m_0 \frac{d^2h}{dt^2} = \mu (z - h) - P$$

and

$$P = \mu (z - h) - m_0 \frac{d^2h}{dt^2}$$

This is the variable force on the bridge at a distance a from the end, so that Equation [12] gives

$$EI \frac{\partial^4 y}{\partial x^4} + m \frac{\partial^2 y}{\partial t^2} = \frac{2}{l} \left[\mu (z - h) - m_0 \frac{d^2h}{dt^2} \right] \sin \frac{\pi a}{l} \sin \frac{\pi x}{l} \quad \ldots.[13]$$

For the motion of m_s we have

$$m_s \frac{d^2z}{dt^2} = -\mu (z - h) \quad \ldots\ldots\ldots\ldots [14]$$

These equations are satisfied by oscillations of the same frequency, of the form

$$y = A \sin \frac{\pi x}{l} \sin 2\pi nt, \qquad z = B \sin 2\pi nt$$

Then

$$h = A \sin \frac{\pi a}{l} \sin 2\pi nt$$

and n is the natural frequency we wish to evaluate.

Substituting these expressions for y, z, and h in Equations [13] and [14] we obtain

$$A \left[EI \frac{\pi^4}{l^4} - 4\pi^2 n^2 m - \frac{8\pi^2 m_0}{l} n^2 \sin^2 \frac{\pi a}{l} + \frac{2\mu}{l} \sin^2 \frac{\pi a}{l} \right] = B \frac{2\mu}{l} \sin \frac{\pi a}{l}$$

and

$$A\mu \sin \frac{\pi a}{l} = B[\mu - 4\pi^2 n^2 m_s]$$

Then, by elimination of the ratio A/B,

$$n^4 16\pi^4 \left[1 + \frac{2m_0}{m_B} \sin^2 \frac{\pi a}{l} \right] - n^2 4\pi^2 \left[\frac{\mu}{m_s} \left(1 + \frac{2m_0}{m_B} \sin^2 \frac{\pi a}{l} \right) + \left(\frac{EI}{m} \frac{\pi^4}{l^4} + \frac{2\mu}{m_B} \sin^2 \frac{\pi a}{l} \right) \right] + \frac{\mu}{m_s} \cdot \frac{\pi^4 EI}{ml^4} = 0$$

where $m_B = ml$, the total mass of the bridge. If n_s is the natural frequency of the spring mass m_s on its springs, and n_0 that of the unloaded beam,

$$\mu = 4\pi^2 n_s^2 m_s \qquad \text{and} \qquad \frac{\pi^4 EI}{ml^4} = 4\pi^2 n_0^2$$

Using these, the equation becomes

$$n^4 \left[1 + \frac{2m_0}{m_B} \sin^2 \frac{\pi a}{l} \right] - n^2 \left[n_0^2 + n_s^2 \left(1 + \frac{2m_L}{m_B} \sin^2 \frac{\pi a}{l} \right) \right] + n_0^2 n_s^2 = 0$$

where $m_L = m_0 + m_s$, the total mass of the locomotive. This is Equation [9] of the paper, and is a quadratic in n^2, determining the natural frequencies for the system.

SHORT BIBLIOGRAPHY

1 Report of the Bridge Stress Committee, Department of Scientific and Industrial Research, His Majesty's Stationery Office, London, 1928.

2 C. E. Inglis, "Oscillations in a Bridge Caused by the Passage of a Locomotive," Proc. Roy. Soc., London, vol. 118, series A, 1928.

3 S. Timoshenko, "Vibrations of Bridges," A.S.M.E. Annual Meeting, 1927, Railroad Division.

4 C. E. Inglis, "Theory of Transverse Oscillations in Girders, and Its Relation to Live-Load and Impact Allowances," Proc. Inst. C. E., London, vol. 218, p. 225, 1924.

5 R. C. J. Howland, "Transverse Oscillations in Girders," Inst. C. E., London, Selected Papers, 1924, no. 23.

Discussion

J. B. HUNLEY.[4] The author has introduced some interesting formulas as to the frequency of loaded and unloaded spans, and has discussed in the body of the paper and has developed in the appendix a formula relating to the effect on the oscillation due to the sprung weight of the locomotive. In the tests carried on by the Bridge Stress Committee, this was found to be quite an important factor in computing impact effects on spans of medium length, which are very common on American railroads.

Unfortunately, in the impact tests made by the American Railway Engineering Association in 1910 or 1911, no attempt was made to segregate the different factors affecting impact, one of which was the resonance of the sprung weight of the locomotive or cars, nor was any attempt made to select or classify spans in reference to their stiffness and mass, nor were the loadings or spans selected in such a way as to have some fixed or known ratio of load to capacity of span.

Just as soon as conditions will permit, it is hoped that further tests will be made, taking into account and if possible segregating the effects of the various factors producing impact, and the effect of a spring-borne load of American locomotives should be given some study. It would seem that this might also apply to train loads. While it is true that we have no hammer blow resulting

[4] Engineer, Bridges and Structures, Cleveland, Cincinnati, Chicago & St. Louis Railway, Cincinnati, Ohio.

from counterbalance weights on car wheels, many of these car wheels have flat spots, and with the trend toward heavy cars, it frequently happens that the train load produces a greater static effect on the structure than the engine handling the train, and it is not difficult to imagine a train running at such a speed over a bridge that the rough wheels would produce synchronism between the sprung weight of the car and the loaded frequency of the span, and with long trains, such as are operated in this country, we might have resonance continued and oscillations amplified to a greater extent than with the passage of the locomotive alone, which at the worst can produce but a few impulses on the span.

The paper emphasizes and clarifies some of the points discussed in the report of the Bridge Stress Committee and should suggest to American engineers lines of research and study to enable them to determine, with some degree of accuracy, the impact-producing effects of American equipment. Our present impact formulas are at the best but a crude approximation, fit only for the design of new structures, and they are altogether unsuitable for determining the capacity of existing structures under any given loading.

B. S. CAIN.[5] The paper has rather a theoretical appearance, but the author could hardly show in so small a space that it is actually based on tests and that every detail of the theory accurately represents the facts. Until a few years ago the theory of bridge impact was not in this happy state, and the "impact allowance" generally used was based on facts not properly understood and was really no more than a big factor of ignorance. This old impact allowance is today a part of the railroad engineering standards of this country. Papers like the present one are valuable in showing that most of the impact effect follows definite laws and that, with the help of the theory, we can interpret tests so that no factor of ignorance need remain in our results. Bridge tests today are much easier to carry out and more reliable than they were a year or two ago. We have available simple, accurate stress recorders, operating electrically and giving as many simultaneous records of stress in different members as may be required.

These tests have value in another connection. The paper shows that the principal source of bridge impact is the unbalance of steam-locomotive driving wheels. With an electric train the impact is very small. There are today in this country several programs of main-line railroad electrification announced or already under way. It is most important to realize that electrification will reduce many bridge restrictions both on weight and particularly on speed. It must also be realized that there can be made some simple tests which will show exactly the capacity of old bridges for modern trains, and facilities at present available can be used to the greatest possible extent, employing "facts" instead of "factors."

S. TIMOSHENKO.[6] The problem of dynamical deflections of bridges has always interested engineers since the beginning of railway-bridge construction. The general feeling was that under the action of a load moving with a high velocity larger deflections and larger stresses are produced than if the same load is acting statically. A theoretical solution for a bridge of uniform cross-section and subjected to the action of a constant vertical load moving with a constant velocity v shows that there is an increase in the deflection due to the velocity of motion. If δ_d and δ_s are dynamical and statical maximum deflections, T the period of the fundamental type of natural vibration of the bridge, and $T_1 = 2l/v$ = twice the time it takes the force to pass over the bridge, then

$$\delta_d = \delta_s \frac{1}{1 - \frac{T}{T_1}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [15]$$

For usual conditions T/T_1 is small, and from Equation [15] it can be concluded that a smoothly running load does not produce a substantial increase in deflection.

Much more important is the action on a bridge of a pulsating force produced by unbalanced weights of a moving locomotive. If this force is in resonance with the natural vibration of the bridge and if damping is neglected, the amplitude of vibration produced under these conditions is:

$$\delta = \delta_0 \frac{T_1}{T} \quad \ldots\ldots\ldots\ldots\ldots\ldots [16]$$

In this equation T and T_1 have the same meaning as before, and δ_0 is the statical deflection which is produced if the maximum value of the pulsating force is applied at the middle span of the bridge.

The deflection given by Equation [16] may be a considerable one, especially for a comparatively small bridge whose frequency of natural vibration approaches the largest practical value for the number of revolutions per second of the drivers of a locomotive. Taking, for instance, the maximum number of revolutions per second of drivers as 6, it is found that pulsating forces have the largest effect on bridges with spans around 100 ft.

From the foregoing equations it is seen that the dynamical effect of a moving load depends on the frequency of natural vibration of the bridge. One can predict the magnitude of vibration only if the frequency of natural vibration of the bridge is known. The author gives several approximate formulas for calculating this frequency.

In the derivation of these formulas the deflection curve of the vibrating bridge is taken as a sine curve in calculating the kinetic energy of the bridge, and the deflection curve produced by a load at the middle is taken in calculating the strain energy. Although the formulas given are in satisfactory agreement with experiments, it seems more logical in this case to use the well-known Rayleigh method and take as a basis of calculation the actual deflection curve of the bridge. If y_1, y_2, y_3. . . . . are the deflections at the points 1, 2, 3 . . . . . along the span of the bridge, and Q_1, Q_2, ... are the portions of the weight corresponding to these points, the frequency n of the fundamental type of vibration is given by the equation:

$$n^2 = \frac{g\,(Q_1y_1 + Q_2y_2 + Q_3y_3 + \ldots.)}{Q_1y_1^2 + Q_2y_2^2 + Q_3y_3^2 + \ldots..} \quad \ldots\ldots\ldots [17]$$

This equation is used very often in calculating the frequencies of the natural lateral vibration of shafts and long rotors of modern turbo-generators.

The writer would like to mention a misprint in the line following Equation [2] for calculating frequency. F is the force necessary to produce unit deflection and not the deflection produced by unit force, as stated in the paper. It should be mentioned, also, that the notations m and m_B in Equations [2] to [9] are weights. In the appendix, in the derivation of Equation [13], the same notations are used for masses.

Equation [16] gives an exaggerated value for the effect of the pulsating load. The actual amplitude of vibration set up by an unbalanced locomotive is much smaller. This is due to damping of various kinds and to the effect of the moving masses which may change considerably the initial frequency of the natural vibration

[5] General Electric Company, Erie, Pa.

[6] Professor of Mechanical Engineering, University of Michigan, Ann Arbor, Mich. Mem. A.S.M.E.

of the bridge. The effect of damping and the effect of moving masses are discussed in the paper by the author.

It is shown that, assuming damping to be a force distributed along the length of the bridge proportional at each cross-section to the velocity of the vibratory motion, a comparatively simple solution can be obtained for the damped vibration produced by a pulsating force.

In this manner a much better agreement with experiment is obtained than that given by Equation [2].

In discussing the effect of moving masses on the frequency of natural vibration, the author considers the case of a sprung locomotive and shows that, by introducing springs between the mass of the locomotive and the bridge, two types of vibration are possible instead of the one fundamental type of vibration of the bridge. Equations for calculating the two corresponding frequencies are also given by the author. The diagram in Fig. 1, illustrating the results obtained, does not seem very satisfactory. It does not show that the frequency of the lower type of vibration is smaller than that for locked springs. The writer also would like to mention that Equation [9] for calculating the two frequencies does not coincide with the equation derived in the appendix.

The effect of moving loads on the cumulative effect of a pulsating force is only illustrated by Fig. 5, without giving any discussion of this important question.

The problem of torsional vibration and lateral vibration of bridges, although of great practical importance, is left without discussion in the paper. It is only mentioned that such vibration may occur in double-track bridges. However, torsional and lateral vibration may occur also in single-track bridges under the action of unbalanced weights and variable tractive efforts.

In connection with the development of a more rational formula for impact effect on bridges a considerable amount of experimental and theoretical work has been done in England, Germany, and Russia. There is no doubt that this kind of work must be developed in this country also, and it is desirable that the author continue his investigations on bridge vibration. The study of lateral vibrations and a comparative study of the vibration produced by steam engines and by electric locomotives are especially interesting at this time. It is important also to develop experimental work on the vibration of bridges to check the results of theoretical investigations.

George E. Beggs.[7] It would be interesting to know whether the author is of the opinion that the vibration period of a bridge can be predicted from the measured vibration period of a loaded model of the bridge. In the accompanying picture, Fig. 8, is shown a steel model of the Mt. Hope Suspension Bridge. If it be assumed that this model is a true scalar reduction of the larger structure in the ratio n to unity, and, further, if it be assumed that the weights shown suspended from this model are such that the unit stresses and unit strains in the model are equal to the corresponding dead-load unit stresses and strains in the full-sized structure, then the problem is to determine what power of the reduction factor n should be applied to the measured natural period of vertical vibration of the loaded model in order to predict correctly the corresponding period of vertical vibration of the actual bridge.

The deformations and stresses in the Mt. Hope bridge have been quantitatively checked by this model, by employing very simple conversion formulas, and one can reasonably expect that there must be a simple relationship between the measured period of this model and the corresponding period of the large suspension bridge.

[7] Professor of Civil Engineering, Princeton University, Princeton, N. J.

R. Eksergian.[8] The paper contains two distinct and valuable contributions to the subject of vibration of bridges. First, by a direct application of the energy principle, together with representation of deflections by a trigonometric series, a simple procedure is outlined for the determination of the natural frequency of a bridge with distributed concentrated masses, and, second, consideration is given to the additional degree of freedom due to the spring suspension of the locomotive.

While the simplified assumption regarding the locomotive as a concentrated mass with an equivalent single-spring suspension

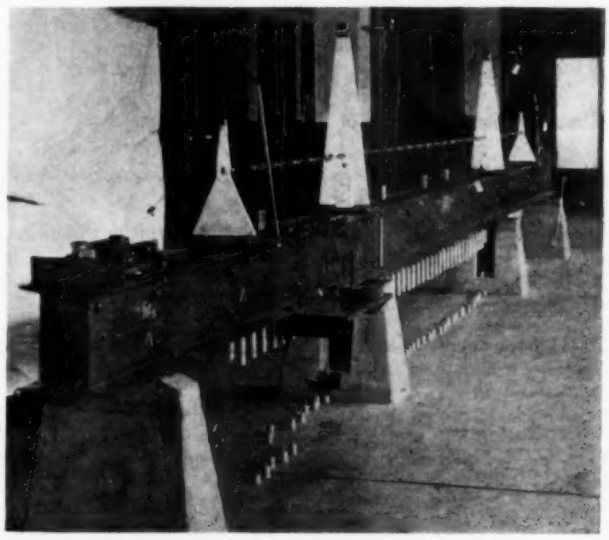


Fig. 8 Cable-Testing Apparatus on Model of Mt. Hope Suspension Bridge

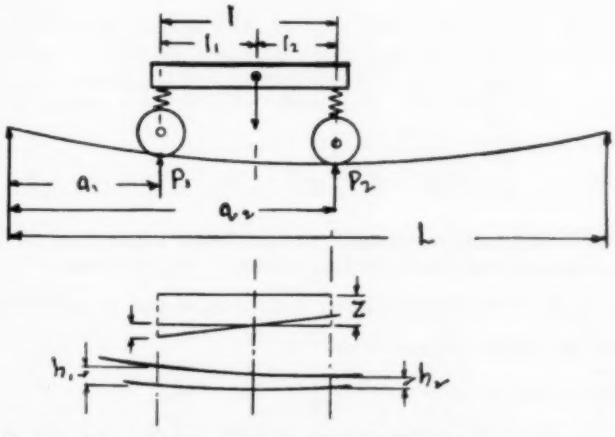


Fig. 9 Spring Suspension of Four-Wheel Vehicle, With a Front and Rear Spring Support

is the first logical step in attacking the problem, the writer believes a better approximation is needed due to the peculiar suspension and considerable spread of a modern locomotive over moderate-length spans.

I—Elementary Spring Suspension

Let us first consider the very simple case of an elementary spring suspension, i.e., of a four-wheel vehicle with a front and rear spring support, as shown in Fig. 9. The vehicle chassis has two degrees of freedom, viz., an angular motion represented by

[8] Consultant, Engineering Department, E. I. du Pont de Nemours & Co., Wilmington, Del. Mem. A.S.M.E.

the coordinate ϕ_r, and the vertical translation component of the c. of g., Z_r.

Let l_1 and l_2 = the respective distances from center of gravity to the rear and front spring supports
h_{1r} and h_{2r} = the respective deflections of the bridge under the rear and front wheels
a_1 and a_2 = the position of the wheels from the left abutment
M, m_1, and m_2 = the respective masses of the vehicle chassis and the rear and front wheels
μ_1 and μ_2 = the respective spring constants.

The compressions of the rear and front springs are

$$Z_r + l_1\phi_r - h_{1r} \quad \text{and} \quad Z_r - l_2 - h_{2r}$$

so that the potential energy of the spring system is:

$$V = \frac{1}{2}\mu_1(Z_r + l_1\phi_r - h_{1r})^2 + \frac{1}{2}\mu_2(Z_r - l_2 - h_{2r})^2$$

The work of the applied forces on the total vehicle, which includes the work of the reactions of the wheel supports, is:

$$\delta W = Mg\delta Z_r - P_{1r}\delta h_{1r} - P_{2r}\delta h_{2r} + m_1 g\delta h_{1r} + m_2 g\delta h_{2r}$$

The kinetic energy of the vehicle is

$$T = \frac{1}{2}M(\dot{Z}_r^2 + k_g^2\dot{\phi}_r^2) + \frac{1}{2}m_1\dot{h}_{1r}^2 + \frac{1}{2}m_2\dot{h}_{2r}^2$$

where k_g is the radius of gyration of the chassis about its center of gravity. We have, then,

$$M\ddot{Z}_r = -\frac{\partial V}{\partial Z_r} + Mg = \mu_1(Z_r + l_1\phi_r - h_{1r}) - \mu_2(Z_r - l_2 - h_{2r}) + Mg$$

$$Mk_g^2\ddot{\phi}_r = -\frac{\partial V}{\partial \phi_r} = -\mu_1(Z_r + l_{1r} - h_{1r})l_1 + \mu_2(Z_r - l_2\phi_r - h_{2r})l_2$$

$$m_1\ddot{h}_{1r} = -\frac{\partial V}{\partial h_{1r}} + m_1 g - P_1 = \mu_1(Z_r + l_1\phi_r - h_{1r}) + m_1 g - P_{1r}$$

$$m_2\ddot{h}_2 = -\frac{\partial V}{\partial h_2} + m_2 g - P_2 = \mu_2(Z_r - l_2\phi_r - h_{2r}) + m_2 g - P_{2r}$$

It is preferable to separate the equilibrium values from the oscillations about the equilibrium positions. We note that:

$Z_r = Z_0 + Z$, $\phi_r = \phi_0 + \phi$, $h_{1r} = h_{01} + h_0$, and $h_{2r} = h_{02} + h_2$

and the equilibrium conditions

$$Mg = \mu_1(Z_0 + l_1\phi_0 - h_{01}) + \mu_2(Z_0 - l_2\phi_0 - h_{02}),$$

$$-\mu_1(Z_0 + l_1\phi_0 - h_{01})l_1 + \mu_2(Z_0 - l_2\phi_0 - h_{02})l_2 = 0$$

so that

$$\mu_1(Z_0 + l_1\phi_0 - h_{01}) = Mg\frac{l_2}{l}, \quad \text{and} \quad \mu_2(Z_0 - l_2\phi_0 - h_{02}) = Mg\frac{l_1}{l}$$

Therefore, the oscillations about their equilibrium positions are given by

$$M\ddot{Z} = -\mu_1(Z + l_1\phi - h_1) - \mu_2(Z - l_2\phi - h_2)$$

$$Mk_g^2\ddot{\phi} = -\mu_1(Z + l_1\phi - h_1)l_1 + \mu_2(Z - l_2 - h_2)l_2$$

$$m_1\ddot{h}_1 = \mu_1(Z + l_1\phi - h_1) + \left(M\frac{l_2}{l} + m_1\right)g - P_{1r}$$

$$m_2\ddot{h}_2 = \mu_2(Z - l_2\phi - h_2) + \left(M\frac{l_1}{l} + m_2\right)g - P_{2r}$$

The dynamic loadings on the bridge due to the oscillations of the vehicle are therefore

$$P_1 = \mu_1(Z + l_1\phi - h_1) - m_1\ddot{h}_1; \; P_2 = \mu_2(Z - l_2\phi - h_2) - m_2\ddot{h}_2$$

If we assume the deflection curve of the bridge from its equilibrium position of the form

$$y = A\sin\frac{\pi x}{L}\sin\omega t = \theta\sin\frac{\pi x}{L}$$

$$\frac{\partial^2 y}{\partial x^2} = -\frac{\pi^2}{L^2}A\sin\frac{\pi x}{L}\sin\omega t = \frac{\pi^4}{L^4}\theta\sin\frac{\pi x}{L}$$

$$\frac{\partial^4 y}{\partial x^4} = \frac{\pi^4}{L^4}A\sin\frac{\pi x}{L}\sin\omega t = \frac{\pi^4}{L^4}\theta\sin\frac{\pi x}{L}$$

also

$$\dot{y} = \dot{\theta}\sin\frac{\pi x}{L} \quad \text{and} \quad \ddot{y} = \ddot{\theta}\sin\frac{\pi x}{L}$$

The kinetic energy of the bridge is

$$T = \frac{1}{2}\int_0^L m\dot{y}dx = \frac{1}{2}\int_0^L m\dot{\theta}^2\sin^2\frac{\pi x}{L}dx = \frac{mL}{4}\dot{\theta}^2$$

and the potential energy is

$$V = \frac{EI}{2}\int_0^L\left(\frac{\partial^2 y}{\partial x^2}\right)^2 dx = \frac{EI}{2}\frac{\pi^4}{L^4}\theta^2\int_0^L\sin^2\frac{\pi x}{L}dx = \frac{EI\pi^4}{4L^3}\theta^2$$

The work of the applied forces is

$$P_1\delta h_1 + P_2\delta h_2 = P_1\delta y_1 + P_2\delta y_2 = \left(P_1\sin\frac{\pi a_1}{L} + P_2\sin\frac{\pi a_2}{L}\right)\delta\theta$$

Assuming θ as a generalized coordinate according to Rayleigh and Timoshenko, we obtain, from the Lagrangian equations,

$$\frac{mL\ddot{\theta}}{2} + \frac{EI\pi^4\theta}{2L^3} = P_1\sin\frac{\pi a_1}{L} + P_2\sin\frac{\pi a_2}{L}$$

Multiplying through by $\frac{2}{L}\sin\frac{\pi x}{L}$ we obtain

$$m\frac{\partial^2 y}{\partial t^2} + EI\frac{\partial^4 y}{\partial x^4} = \frac{2}{L}\left(P_1\sin\frac{\pi a_1}{L} + P_2\sin\frac{\pi a_2}{L}\right)\sin\frac{\pi x}{L}$$

which is the form of equation used by the author.

The system of equations for solution are

$$m\frac{\partial^2 y}{\partial t^2} + EI\frac{\partial^4 y}{\partial x^4} = \frac{2}{L}\left\{[\mu_1(Z + l_1\phi - h_1) - m_1\ddot{h}_1]\sin\frac{\pi a_1}{L} + [\mu_2(Z - l_2\phi - h_2) - m_2\ddot{h}_2]\sin\frac{\pi a_2}{L}\right\}\sin\frac{\pi x}{L} \quad [18]$$

$$M\ddot{Z} = -\mu_1(Z + l_1\phi - h_1) - \mu_2(Z - l_2\phi - h_2) \quad [19]$$

$$Mk_g^2\ddot{\phi} = -\mu_1(Z + l_1\phi - h_1)l_1 + \mu_2(Z - l_2\phi - h_2)l_2 \quad [20]$$

To solve, we let

$$y = A\sin\frac{\pi x}{L}\sin\omega t, \quad Z = B\sin\omega t, \quad = C\sin\omega t$$

$$h_1 = A \sin \frac{\pi a_1}{L} \sin \omega t, \quad h_2 = A \sin \frac{\pi a_2}{L} \sin \omega t$$

hence

$$\left\{\left[m + \frac{2}{L}\left(m_1 \sin^2 \frac{\pi a_1}{L} + m_2 \sin^2 \frac{\pi a_2}{L}\right)\right]\omega^2 - \left[EI\frac{\pi^4}{L^4} + \frac{2}{L}\left(\mu_1 \sin^2 \frac{\pi a_1}{L} + u_2 \sin^2 \frac{\pi a_2}{L}\right)\right]\right\} A + \frac{2}{L}\left(\mu_1 \sin \frac{\pi a_1}{L} + \mu_2 \sin \frac{\pi a_2}{L}\right) B + \frac{2}{L}\left(\mu_1 l_1 \sin \frac{\pi a_1}{L} - \mu_2 l_2 \sin \frac{\pi a_2}{L}\right) C = 0$$

$$\left(\mu_1 \sin \frac{\pi a_1}{L} + \mu_2 \sin \frac{\pi a_2}{L}\right) A + (M\omega^2 - \mu_1 - \mu_2) B + (\mu_2 l_2 - \mu_1 l_1) C = 0$$

$$\left(\mu_1 l_1 \sin \frac{\pi a_1}{L} - \mu_2 l_2 \sin \frac{\pi a_2}{L}\right) A + (\mu_2 l_2 - \mu_1 l_1) B - (\mu_1 l_1 + \mu_2 l_2 - M k_\phi^2 \omega^2) C = 0$$

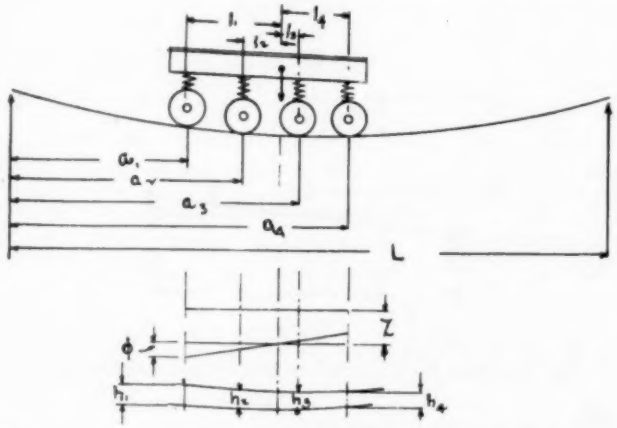


FIG. 10 SEVERAL AXLE LOADINGS, WITH INDIVIDUAL SPRING SUPPORTS PER AXLE

The system has three degrees of freedom, with three possible natural frequencies.

To obtain these frequencies, we equate the determinants of the coefficients of A, B, and C in the above equations to zero.

II—MULTIPLE-AXLE SPRING SUSPENSION

In the more complicated case with distributed axles and individual spring supports per axle supporting the locomotive chassis, we may proceed in a similar manner. The degrees of freedom and natural frequencies, however, remain the same. The space-phase relations, multiplied by the corresponding spring constants and multiplied by spring constants with distances from the locomotive chassis, can all be added vectorially. The inertia components of the dead-axle loads involve the sum of the squares of the space-phase relations multiplied by the masses of the dead-axle loads.

In the case of several axle loadings, with individual spring supports per axle, Fig. 10, let Δ_1, Δ_2, Δ_3, and Δ_4 be the static compression of the springs, so that $\mu_1\Delta_1$, $\mu_2\Delta_2$, etc., are the equilibrium values of the spring loads. Therefore we have

$$\mu_1\Delta_1 + \mu_2\Delta_2 + \mu_3\Delta_3 + \mu_4\Delta_4 = Mg$$

$$\mu_1 l_1 \Delta_1 + \mu_2 l_2 \Delta_2 - \mu_3 l_3 \Delta_3 - \mu_4 l_4 \Delta_4 = 0$$

We may regard the static compressions of the form

$$\Delta_1 = Z_0 + l_1\phi_0, \text{ etc.}; \quad \Delta_3 = Z_0 - l_3\phi_0, \text{ etc.}$$

Then

$$\mu_1\Delta_1 = \mu_1 Mg\left(\frac{C - l_1 B}{AC - B^2}\right), \text{ etc.}$$

$$\mu_3\Delta_3 = \mu_3\left(\frac{C + l_3 B}{AC - B^2}\right), \text{ etc.}$$

where

$$A = \mu_1 + \mu_2 + \mu_3 + \mu_4, \quad B = \mu_1 l_1 + \mu_2 l_2 - \mu_3 l_3 - \mu_4 l_4$$
$$C = \mu_1 l_1^2 + \mu_2 l_2^2 + \mu_3 l_3^2 + \mu_4 l_4^2$$

Assume, now, z, ϕ, h_1, h_2, etc., as the displacements from the equilibrium position. The potential energy of the spring suspension is

$$V = \frac{1}{2}\Big[\mu_1(z + l_1\phi - h_1)^2 + \mu_2(z + l_2\phi - h_2)^2 + \mu_3(z - l_3\phi - h_3)^2 + \mu_4(z - l_4\phi - h_4)^2\Big] + \mu_1(z + l_1\phi - h_1)\Delta_1 + \mu_2(z + l_2\phi - h_2)\Delta_2 + \mu_3(z - l_3\phi - h_3)\Delta_3 + \mu_4(z - l_4\phi - h_4)\Delta_4$$

The total kinetic energy of the vehicle referred to the equilibrium position is

$$T = \frac{1}{2}\Big[M(\dot{z}^2 + h_3^2\dot{\phi}^2) + m_1\dot{h}_1^2 + m_2\dot{h}_2^2 + m_3\dot{h}_3^2 + m_4\dot{h}_4^2\Big]$$

The work of the applied forces on the total vehicle is

$$Mg\delta z + (m_1 g - P_1')\delta h_1 + (m_2 g - P_2')\delta h_2 + (m_3 g - P_3')\delta h_3 + (m_4 g - P_4')\delta h_4$$

where the P's, i.e., the wheel reactions on the bridge, have the equilibrium value P_0 and the dynamic component P.

That is,

$$P_1' = (\mu_1\Delta_1 + m_1 g) + P_1, \quad P_2' = (\mu_2\Delta_2 + m_2 g) + P_2, \text{ etc.}$$

The equations of motion of the vehicle reduce to

$$M\ddot{z} = -\mu_1(z + l_1\phi - h_1) - \mu_2(z + l_2\phi - h_2) - \mu_3(z - l_3\phi - h_3) - \mu_4(z - l_4\phi - h_4) \quad [21]$$

$$Mk_\phi^2\ddot{\phi} = -\mu_1(z + l_1\phi - h_1)l_1 - \mu_2(z + l_2\phi - h_2)l_2 + \mu_3(z - l_3\phi - h_3)l_3 + \mu_4(z - l_4\phi - h_4)l_4 \quad [22]$$

$$m_1\ddot{h}_1 = \mu_1(z + l_1\phi - h_1) - P_1 \quad [23]$$

$$m_2\ddot{h}_2 = \mu_2(z + l_2\phi - h_2) - P_2 \quad [24]$$

$$m_3\ddot{h}_3 = \mu_3(z - l_3\phi - h_3) - P_3 \quad [25]$$

$$m_4\ddot{h}_4 = \mu_4(z - l_4\phi - h_4) - P_4 \quad [26]$$

Now considering the bridge proper, the potential energy is

$$V = \frac{1}{2}\,EI\int_0^L \left(\frac{\partial^2 y}{\partial x^2}\right)^2 dx + EI\int_0^L \frac{\partial^2 y_0}{\partial x^2}\frac{\partial^2 y}{\partial x^2}\,dx$$

and since

$$(\mu_1\Delta_1 + m_1 g)h_1 + (\mu_2\Delta_2 + m_2 g)h_2 \ldots = EI\int_0^L \frac{\partial^2 y_0}{\partial x^2}\frac{\partial^2 y}{\partial x^2}\,dx$$

where $EI\dfrac{\partial^2 y_0}{\partial x^2}$ is the static bending moment, the equation for the

vibration of the bridge becomes, as before,

$$m\frac{\partial^2 y}{\partial t^2} + EI\frac{\partial^4 y}{\partial x^4} = \frac{2}{L}\left(P_1 \sin\frac{\pi a_1}{L} \ldots P_4 \sin\frac{\pi a_4}{L}\right)\sin\frac{\pi x}{L} \quad [27]$$

Substituting values of P_1, P_2, etc., from Equations [23] to [26] in Equation [27], and letting $y = A \sin\frac{\pi x}{L}\sin\omega t$,

$$h_1 = A \sin\frac{\pi a_1}{L}\sin\omega t$$

$$h_2 = A \sin\frac{\pi a_2}{L}\sin\omega t;\ h_3 = A \sin\frac{\pi a_3}{L}\sin\omega t;\ h_4 = A \sin\frac{\pi a_4}{L}\sin\omega t$$

$$z = B \sin\omega t;\ \phi = C \sin\omega t$$

We have, on substituting in Equations [21], [22], and [27], the three equations in terms of the amplitudes A, B, and C. Equating the determinant of their coefficients to zero, we obtain the frequencies.

III—Equilateral Spring Suspension

However, in the majority of locomotives, the spring suspension is equalized in an equivalent front and rear system, Fig. 11. If l_r and l_f are the distances from the center of gravity of the equilization to the center of gravity of the chassis, we may roughly approximate the situation by expressing the potential energy of the spring system by

$$V = \frac{1}{2}\mu_r(Z + l_r\phi - h_r)^2 + \frac{1}{2}\mu_f(Z - l_f - h_f)^2$$

where

$$h_r = \frac{h_1 + h_2}{2} \qquad h_f = \frac{h_3 + h_4}{2}$$

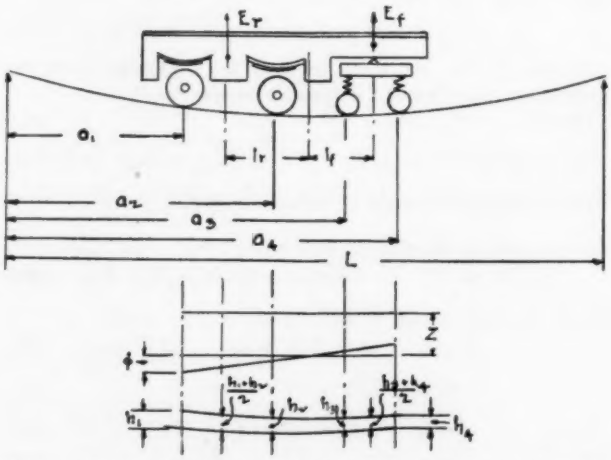


Fig. 11 Spring Suspension Equalized in an Equivalent Front and Rear System

and

$$\mu_r = \Sigma\,\mu_m + k\,\Sigma\,\mu_l = \mu_1 + \mu_2$$

$$\mu_f = \Sigma\,\mu_m + k'\,\Sigma\,\mu_l' = \mu_3 + \mu_4$$

The equations of motion then become (approximately)

$$M\ddot{z} = -\mu_r[z + l_r\phi - 0.5(h_1 + h_2)] - \mu_f[z - l_r\phi - 0.5(h_3 + h_4)]$$

$$Mk_g^2\ddot{\phi} = -\mu_r[z + l_r\phi - 0.5(h_1 + h_2)]l_r - \mu_f[z - l_r\phi - 0.5(h_3 + h_4)]l_f$$

$$m_1\ddot{h}_1 = 0.5\mu_r[z + l_r\phi - 0.5(h_1 + h_2)] - P_1, \text{ etc.}$$

Actually the analysis becomes much more complicated. Consider the rear equalization of Fig. 11, shown in detail in Fig. 12.

The angling of the equalizer beams introduces an additional degree of freedom θ. If we neglect the angular inertia of the equalizer rigging, we then can eliminate θ by the very condition of equalization.

The compressions of springs 1 and 2, are (Fig. 12)

$$\frac{1}{2}\left[(z + l_r\phi - a\theta) + (z + l_{r1}\phi)\right] - h_1$$

$$= z + 0.5(l_r + l_{r1})\phi - 0.5a\theta - h_1$$

$$\frac{1}{2}\left[(z + l_r\phi + b\theta) + (z + l_{r2}\phi)\right] - h_2$$

$$= z + 0.5(l_r + l_{r2}) + 0.5b\theta - h_2$$

But we have the condition due to equalization, i.e.,

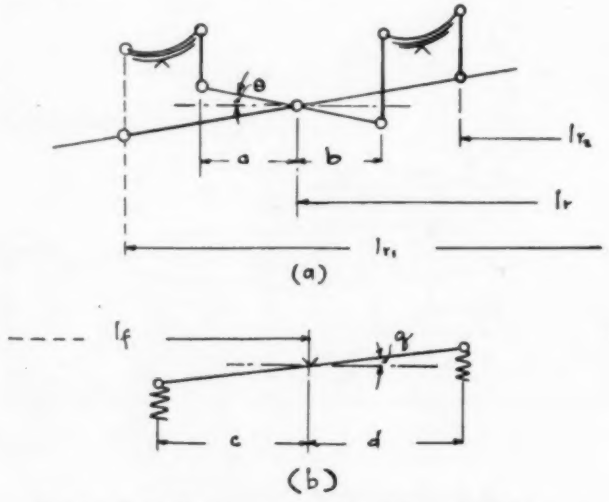


Fig. 12 Rear Equalization of Fig. 11 Shown in Detail

$$\frac{\mu_1}{2}\left[z + 0.5(l_r + l_{r1})\phi - 0.5a\theta - h_1\right]a =$$

$$\frac{\mu_2}{2}\left[z + 0.5(l_r + l_{r2})\phi + 0.5b\theta - h_2\right]b$$

so that θ can be expressed in terms of z and ϕ.

Likewise the compressions of springs 3 and 4 for the front equalization are

$$(z - l_f\phi + cq) - h_3 \quad \text{and} \quad (z - l_f\phi - dq) - h_4$$

with the condition,

$$\mu_3(z - l_f\phi + cq - h_3) = \mu_4(z - l_f\phi - dq - h_4)$$

from which q can be expressed in terms of z and ϕ.

The potential energy has the form,

$$V = \frac{\mu_1}{2}(\alpha_1 z + \beta_1\phi - \gamma_1 h_1 - \xi_1 h_2)^2 + \frac{\mu_2}{2}(\alpha_2 z + \beta_2\phi - \gamma_2 h_2 - \xi_2 h_1)^2$$

$$+ \frac{\mu_3}{2}(\alpha_3 z - \beta_3\phi - \gamma_3 h_3 - \xi_3 h_4)^2 + \frac{\mu_4}{2}(\alpha_4 z - \beta_4\phi - \gamma_4 h_4 - \xi_4 h_3)^2$$

where

$$\alpha_1 = 1 - \frac{\mu_1 a^2 - \mu_2 ab}{\mu_1 a^2 + \mu_2 b^2} \qquad \beta_1 = \frac{l_r + l_{r1}}{2} - \frac{\mu_1 a^2(l_r + l_{r1}) - \mu_2 ab(l_r + l_{r2})}{2(\mu_1 a^2 + \mu_2 b^2)}$$

$$\gamma_1 = 1 - \frac{\mu_1 a^2}{\mu_1 a^2 + \mu_2 b^2}, \qquad \xi_1 = \frac{\mu_2 ab}{\mu_1 a^2 + \mu_2 b^2}$$

$$\alpha_2 = 1 + \frac{\mu_1 ab - \mu_2 b^2}{\mu_1 a^2 + \mu_2 b^2}, \qquad \beta_2 = \frac{l_r + l_{r2}}{2} + \frac{\mu_1 ab(l_r + l_{r1}) - \mu_2 b^2 (l_r + l_{r2})}{2(\mu_1 a^2 + \mu_2 b^2)}$$

$$\gamma_2 = 1 - \frac{\mu_2 b^2}{\mu_1 a^2 + \mu_2 b^2}, \qquad \xi_2 = \frac{\mu_1 ab}{\mu_1 a^2 + \mu_2 b^2}$$

$$\alpha_3 = 1 + \frac{\mu_4 cd - \mu_3 c^2}{\mu_3 c^2 + \mu_4 d^2}, \qquad \beta_3 = l_f - \frac{\mu_3 l_f c^2 - \mu_4 l_f cd}{\mu_3 c^2 + \mu_4 d^2}$$

$$\gamma_3 = 1 - \frac{\mu_3 c^2}{\mu_3 c^2 + \mu_4 d^2}, \qquad \xi_3 = \frac{\mu_4 cd}{\mu_3 c^2 + \mu_4 d^2}$$

$$\alpha_4 = 1 - \frac{\mu_4 d^2 - \mu_3 cd}{\mu_3 c^2 + \mu_4 d^2}, \qquad \beta_4 = l_f + \frac{\mu_3 l_f cd - \mu_4 l_f d^2}{\mu_3 c^2 + \mu_4 d^2}$$

$$\gamma_4 = 1 - \frac{\mu_4 d^2}{\mu_3 c^2 + \mu_4 d^2}, \qquad \xi_4 = \frac{\mu_3 cd}{\mu_3 c^2 + \mu_4 d^2}$$

The kinetic energy from the equilibrium position is,

$$T = {}^1/_2[M\dot{z}^2 + Mk_\theta^2\dot{\phi}^2 + m_1\dot{h}_1^2 + m_2\dot{h}_2^2 + m_3\dot{h}_3^2 + m_4\dot{h}_4^2]$$

and the work of the reaction of the bridge is

$$-(P_1\delta h_1 + P_2\delta h_2 + P_3\delta h_2 + P_4\delta h_4)$$

The equations of motion reduce to,

$$M\ddot{z} = -\mu_1(\alpha_1 z + \beta_1\phi - \gamma_1 h_1 - \xi_1 h_2)\alpha_1 - \mu_2(\alpha_2 z + \beta_2\phi - \gamma_2 h_2 - \xi_2 h_1)\alpha_2 - \mu_3(\alpha_3 z - \beta_3\phi - \gamma_3 h_3 - \xi_3 h_4)\alpha_3 - \mu_4(\alpha_4 z - \beta_4\phi - \gamma_4 h_4 - \xi_4 h_3)\alpha_4 \quad [28]$$

$$Mk_\theta^2\ddot{\phi} = -\mu_1(\alpha_1 z + \beta_1\phi - \gamma_1 h_1 - \xi_1 h_2)\beta_1 - \mu_2(\alpha_2 z + \beta_2\phi - \gamma_2 h_2 - \xi_2 h_1)\beta_2 + \mu_3(\alpha_3 z - \beta_3\phi - \gamma_3 h_3 - \xi_3 h_4)\beta_3 + \mu_4(\alpha_4 z - \beta_4\phi - \mu_4 h_4 - \xi_4 h_3)\beta_4 \quad [29]$$

and

$$m\frac{\partial^2 y}{\partial t^2} + EI\frac{\partial^4 y}{\partial x^4} = \frac{2}{L}\left(P_1 \sin\frac{\pi a_1}{L} + P_2 \sin\frac{\pi a_2}{L} + P_3 \sin\frac{\pi a_3}{L} + P_4 \sin\frac{\pi a_4}{L}\right)\sin\frac{\pi x}{L} \quad [30]$$

where

$$P_1 = \mu_1(\alpha_1 z + \beta_1\phi - \gamma_1 h_1 - \xi_1 h_2)\gamma_1 + \mu_2(\alpha_2 z + \beta_2\phi - \gamma_2 h_2 - \xi_2 h_1)\,\xi_2 - m_1\ddot{h}_1$$

$$P_2 = \mu_1(\alpha_1 z + \beta_1\phi - \gamma_1 h_1 - \xi_1 h_2)\xi_1 + \mu_2(\alpha_2 z + \beta_2\phi - \gamma_2 h_2 - \xi_2 h_1)\,\gamma_2 - m_2\ddot{h}_2$$

$$P_3 = \mu_3(\alpha_3 z - \beta_3\phi - \gamma_3 h_3 - \xi_3 h_4)\gamma_3 + \mu_4(\alpha_4 z - \beta_4\phi - \gamma_4 h_4 - \xi_4 h_3)\,\xi_4 - m_3\ddot{h}_3$$

$$P_4 = \mu_3(\alpha_3 z - \beta_3\phi - \gamma_3 h_3 - \xi_3 h_4)\xi_3 + \mu_4(\alpha_4 z - \beta_4\phi - \gamma_4 h_4 - \xi_4 h_3)\,\gamma_4 - m_4\ddot{h}_4$$

Alternately, we note

$$\theta = f(z,\phi,h_1,h_2) \text{ and } q = f(z,\phi,h_3,h_4)$$

The potential energy has the form

$$V = \frac{1}{2}\left[\mu_1\left(z + \frac{l_r + l_{r1}}{2}\phi - \frac{a\theta}{2} - h_1\right)^2 + \mu_2\left(z + \frac{l_r + l_{r2}}{2}\phi + \frac{b\theta}{2} - h_2\right)^2 + \mu_3(z - l_f\phi + cq - h_3)^2 + \mu_4(z - l_f\phi - dq - h_4)^2\right]$$

$$-\frac{\partial V}{\partial h_1} = \mu_1\left(z + \frac{l_r + l_{r1}}{2}\phi - \frac{a\theta}{2} - h_1\right)\left(1 + \frac{a}{2}\frac{\partial\theta}{\partial h_1}\right) - \mu_2\left(z + \frac{l_r + l_{r2}}{2}\phi + \frac{b\theta}{2} - h_2\right)\frac{b}{2}\frac{\partial\theta}{\partial h_1}$$

$$-\frac{\partial V}{\partial h_3} = \mu_3(z - l_f\phi + cq - h_3)\left(1 - c\frac{\partial q}{\partial h_3}\right) + \mu_4(z - l_f\phi - dq - h_4)d\frac{\partial q}{\partial h_3}$$

But since

$$\mu_1\left(z + \frac{l_r + l_{r1}}{2}\phi - \frac{a\theta}{2} - h_1\right)a = \mu_2\left(z + \frac{l_r + l_{r2}}{2}\phi + \frac{b\theta}{2} - h_2\right)b \text{ and}$$

$$\mu_3(z - l_f\phi + cq - h_3)\,c = \mu_4(z - l_f\phi - dq - h_4)d$$

we have

$$-\frac{\partial V}{\partial h_1} = \mu_1\left(z + \frac{l_r + l_{r1}}{2}\phi - \frac{a\theta}{2} - h_1\right), \text{ and}$$

$$-\frac{\partial V}{\partial h_3} = \mu_3(z - l_f\phi + cq - h_3)$$

Proceeding in a similar manner for the other coordinates and then substituting for θ and q, we obtain the equations of motion in the following more simplified form:

$$M\ddot{z} = -\mu_1(\alpha_1 z + \beta_1\phi - \gamma_1 h_1 - \xi_1 h_2) - \mu_2(\alpha_2 z + \beta_2\phi - \gamma_2 h_2 - \xi_2 h_1) - \mu_3(\alpha_3 z - \beta_3\phi - \gamma_3 h_3 - \xi_3 h_4) - \mu_4(\alpha_4 z - \beta_4\phi - \gamma_4 h_4 - \xi_4 h_3) \quad [31]$$

$$Mk_\theta^2\ddot{\phi} = -\mu_1(\alpha_1 z + \beta_1\phi - \gamma_1 h_1 - \xi_1 h_2)\frac{l_r + l_{r1}}{2} - \mu_2(\alpha_2 z + \beta_2\phi - \gamma_2 h_2 - \xi_2 h_1)\frac{l_r + l_{r2}}{2} + \mu_3(\alpha_3 z - \beta_3\phi - \gamma_3 h_3 - \xi_3 h_4)l_f + \mu_4(\alpha_4 z - \beta_4\phi - \gamma_4 h_4 - \xi_4 h_3)l_f \quad [32]$$

and

$$m\frac{\partial^2 y}{\partial t^2} + EI\frac{\partial^4 y}{\partial x^4} = \frac{2}{L}\left(P_1 \sin\frac{\pi a_1}{L} \ldots P_4 \sin\frac{\pi a_4}{L}\right)\sin\frac{\pi x}{L} \quad [33]$$

where now

$$P_1 = \mu_1(\alpha_1 z + \beta_1\phi - \gamma_1 h_1 - \xi_1 h_2) - m_1\ddot{h}_1$$

$$P_2 = \mu_2(\alpha_2 z + \beta_2\phi - \gamma_2 h_2 - \xi_2 h_1) - m_2\ddot{h}_2$$

$$P_3 = \mu_3(\alpha_3 z - \beta_3\phi - \gamma_3 h_3 - \xi_3 h_4) - m_3\ddot{h}_3$$

$$P_4 = \mu_4(\alpha_4 z - \beta_4\phi - \gamma_4 h_4 - \xi_4 h_3) - m_4\ddot{h}_4$$

$$y = A\sin\frac{\pi x}{L}\sin\omega t, \qquad z = B\sin\omega t, \qquad \phi = C\sin\omega t$$

$$h_1 = A\sin\frac{\pi a_1}{L}\sin\omega t, \qquad h_2 = A\sin\frac{\pi a_2}{L}\sin\omega t$$

$$h_3 = A\sin\frac{\pi a_3}{L}\sin\omega t, \qquad h_4 = A\sin\frac{\pi a_4}{L}\sin\omega t$$

On substituting we obtain three equations in terms of the amplitudes A, B, and C. Equating the determinant of their coefficients to zero, we obtain the frequency equation.

In the actual case of an extended equalization the problem is more complicated, but the system still reduces to one of three degrees of freedom.

The friction of a spring equalization is usually considerable, and from a practical aspect, due to this large damping, ordinarily the secondary oscillation of the spring system of the locomotive can often be neglected. If, however, the spring system is considered, the two additional degrees of freedom should be considered.

On the whole the author must be congratulated for an interesting method of attack on a very complicated subject. The object of the discussion is possibly an additional feature, rather than any criticism on this very ably presented paper.

Author's Closure

It is encouraging to find that this necessarily brief outline of what is now an extensive subject should stimulate so much valuable discussion. It was not the author's intention to include anything which has not yet been substantiated by direct measurement, and it was for this reason that he omitted anything more than a mention of the torsional and lateral vibrations to which Dr. Timoshenko refers, important though they undoubtedly are. The author is not aware of any practical investigations of such vibrations, and without experimental guidance theory would be apt to become too speculative.

Professor Inglis' analysis of the very intractable problem of determining the vibrations of a heavy uniform beam carrying a moving mass together with a pulsating force, is such, that although it has yielded some results of striking accuracy, it cannot be conveniently summarized. The author would refer those interested in this matter to paper No. 4 of the Bibliography.

The author's thanks are due to Mr. Erksergian for extending the rudimentary calculation for the sprung locomotive to more realistic cases. It seems very desirable that the effects of friction in the spring mountings should be considered, but the author has not found any measurements of this quantity for American locomotives or cars. In the locomotives used by the Bridge Stress Committee, it was of the order of eight tons.

The question Mr. Beggs raises does not admit of a simple answer. The effect of the cable makes the vibrations of a suspension bridge more complicated than those of a simple girder with which we have been concerned in the paper itself.

The modes of vibration fall into two classes. In the first we have an antinode at the middle as in the fundamental and "odd" modes of a simple beam. The cable is stretched and its tension consequently increased in the vibration, and this prevents the motion from being simple harmonic. Only when this increase in tension is negligible in comparison with the equilibrium tension do we have an approach to simple harmonic motion.

In the modes of the second class there is a node at the middle. In the first of these modes, for example, consisting of two loops, one half of the cable is moving upward and contracting while the other is moving downward and extending. The total stretch is zero to the first order in the small quantity deflection divided by span. There is consequently no appreciable alteration of cable tension and a simple harmonic vibration is possible. A closely approximate investigation of this class is easily made. The author finds that the frequency for the rth mode, r being the (even) number of loops in the deflection curve, is given by

$$n^2 = \frac{\pi^2 EI\frac{r^4}{l^4} + H\frac{r^2}{l^2}}{4(m_G + m_C)} \qquad [34]$$

n being the frequency, EI referring to the girder, l the span, H the horizontal component of the (constant) cable tension, m_G the mass per unit length of the girder, and m_C the mass per unit length of the cable.

This formula will hold also, as an approximation, for the other class of vibrations, the extensional modes, for which r is odd, provided the amplitude is small, since we may then neglect the variation in tension.

It follows that the ratio of frequencies of the rth mode of two bridges characterized by subscripts 1 and 2 is

$$\frac{n_1^2}{n_2^2} = \frac{E_1 I_1\frac{r^4}{l_1^4} + H_1\frac{r^2}{l_1^2}}{E_2 I_2\frac{r^4}{l_2^4} + H_2\frac{r^2}{l_2^2}}\,\frac{m_{G2} + m_{C2}}{m_{G1} + m_{C1}} \qquad [35]$$

We have now to consider the relative magnitudes of the quantities involved for two geometrically similar bridges, the linear dimensions of one being l_1/l_2 times those of the other, and the stresses being the same.

Taking the curve of the cable as a parabola, its equation is expressible in the form

$$y = \frac{x^2}{4a}$$

where a is a linear dimension. The load per unit length sustained in this configuration is $H\frac{d^2y}{dx^2}$, that is, $\frac{H}{2a}$. The total load per unit length applied to the girder is then

$$m_G g - \frac{H}{2a}$$

producing a maximum bending stress

$$\frac{1}{8}\left(m_G g - \frac{H}{2a}\right) l^2 \frac{d}{EI}$$

where d is half the depth of the girder. If this stress is to be the same in both bridges we must have

$$\left(m_{G1} g + \frac{H_1}{2a_1}\right)\frac{l_1^2 d_1}{I_1} = \left(m_{G2} g + \frac{H_2}{2a_2}\right)\frac{l_2^2 d_2}{I_2} \qquad [36]$$

the material being the same in each girder. On account of the geometrical similarity,

$$\frac{a_1}{a_2} = \frac{l_1}{l_2};\quad \frac{I_1}{I_2} = \frac{l_1^4}{l_2^4};\quad \frac{d_1}{d_2} = \frac{l_1}{l_2};\quad \frac{m_{C1}}{m_{C2}} = \frac{l_1^2}{l_2^2} \qquad [37]$$

so that we require, for [36]

$$\frac{m_{G1} g + \frac{H_1}{2a_1}}{m_{G2} g + \frac{H_2}{2a_2}} = \frac{l_1}{l_2} \qquad [38]$$

Since the stress is the same in both cables, the total tensions are in the ratio of the squares of the linear dimensions, i.e.,

$$\frac{H_1}{H_2} = \frac{l_1^2}{l_2^2} \quad \text{and therefore} \quad \frac{H_1}{2a_1}\Big/\frac{H_2}{2a_2} = \frac{l_1}{l_2}$$

In view of this it is sufficient, to satisfy [38], to make

$$\frac{m_{G1}}{m_{G2}} = \frac{l_1}{l_2}$$

that is, the loadings per unit length of the girders are in the ratio of the linear dimensions.

Since

$$\frac{I_1}{l_1^4} = \frac{I_2}{l_2^4}; \quad \frac{H_1}{l_1^2} = \frac{H_2}{l_2^2}$$

we have from [35]

$$\frac{n_1^2}{n_2^2} = \frac{m_{G2} + m_{C2}}{m_{G1} + m_{C1}}$$

and since

$$\frac{m_{C1}}{m_{C2}} = \frac{l_1^2}{l_2^2}; \quad \frac{m_{G1}}{m_{G2}} = \frac{l_1}{l_2}$$

this is not a function of l_1/l_2 only. It can be written as

$$\frac{n_1^2}{n_2^2} = \frac{l_2}{l_1}\left\{1 + \left(\frac{l_1}{l_2} - 1\right)\frac{m_{C1}}{m_{G1} + m_{C1}}\right\}$$

and is seen to depend on the ratio of cable mass to girder mass. The latter term signifies, of course, all the mass which moves with the girder, excluding tie rods and cable.

The whole subject of suspension-bridge vibrations, however, calls for a more exacting investigation than can be included in this brief discussion, and the author hopes to present further considerations in a future paper.

Equation [9] of the paper, to which Dr. Timoshenko has called attention, is incorrectly copied. The form given in the Appendix is correct. The "m-symbols" may be taken as either masses or weights since they occur only in ratios. The quadratic at the top of page 15, first column, should be

$$1.2n^4 - 51.3n^2 + 324 = 0$$

and the two frequencies obtained from this are 3.28 and 5.66.

APM-53-3

Plastic Torsion

An Experimental Determination of the Stress Distribution in a Bar Which Has Been Twisted to the Limit of Plasticity

By A. NADAI,[1] EAST PITTSBURGH, PA.

Among the locally varying distributions of stress with partial yielding, that of a twisted bar is treated, for which the transition from a state of equilibrium with purely elastic deformations to one with plastic deformations was investigated. An analogy described on an earlier occasion was utilized to determine by an experimental method the distribution of stress in the twisted bar. This distribution of stress was compared with the position of the thin layers of slip observed in the etched cross-sections of twisted steel bars.



IN RECENT years it has repeatedly been noted by engineers and mathematicians that the same means by which cases of the equilibrium of the internal forces in continuously distributed elastic masses have been treated in the past, might to a certain extent also be applied with success in the analysis of cases concerning the beginning of plastic flow in solids. It was possible by methods similar to those used in the theory of elasticity to treat cases in which the internal forces or stresses in certain portions of material bodies reach the limits at which permanent deformations of considerable amount start to develop, while in other portions of these bodies the deformations still remain small and purely elastic. Although, due to the beginning of plastic flow, the internal forces in such cases cannot be considered as being in a state of true equilibrium, the disturbances in this latter in the case of slow changes of shape may be regarded as slight and negligible.

Among the locally varying distributions of stress with partial yielding, one in which the equilibrium of the internal forces might be looked upon as only slightly disturbed and in which it was possible to follow more precisely the transition from a state of equilibrium with purely elastic deformations into one with gradually increasing permanent deformations, is that of a twisted cylindrical or prismatical bar of constant cross-section subjected to a slowly increasing torque acting about its axis. Since the methods used to determine the distribution of stresses in a bar twisted above the yield point utilize more visual and experimental rather than analytical means and are analogous to those used in the elastic-torsion problem, they may be of interest to engineers.

There are, however, reasons to believe that from a description of the transition from the elastic into the plastic state in a special case such as that of a twisted bar, valuable information regarding the yielding process in ductile metals under more complicated states of stress may be obtained.

A peculiar and, from the point of view of mechanics, very interesting phenomenon accompanying the transition from the elastic to the plastic state in mild steel—or, more precisely, in a metal or substance having a sharply defined elastic and well-defined plastic range—is the formation of thin plastic layers in which the metal apparently has been deformed more than in the immediate neighborhood thereof. The intersections of these thin layers with the surface of a structural member or a test piece frequently form regular lines known to engineers as "Lüders lines" or as "slip lines." These thin layers, where they intersect the surface of well-polished test specimens, appear as fine, dull strips, often of a very regular shape, and may easily be observed. They can also be made easily visible by etching in the cross-sections. These thin layers of plastically deformed metal, or inner *surfaces of slip*, have a close connection with the fields of stress under which they were produced. They may therefore serve to check the assumptions on which the computation of the stresses in the plastic state were based. They also make possible the verification not only of the intensity of the stress, but of the geometrical distribution of the directions of principal stress in the plastic state for cases for which these stresses and their directions were predicted. In other cases, where this is not possible, they may serve to determine the position of the principal-stress directions (of the trajectories of principal stress). It was found, further, that the process of fracture is related in certain cases to the presence of these thin layers, in which the iron apparently is more severely distorted and the grain structure weakened than in the surrounding portions of material.

As has already been pointed out,[2] among the ordinary methods for testing the mechanical properties of the ductile metals, the torsion test of a cylindrical bar with a circular cross-section has an advantage over the ordinary tensile test in that it allows the determination of the maximum breaking stress. From a torsion test with a round bar it is possible to determine the true stress which will cause fracture of the material in pure shear; this, however, cannot be said of an ordinary tensile test, because the actual distribution of the stresses across the surface of rupture in the necked portion of the tensile-test bar at the moment of rupture is not known. To determine the breaking stress by means of a torsion test, it is necessary to observe the torque M as it changes with the angular twist θ per unit of length of the twisted cylinder, and to compute from the observed moment-twist diagram or curve $M = F(\theta)$ the stress-strain diagram in pure shear. This leads to a yield stress s, which in low-temperature tests gradually increases with the unit shearing defor-

[1] Research Laboratories of the Westinghouse Elec. & Mfg. Co. Dr. Nadai was for three years assistant to Prof. Dr. Eugene Meyer, Director of Laboratory of Testing of Materials and Professor of Applied Mechanics at the Technical University (Technische Hochschule), Berlin. For ten years he worked at the Institute of Applied Mechanics of the University of Göttingen in Germany with Prof. Dr. L. Prandtl. He was extraordinary professor of applied mechanics at the University of Göttingen. Since March, 1929, he has been connected with the Research Laboratories of the Westinghouse Company and with the Graduate School of the University of Pittsburgh.

Contributed by the Applied Mechanics Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] Trans. A.S.M.E., Applied Mechanics Division, 1930.

mation to the breaking point, thus showing what is known as "work hardening" of the metal.

So far it has not been possible to compute the distribution of the stresses in twisted bars of other than circular cross-section if the bar is stressed above the plastic limit and the effect of work hardening is considered. There exists, however, a case in which it is possible to determine the distribution of stresses in twisted bars of other than circular cross-section after the yield point has been reached. This is the case for mild steel or metals having a torsion diagram similar to that shown in Fig. 1.

Pure Shear in Mild Steel

If a torsion test is made with a round bar of mild steel, the initial portion of the moment-twist diagram [$M = F(\theta)$] which is recorded by means of exact torsion instruments has a shape like that in Fig. 1. The torque M first increases linearly with the angle of twist θ to a point A. The line then gradually bends in to a curve AB, after which the moment M remains for a while (BC) nearly constant ($M = M_1$), and then finally starts again to increase (CD). Sometimes with very mild steels (having a sharply defined yield stress in pure tension), the torque M may even drop for a while after reaching B, and then start again to increase. The following discussion will be limited to the deformations of twisted steel bars which correspond to the portion ABC (Fig. 1) of the torque-twist diagram $M = F(\theta)$ of a bar of circular cross-section. In other words, it will be assumed that the deformations are limited to such amounts as correspond to the twist θ for points such as B or C in Fig. 1. The shape of this latter diagram indicates that for mild steel the corresponding stress-strain diagram $s = f(\gamma)$ for pure shear (s is the shearing stress which produces a unit shearing deformation γ, as a first approximation and for not too large deformations may be considered as consisting of two separate straight lines, one steeply inclined and one horizontal, as shown in Fig. 2.

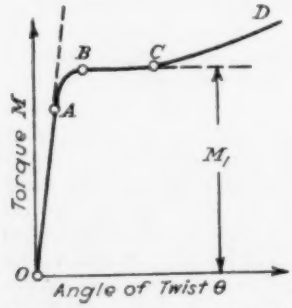


Fig. 1 Moment-Twist Diagram for Mild Steel [$M = F(\theta)$]

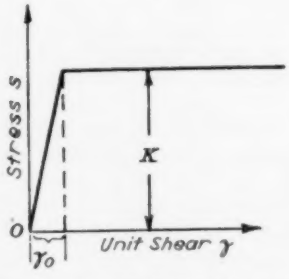


Fig. 2 Stress-Strain Diagram in Pure Shear

If $\gamma < \gamma_0$, i.e., in the elastic range the shearing stress according to Hooke's law is proportional to the unit shear,

$$s = G\gamma \quad [1]$$

where G is the modulus of rigidity of the bar; and if $\gamma > \gamma_0$, i.e., in the plastic range the shearing stress s remains constant,

$$s = k \quad [2]$$

The constant k is the yield stress of the bar in pure shear and evidently the unit shear γ_0 is equal to k/G.

Since all elements of a bar subjected to pure torsion are stressed in the same manner and the only stress in them is pure shear, it will be possible under these assumptions to predict to a certain extent the distribution of stresses in a twisted bar having an arbitrary shape of cross-section as the stresses gradually tend to approach the yield stress k in pure shear. It can therefore be assumed that where the resultant shearing stress s in a bar is smaller than the yield stress k in pure shear ($s < k$), stress and strain are proportional according to Equation [1], and the equations hold, which are valid for elastic torsion, and that where yielding occurs in the bar, the resultant shearing stress $s = k$ remains constant regardless of further increases of the shearing deformations. It will furthermore be assumed that the three components of the displacement of an arbitrary point within the bar with respect to certain fixed and immovable axes remain small quantities compared to the dimensions of the cross-section. If the permanent unit shears do not exceed more than perhaps 20 to 40 times the value of $\gamma_0 = k/G$, it will be permissible to use the simplifications which have been introduced above.

Elastic Torsion

The Elastic-Stress Function for Torsion. Let us first refer briefly to the case of the elastic torsion of a bar with a constant cross-section as treated by St. Venant. Let the z-axis coincide with the (unknown) axis of twist of the bar and the x- and y-axes be chosen in the plane of one of the cross-sections of the bar. Also let

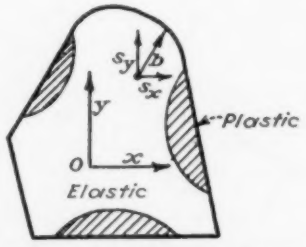


Fig. 3 Elastic and Plastic Areas in Cross-Section of a Twisted Bar

θ = angle of twist per unit of length

G = modulus of rigidity of the bar

s_x, s_y = components of shearing stress acting at the point x, y, z, in the plane z = const. and parallel, respectively, to the x- and y-axes.

$s = \sqrt{s_x^2 + s_y^2}$ = resultant shearing stress acting in the same plane and at the same point.

The elastic-torsion problem of a bar with a cross-sectional area A with simply connected boundary lines requires that the two components of the shearing stress s_x and s_y satisfy the equations:

$$\frac{\partial s_x}{\partial x} + \frac{\partial s_y}{\partial y} = 0 \quad [3]$$

$$\frac{\partial s_x}{\partial y} - \frac{\partial s_y}{\partial x} = -2G\theta = \text{const.} \quad [4]$$

with the condition at the boundary:

$$\frac{s_y}{s_x} = \frac{dy}{dx} \quad [5]$$

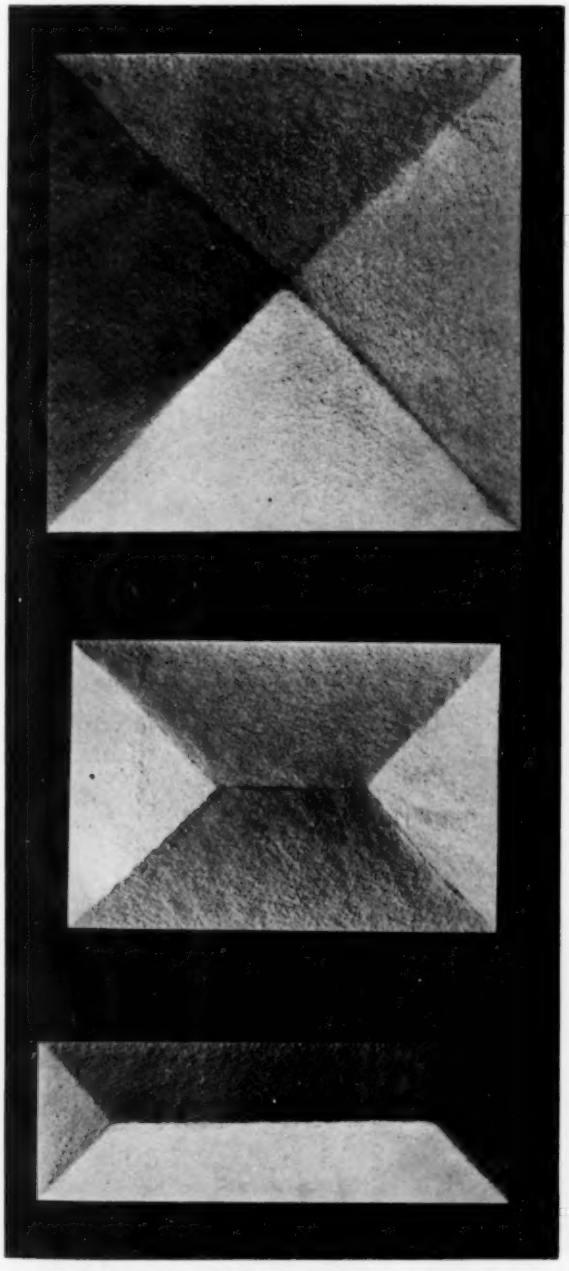
where $y = f(x)$ designates the equation of the boundary curve of the cross-sectional area A. The condition of equilibrium of a small element of the bar is expressed by Equation [3]. Equation [4] is a consequence of Hooke's law and the subsequent rotation of cross-sections through an angle proportional to z. Equation [5] expresses the fact that the resultant shearing stress s along the boundary line $y = f(x)$ must be tangent to this line at every point.

To solve the three equations, a function $F(x, y)$ can be introduced. Equation [3] is satisfied if

$$s_x = \frac{\partial F}{\partial y}, \qquad s_y = -\frac{\partial F}{\partial x} \ldots\ldots\ldots\ldots\ldots [6]$$

and Equation [4] if the function F satisfies the equation

$$\frac{\partial^2 F}{\partial x^2} + \frac{\partial^2 F}{\partial y^2} = \Delta F = -2G\theta = \text{const.} \ldots\ldots\ldots [7]$$



FIGS. 4–6 SAND HEAPS PRODUCED OVER RECTANGLES

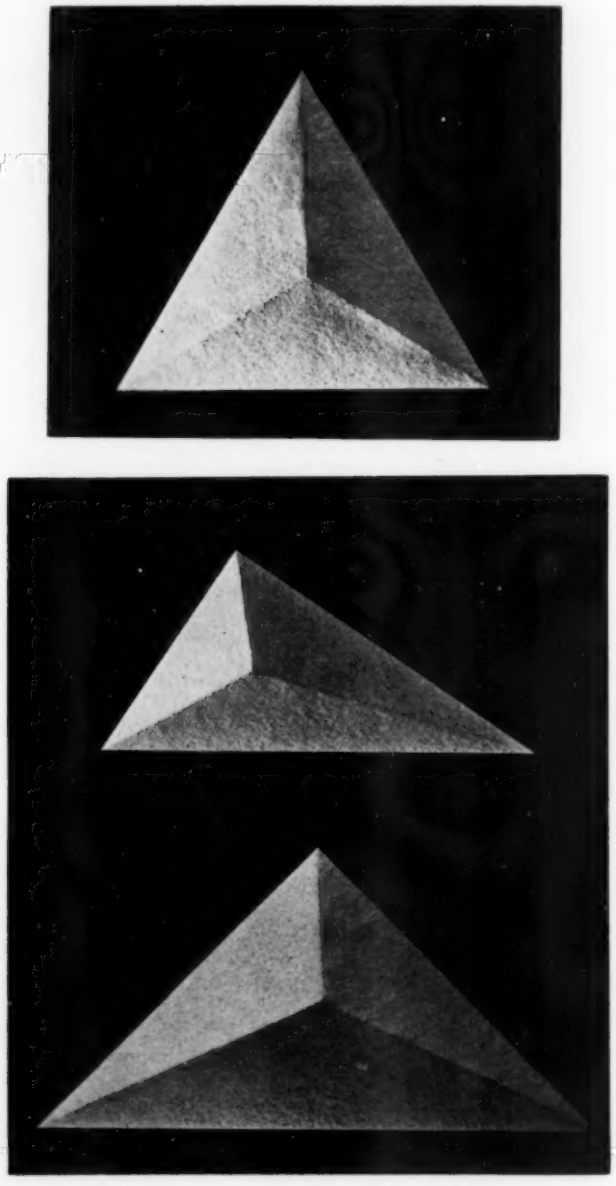
while the condition [5] is equivalent to

$$\frac{\partial F}{\partial x}\,dx + \frac{\partial F}{\partial y}\,dy = 0$$

from which it is seen that on the boundary curve $y = f(x)$

$$F = \text{const.}\ (= 0) \ldots\ldots\ldots\ldots\ldots\ldots [8]$$

Equations [7] and [8] contain the "membrane analogy" of Prandtl. The function F satisfying [7] and [8] for a given boundary curve $y = f(x)$ may be called *the elastic stress function of the cross-section*. If a thin, perfectly flexible skin or membrane, which has been stretched in its plane by a uniform tension t per unit of length and acting in all directions, is considered as attached to the boundary curve $y = f(x)$ of the cross-section for which the torsion problem is to be solved and loaded by a lateral pressure p, this membrane will be slightly deflected out of its



FIGS. 7–9 SAND HEAPS OVER TRIANGLES

plane. The deflections w of such a thin skin must satisfy the equation

$$\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} = \Delta w = -\frac{p}{t} \ldots\ldots\ldots\ldots [9]$$

with the boundary condition $w = 0$ along the curve $y = f(x)$. A comparison of Equations [7] and [9] and of the correspond-

ing boundary conditions shows that they are identical in form. The curved surface w (x, y) into which this membrane is distorted is an image of the elastic stress function F (x, y) of the cross-sectional area which determines the torsional stresses. The curved surface representing the elastic stress function

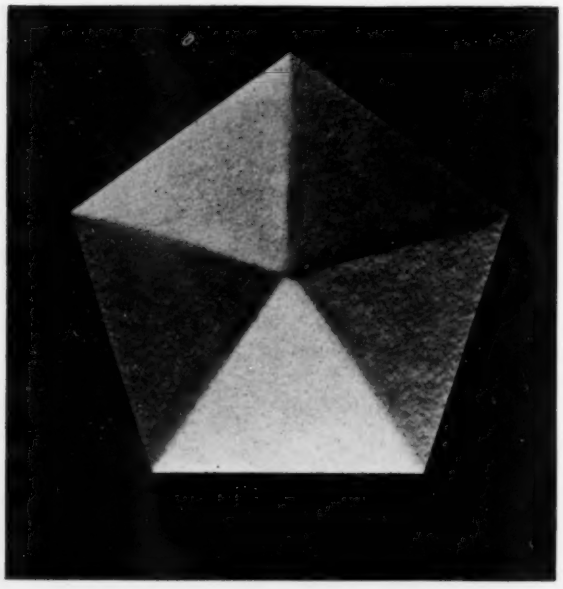


FIG. 10 SAND HEAP OVER PENTAGON

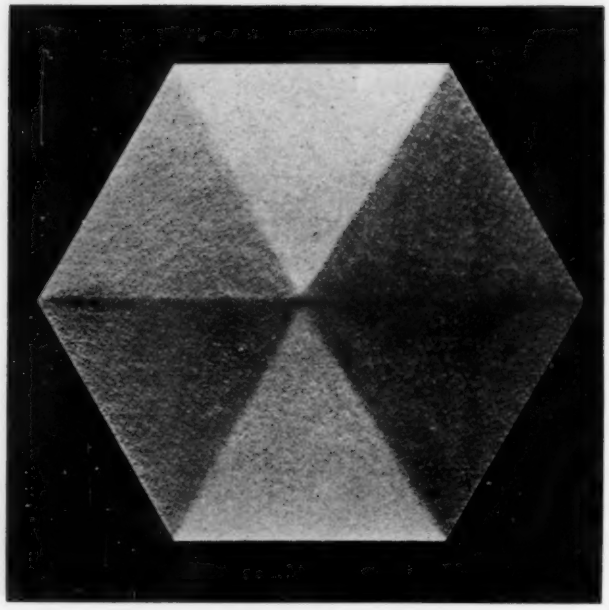
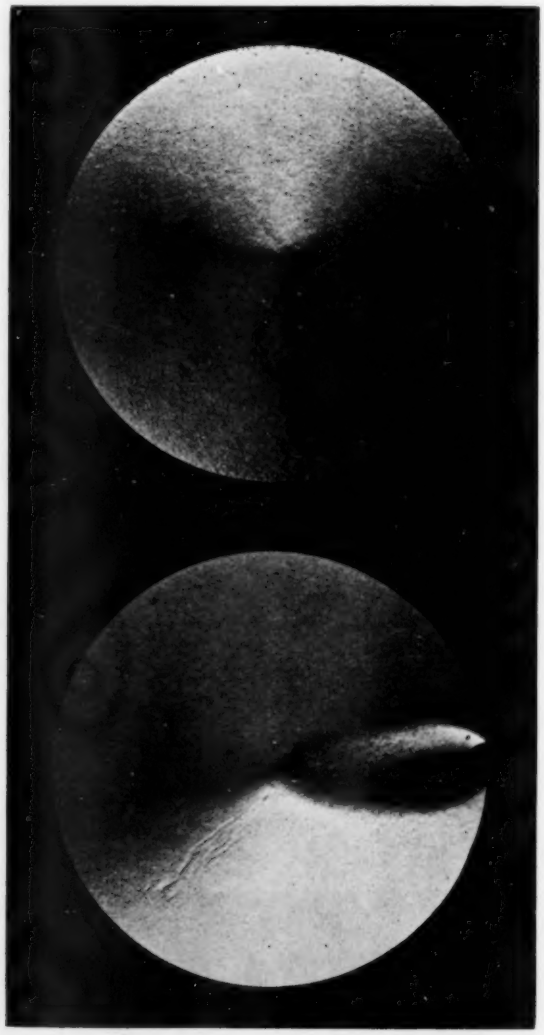


FIG. 11 SAND HEAP OVER HEXAGON

F (x, y) may therefore also be called the "stress bulge" of the cross-sectional area A. According to Prandtl's membrane analogy, the following statements hold: (1) The absolute value of the resultant shearing stress s at any point x, y of the cross-section of a twisted bar is equal to the gradient (to the greatest slope) of the elastic stress function F (x, y) or of the stress bulge of the cross-section, because the resultant shearing stress s is

$$s = \sqrt{s_x^2 + s_y^2} = \sqrt{\left(\frac{\partial F}{\partial x}\right)^2 + \left(\frac{\partial F}{\partial y}\right)^2} \quad \ldots\ldots [10]$$

(2) The resultant shearing stress s is tangent to the contour lines of the stress function F (x, y). The contour lines of F (x, y) are the stress lines of the cross-sectional area of an elastically twisted bar. (3) The torque M is equal to twice the volumetric content of the stress function F over the cross-sectional area, or



FIGS. 12 AND 13 SAND HEAPS OVER CIRCLE AND CIRCLE WITH SMALL CIRCULAR NOTCH

$$M = 2 \int\int F\ (x,\ y)\ dx\ dy \quad \ldots\ldots [11]$$

As the absolute values of the ordinates of the elastic-stress function F, according to Equation [7], are proportional to the angle of twist θ of the bar, Equation [11] determines the torsional rigidity of the cross-sectional area.

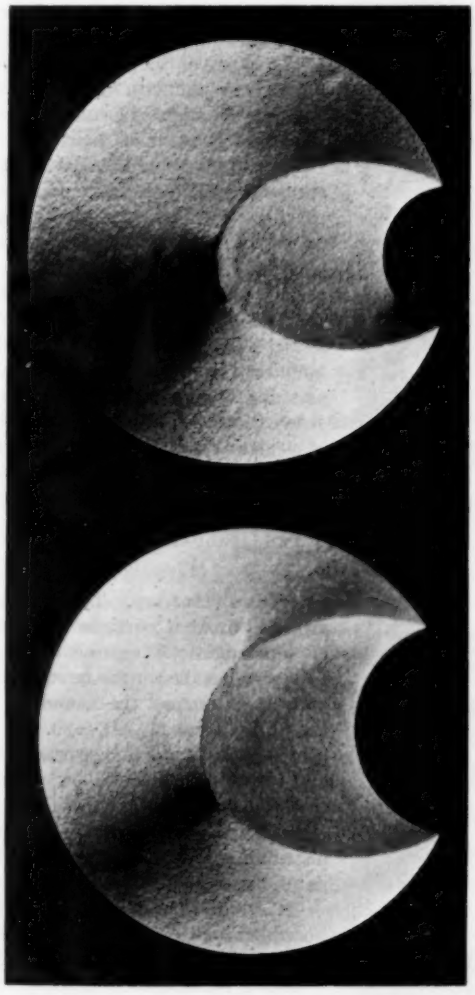
PLASTIC TORSION

The Plastic Stress Function for Torsion. If the torque M by which the bar is twisted is gradually increased, the ordinates of the elastic stress function F (x, y) and consequently the slopes of this surface, which latter determine the shearing

stresses s, will also increase. There will exist a certain twisting moment M_0 at which the greatest shearing stress s_{max} in the cross-section of the bar will just become equal to the yield stress k of the metal in pure shear:

$$s_{max} = k \quad [12]$$

The first yielding in the bar will occur under this torque M_0. If the twisting moment M is further increased, so that $M > M_0$, the yielding will spread out. As the maximum shearing stress under pure elastic torsion will occur mostly at one point of the boundary line of the cross-sectional area (or in more than one point on this line), if $M > M_0$, yielding will spread out, starting from this or these points. For torques M greater than M_1, we must therefore distinguish between certain "plastic" areas A_1, A_2, ...in the cross-section (see shaded portion of Fig. 3) where



FIGS. 14 AND 15 SAND HEAPS OVER AREAS BOUNDED BY TWO CIRCULAR ARCS

the shearing stresses s will reach the yield stress k and in which plastic flow will start to develop and an elastic portion A_0, where $s < k$ and the material of the bar will be stressed only elastically.

In the plastic areas A_1, A_2, ...the components s_x and s_y of the shearing stress s must obviously satisfy the two equations

$$\frac{\partial s_x}{\partial x} + \frac{\partial s_y}{\partial y} = 0 \quad [13]$$

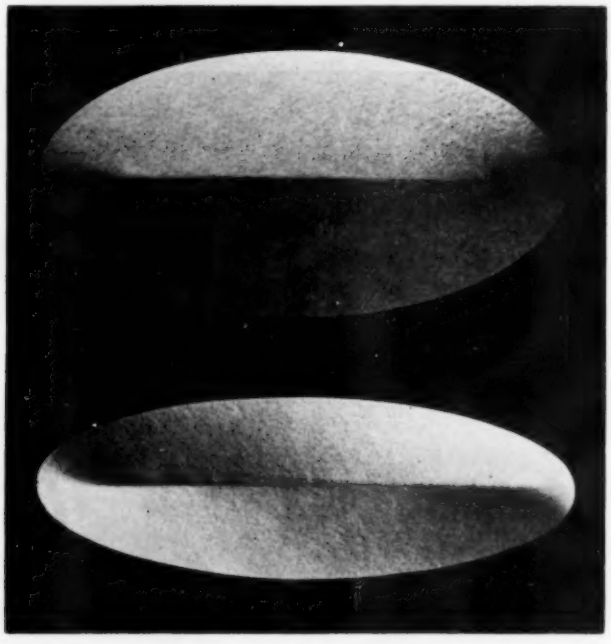
and

$$s_x^2 + s_y^2 = s^2 = k^2 = \text{const} \quad [14]$$

The first of these is the condition of equilibrium. The second expresses the fact that the resultant shearing stress s now reaches the limiting or yield stress k.

Using again

$$s_x = \frac{\partial F}{\partial y}, \qquad s_y = -\frac{\partial F}{\partial x} \quad [15]$$



FIGS. 16 AND 17 SAND HEAP OVER ELLIPSES



FIG. 18 SAND HEAP OVER AREA BOUNDED BY TWO CIRCULAR ARCS AND TWO STRAIGHT LINES

[13] is satisfied, and from [14] is seen that now the function F must satisfy the equation:

$$\left(\frac{\partial F}{\partial x}\right)^2 + \left(\frac{\partial F}{\partial y}\right)^2 = k^2 = \text{const} \quad [16]$$

This new function F will be called *the plastic stress function of the cross-sectional area.* As obviously in the areas A_1, A_2, ...

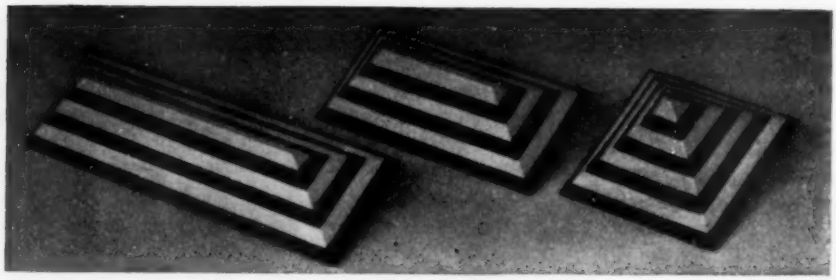


Fig. 19

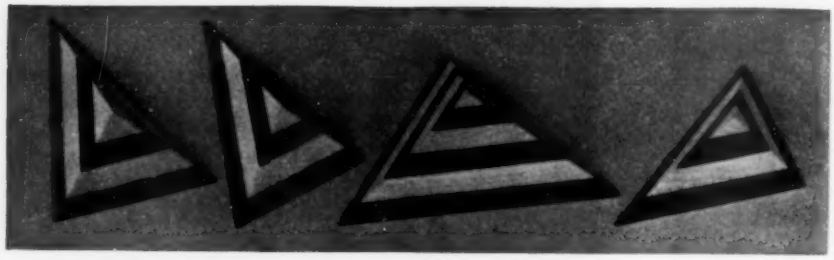


Fig. 20

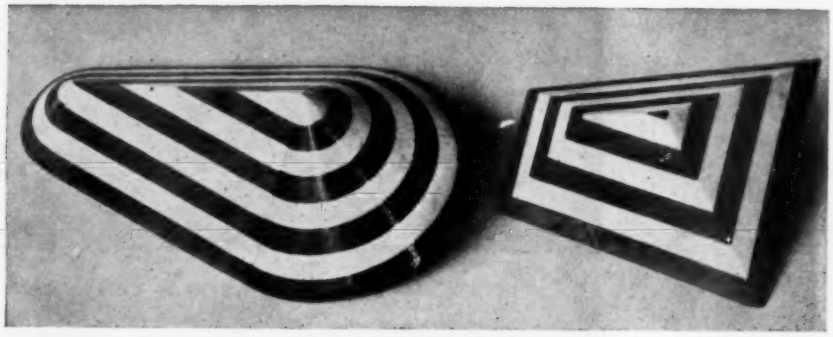


Fig. 21

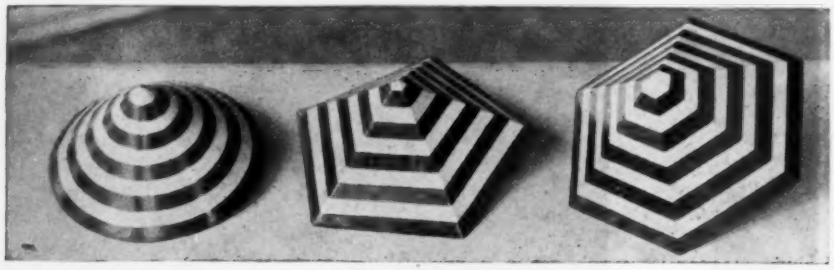


Fig. 22

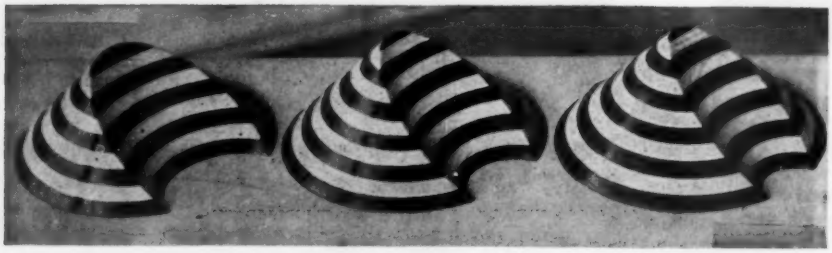


Fig. 23

FIGS. 19–23 WOODEN MODELS OF PLASTIC STRESS FUNCTION FOR TORSION FOR VARIOUS CROSS-SECTIONS

the resultant shearing stress $s = k =$ const. and this latter is also determined by the greatest slopes of the surface $F(x, y)$, it is seen that the surface F must now within $A_1, A_2, \ldots$ have a constant slope. Hence the plastic stress function $F(x, y)$ is determined by one or the other of the two *surfaces of constant slope, which can be erected over the boundary line on both sides of the cross-sectional area of the twisted bar.* The shape of this surface is easily found by sprinkling a fine powder on a piece of cardboard held in a horizontal position and having the same shape as the cross-sectional area. Using sufficient powder, the sloping surface or a "heap" will cover the area, and will be an image of the plastic stress function F of the cross-section.

The slopes of the plastic stress function for a given boundary curve $y = f(x)$, can easily be constructed geometrically or represented by means of their contour lines. Obviously, these lines must be a set of curves equidistant from the boundary curve $y = f(x)$. If, for example, the boundary line is a circle, the contour lines of F will be concentric circles; if in a certain part it is a straight line, the contour lines in the neighborhood will be straight lines parallel thereto. If the boundary curve consists only of circular arcs and of straight portions, the plastic stress function $F(x, y)$ will consist only of straight circular cones and of inclined planes. Several examples of such cross-sections will be referred to later on.

THE GRADUAL TRANSITION INTO THE PLASTIC STATE

Sandheap Analogy. The question now arises as to what portions of the surfaces of constant slope represent the plastic regions when the torque increases further, and also how may the boundary lines of the plastic areas $A_1, A_2, \ldots$ (see Fig. 3) be determined or found, corresponding to a given torque $M > M_0$. The answer to these questions is given by the following experimental representation of the stress distribution in a twisted bar after the yield point has been passed.

On a horizontally placed piece of cardboard having the geometrical shape of the cross-section, sand is heaped so as to form a sloping surface under gravity over the section. From this heap, a negative or "hollow" may be made, or a flat roof of constant slope may be erected above the boundary of the section. If the plane base of this flat roof is closed with a thin membrane stretched in its plane and this latter is loaded by pressure, when the pressure reaches a certain value, certain parts of the membrane will touch the sloping surface of the roof erected above the section.

The free parts of the membrane and those resting on the sloping surface together form the stress surface or the "stress bulge" for a plastically deformed bar in torsion. At those portions of the membrane touching the sloping surface, the metal yields, while at those corresponding to the free parts of the membrane, the metal remains elastic.

The following rules stated for the elastic torsion still hold true: After the yield point has been reached the resulting shearing stress at an arbitrary point x, y of the cross-section is the gradient of the stress surface $F(x, y)$; the twisting moment M of the bar is equal to twice the volume $M = 2\int\int F(x, y)dxdy$ enclosed by this surface; the contour lines $F(x, y) = \text{const.}$ of the complete elastic and plastic stress surface are the stress lines of the cross-section of the plastically twisted bar.

Shape of Plastic Stress Function Produced by Sand Heaps

From the preceding is seen that for a given cross-section, the distribution of stress in a twisted bar at the moment of first yielding will be found by using the shape of the stretched membrane when it first touches the flat roof as the elastic stress function of torsion. However, if the torque is further increased beyond its value M_0 corresponding to first yielding, the thin skin will partially cover the flat roof. In this case the stress function will be represented by the shape of the free part of the membrane, together with the parts of the roof covered by it. There will exist a certain torque M_{max} under the action of which the flat roof will be completely covered by the membrane. Under this torque the twisted bar will yield completely. The stress lines for complete yielding are exactly the contour lines of the flat roof or of the surface of constant slope erected above the boundary line of the cross-sectional area of the twisted bar.

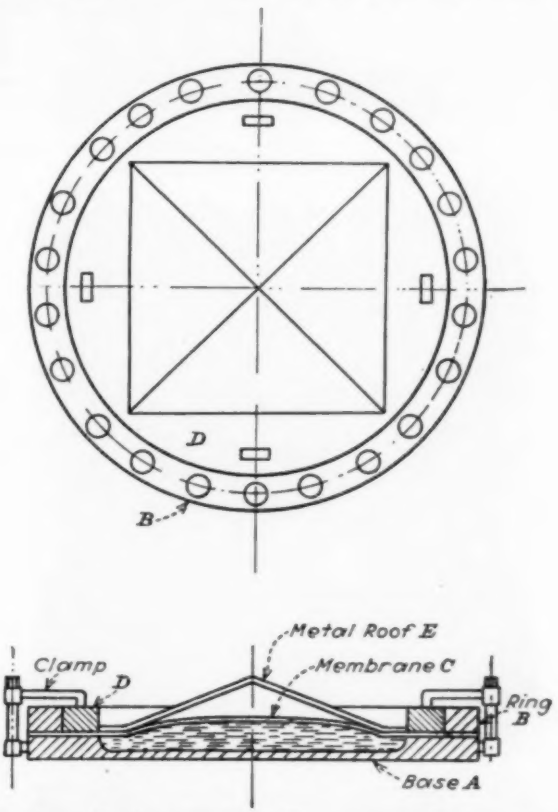
The shape of this latter surface for a given cross-section can be produced by the sloping surfaces of a sand heap having the cross-sectional area as a base. Figs. 4 to 18 are photographs of such heaps of a loose powder which were obtained above brass plates of different shape.[3]

The shape of these surfaces of constant slope can be seen more exactly on the sets of wooden models shown in Figs. 19 to 23. These models were prepared from two different kinds of wood. A dark and a light-colored wood were glued together in alternate layers of constant thickness, and the models cut out from blocks thus formed. In this way the contour lines of the surfaces of constant slope can be obtained automatically. Thus a projection of these contour lines on the base of the wooden models furnishes at once the picture of the stress lines for the case of complete yielding of a twisted bar, or of the plastic-stress function for torsion corresponding to the given cross-section.

Apparatus for the Experimental Determination of Stress Distribution in Torsion Under Partial Yielding

The sand-heap analogy permits, by means of a special apparatus, the study of how plastic flow will penetrate into a twisted prismatical or cylindrical bar of steel. In this apparatus the shape of the stress function is experimentally determined with the help of a thin rubber sheet uniformly stretched in its plane and then loaded by lateral pressure.[4]

The apparatus (Figs. 24, 25, and 26) consists of an aluminum disk A about 12 in. in diameter, upon which a very uniformly stretched thin rubber sheet C is fastened by means of an aluminum ring B and screws. The rubber sheet is partly supported on one side by a flat surface on the aluminum disk, but a central circular portion of the rubber membrane is completely free. The membrane covers a depression in the aluminum disk in which space, by means of a tire pump, pressure can be applied. (See Figs. 24 and 25.) A second exchangeable disk D fitting snugly in the ring B contains the flat roof E. Several such exchangeable disks with different surfaces of constant slope were made. One was of glass, thus permitting observation of the movement of the rubber membrane. In other cases accurately machined brass plates were used. After the disk (with either the glass or metal roof) was fastened on the aluminum disk, pressure was applied below the rubber membrane and the latter allowed to bulge out until finally it partly or almost entirely covered the flat solid surfaces of the roof.



Figs. 24 and 25 Apparatus for Experimental Demonstration of Stress Distribution in Plastic Torsion

To make visible the boundaries of the plastic parts in a cross-

---

[3] The author is greatly indebted to Mr. H. Friedman, of Boston, for having carefully prepared and photographed these sand heaps and also for having carried out tests with two special pieces of apparatus, and which served to demonstrate the stress distribution in twisted bars.

[4] For the case of elastic torsion, Prandtl's membrane analogy was first applied experimentally by Anthes, who photographed the picture of a rectangular network of lines which was reflected by a soap film produced over a hole in a metal sheet. Another method also utilizing soap films was worked out by A. A. Griffith and G. F. Taylor, who measured the deflections of such films by an electric-contact method (Proc. 1st Int. Congress of Applied Mechanics, Delft, 1924, p. 39). A very handy apparatus serving for the same purpose and consisting of a uniformly stretched rubber sheet, over which a difference of pressure could be produced by evacuating the air on one side of the membrane, was demonstrated to the writer by Professor Enger Urbana at the University of Illinois in 1928. The means used in this latter apparatus were adapted somewhat in the one described above and constructed at the suggestion of the writer by H. Friedman at the Research Laboratories of the Westinghouse Electric & Mfg. Company in East Pittsburgh, Pa. Another experimental method for determining the distribution of stress in elastic torsion has been proposed by A. Piccard and W. Tohner utilizing a meniscus of a fluid. (Compare "Stodola Festschrift," J. Springer, Berlin, 1929.)

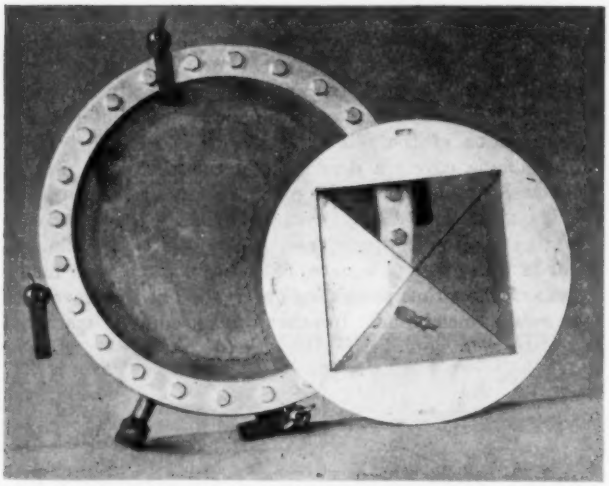


FIG. 26 APPARATUS FOR EXPERIMENTAL DEMONSTRATION OF STRESS DISTRIBUTION IN PLASTIC TORSION

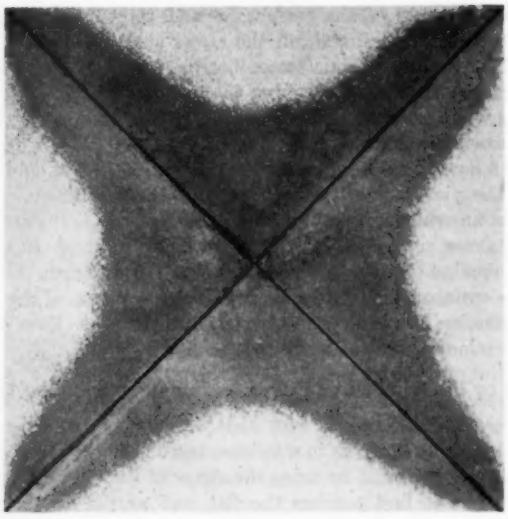


Fig. 29

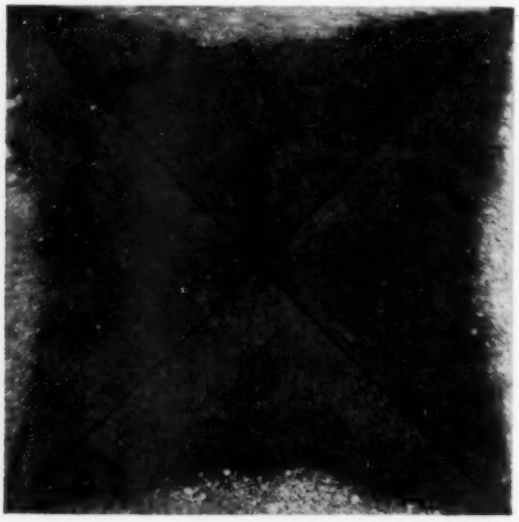


Fig. 27

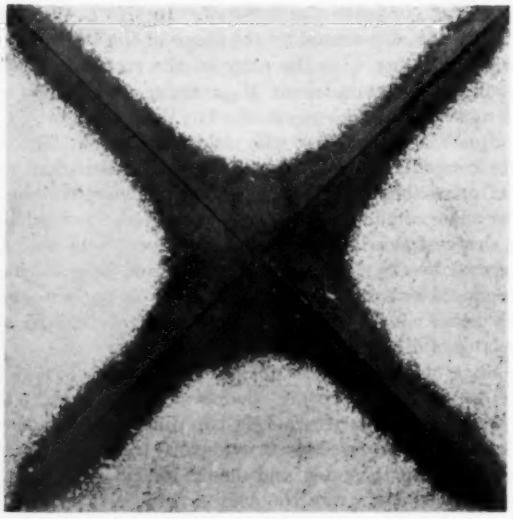


Fig. 30

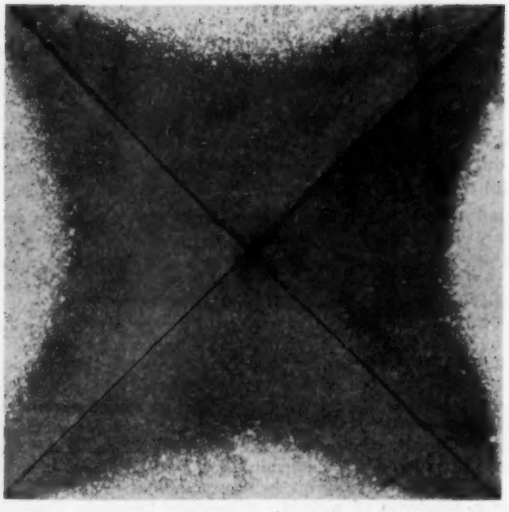


Fig. 28

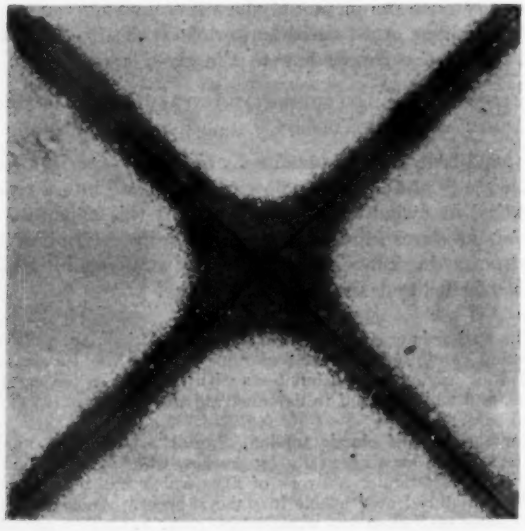


Fig. 31

FIGS. 27–31 DEVELOPMENT OF PLASTIC REGIONS IN A TWISTED BAR OF SQUARE CROSS-SECTION PRODUCED BY MEMBRANE
(The white areas are those stressed above the yield stress.)

section of a twisted bar (see following paragraph), the rubber membrane was covered with fine white powder and the roof with a thin film of oil. After pressure was applied to the apparatus and the rubber sheet forced against the faces of the flat roof, the powder adhered to those portions where the rubber membrane touched the roof, thus making visible the plastic parts of the cross-section which appeared white in the photograph taken of the hollow side of the roof after it had been removed. Fig. 26 at the right side shows one of the exchangeable disks filled with a flat four-sided pyramid of glass plates. A series of tests showing the gradual extension of the plastic regions in a twisted bar with different cross-sections (square, rectangular, trapezoidal section with rounded edges) are illustrated in Figs. 27–43.

(*a*) *Square.* In Figs. 27–31 it will be seen that the white areas are first formed in the middle of the sides of the square, where the shearing stress first reaches the yield point. With increasing twist or torque these white areas representing the portions which become plastic in the cross-section extend inward, finally leaving only a narrow, black cross (Fig. 31); this shows that the elastic portion in the cross-section is reduced until it practically consists of two narrow strips crossing at right angles and coinciding with the diagonals of the square, these diagonals being the projections of the corresponding four edges of the four-sided pyramid, which in this case is the plastic stress function of the cross-section.

(*b*) *Rectangular Section* (ratio of sides 2:1). Figs. 32–37 show how the plastic area increases with increasing torque. Yielding first starts in the center of the long sides, but soon after two further plastic areas begin to develop along the short sides. In the last two photographs only a narrow region along the five edges of the roof remains black.

(*c*) The mode of penetration of the plastic regions for a twisted bar whose cross-section consists only of straight lines and circular arcs is shown in Figs. 38–43. Compare also Figs. 18 and 21, which show the corresponding sand heaps and surfaces of constant slope.

(*d*) *Torque for Complete Yielding.* In any of the preceding cases it is easy to compute the torque M_{max} under which the bar will yield completely.

On the assumption that deformations are independent of yield stresses, it follows from this schematic representation that the values of the twisting moment M approach asymptotically with the increasing angle θ a value M_{max}, which latter is given by twice the value of the space enclosed *by the plastic stress surface.*

The condition where all parts of a twisted bar yield may be designated as the completely plastic state.

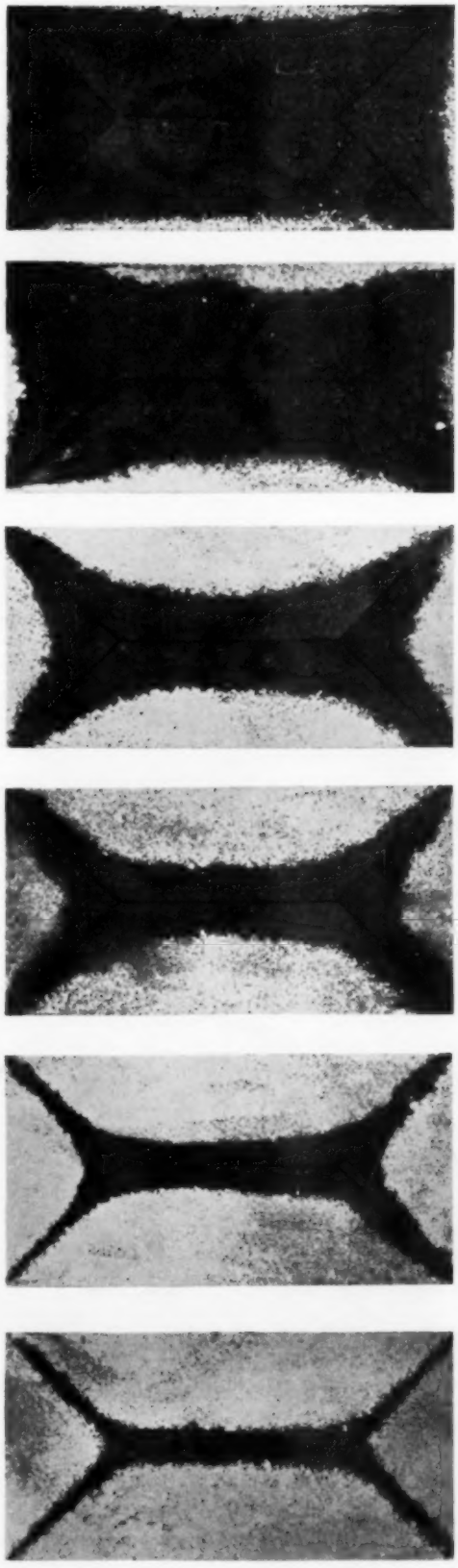
For example, taking a bar with a circular cross-section of radius r, the plastic-stress surface is a circular cone having the equation

$$F = k\,(r - a)$$

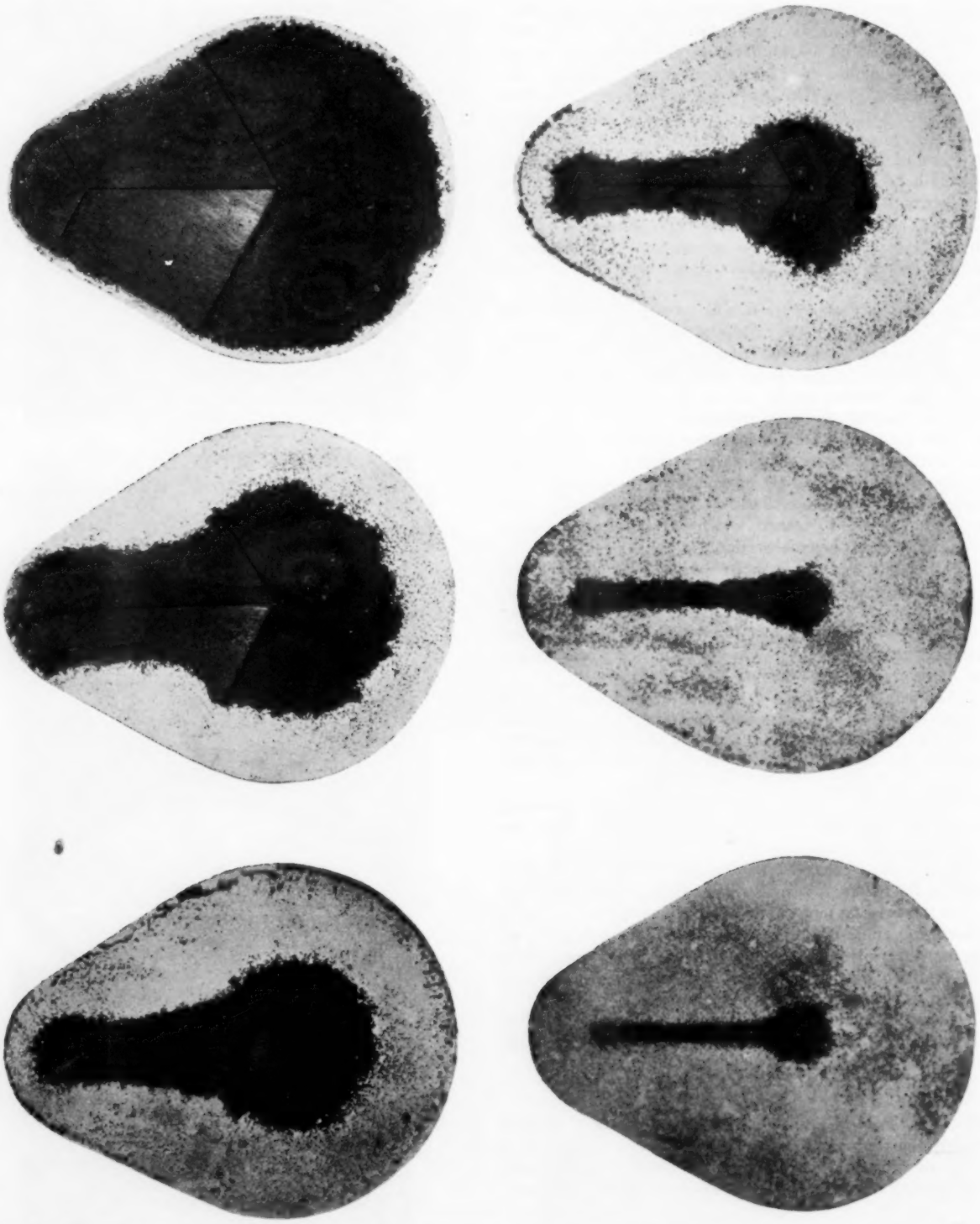
In this, k is the yield stress for pure shear. For $r = 0$, $F = -ka$, and the cone has a height $h = ka$ and a volume $V = \pi a^2 h/3$. According to our analogy, the twisting moment acting on the bar is equal to twice the volume of the cone, or

$$M_{max} = 2V = \frac{2\pi k a^3}{3}$$

For a bar having a cross-section of the shape of an equilateral triangle, the plastic-stress surface is a three-sided pyramid. Each side has a slope equal to $\partial F/\partial n = k$, since the shearing stress is equal to the slope of the stress surface. Therefore the pyramid has the height $h = ka/2\sqrt{3}$. The area of the base is $a^2\sqrt{3}/4$, its volume $V = a^3 h/4\sqrt{3}$, and the twisting



FIGS. 32–37 DEVELOPMENT OF PLASTIC REGIONS IN A TWISTED BAR OF RECTANGULAR CROSS-SECTION



FIGS. 38–43 DEVELOPMENT OF PLASTIC REGIONS IN A TWISTED BAR HAVING ITS CROSS-SECTION BOUNDED BY TWO CIRCULAR ARCS AND TWO STRAIGHT LINES

moment when the bar is completely plastic is equal to twice the volume or

$$M_{max} = 2V = \frac{a^2h}{2\sqrt{3}} = \frac{ka^3}{12}$$

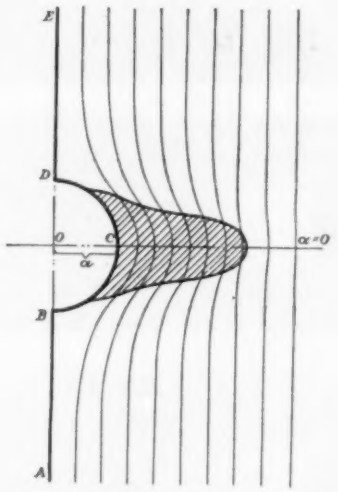


FIG. 44 STRESS LINES AROUND SEMICIRCULAR GROOVE IN STRAIGHT EDGE. SHADED AREA REPRESENTS PLASTIC REGION

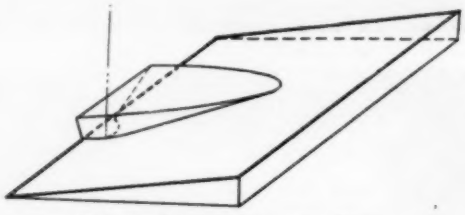


FIG. 45 MEMBRANE AND CONE

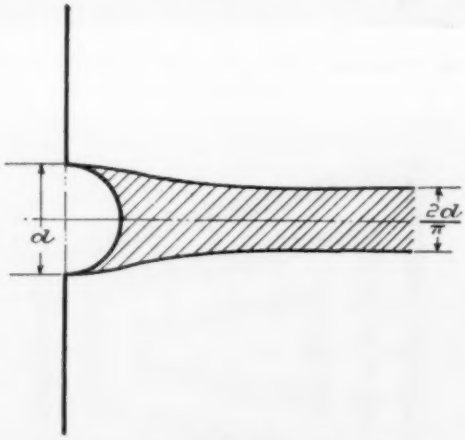


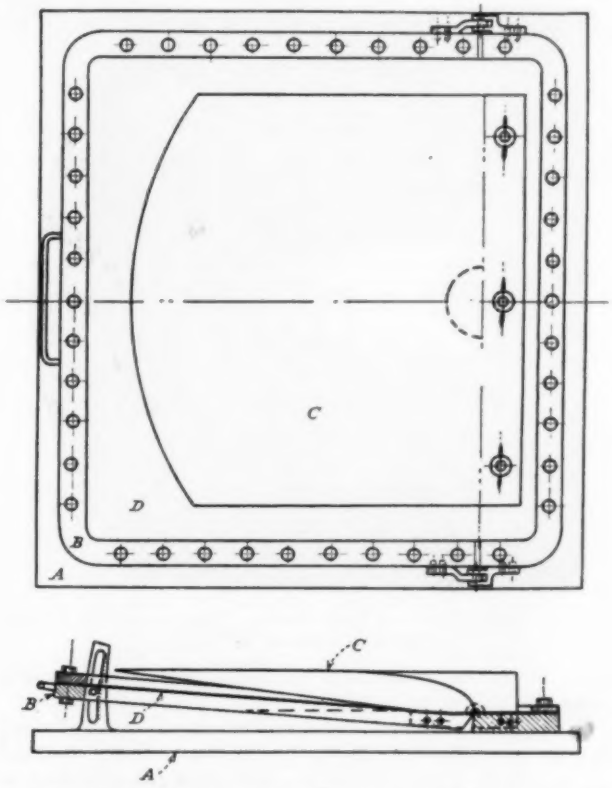
FIG. 46 PLASTIC AREA NEAR NOTCH

For a bar with a rectangular cross-section having the sides a and b (a less than b), the moment in the completely plastic state is

$$M_{max} = \frac{ka^3}{6} + \frac{ka^2(b-a)}{4}$$

EFFECT OF SMALL HOLES OR GROOVES IN A REGION SUBJECTED TO PURE SHEAR

(*a*) *Longitudinal Groove With Semicircular Cross-Section.* For the case of a cylindrical bar in torsion, having a small longitudinal groove of semicircular cross-section, qualitative estimates of the mode of yielding may easily be made. We shall first assume that the radius a of the groove is small in comparison with the diameter d of the bar. It is then permissible to replace



FIGS. 47 AND 48 APPARATUS FOR EXPERIMENTAL DEMONSTRATION OF HOW YIELDING SPREADS FROM A CYLINDRICAL HOLE
(*A*, base; *B*, bed frame; *C*, micarta cone; *D*, rubber membrane.)

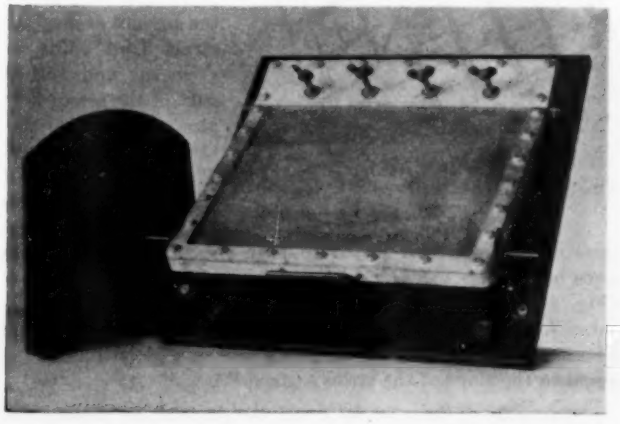


FIG. 49 APPARATUS FOR THE EXPERIMENTAL DEMONSTRATION OF SPREADING OF PLASTIC REGION FROM A CYLINDRICAL HOLE OR NOTCH

the boundary circle by the tangent. The disturbances in the shape of the stress lines may be determined under the assumption that at a considerable distance from the groove a constant shearing stress acts.

If a perfectly elastic material having a straight edge with a small semicircular groove is subjected to pure shear acting in

a direction parallel to the straight edge (Fig. 44), the stress lines in the neighborhood of the groove may be determined easily by means of the potential theory or also with the help of the membrane analogy. At considerable distances from the groove the stress lines will be parallel to the straight parts of the

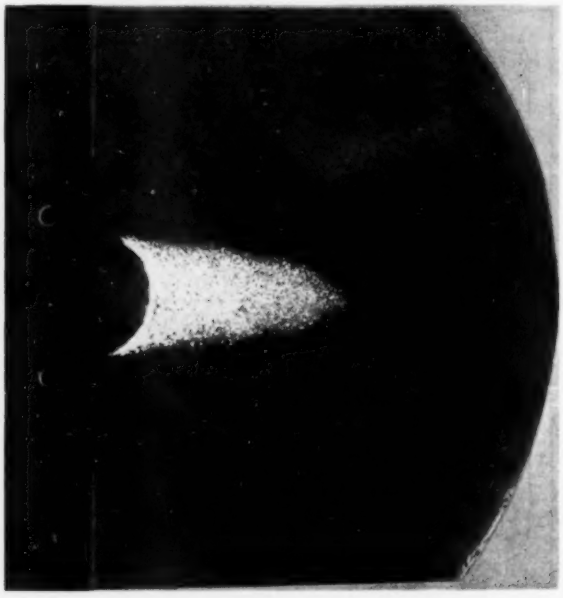


FIG. 50 FRONT VIEW OF FLAT CONE. WHITE AREA IS PLASTIC REGION DESCRIBED IN TEXT

edge; in the neighborhood of the groove, however, they will follow the semicircle.

One has only to conceive of a thin membrane attached along an edge *ABCDE* of the body, this membrane being stretched in a plane inclined to the plane of the cross-section (Fig. 44). The slope of the inclined plane is a measure of the amount of shearing stress (the value of s) in the undisturbed part of the stress field at a considerable distance from the groove. The contour lines of the stretched membrane are the stress lines of the region subjected to pure shear. Evidently these lines are closest together in the neighborhood of the point C (Fig. 44) at the edge of a groove, and at a considerable distance from it they become more and more nearly parallel to the edge. The shearing stress s which exists at a given point P is equal to the slope of the stress surface F.[5]

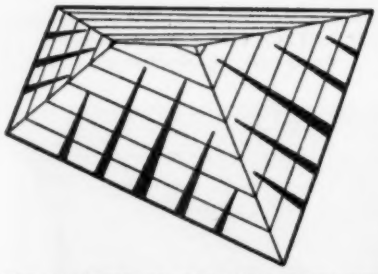
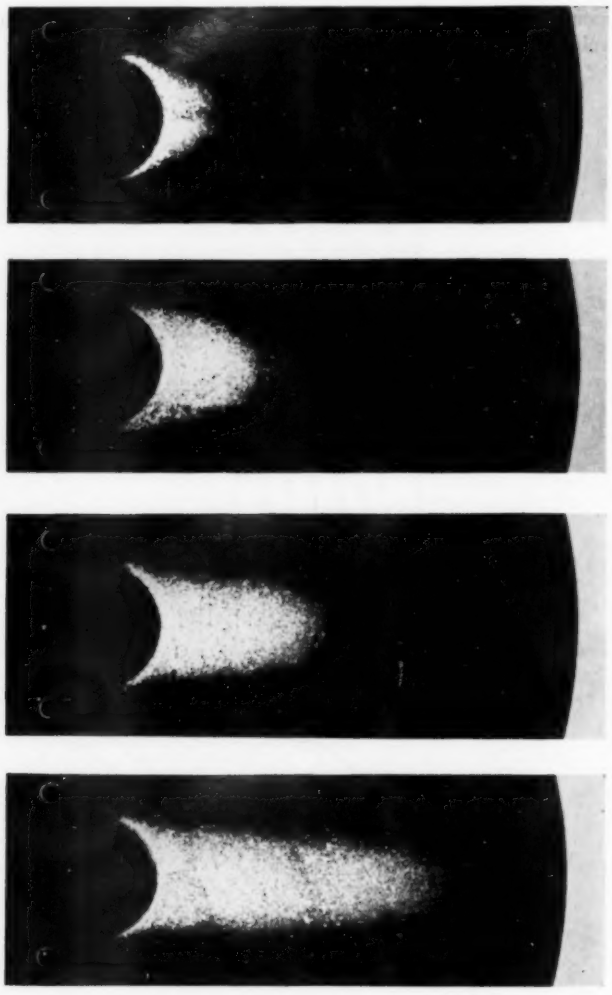


FIG. 55 POSITION OF SLIP LINES RELATIVE TO PLASTIC-STRESS SURFACE

[5] By means of the potential theory the shape of the elastic stress surface F may easily be determined in the polar coordinates r and α:

$$F = c \cos \alpha \left(\frac{a^2}{r} - r \right)$$

By forming the derivatives of this surface the maximum shearing stress is found to occur at the edge of the groove at C, where $r = a$ $\alpha = 0$, and $s = 2c$, and is twice as large as the shearing stress $s = c$ at a great distance from the groove. Since for great values of x $F = -cx$, the shearing stress there is $s = -\frac{\partial F}{\partial x} = c$. The figure



FIGS. 51–54 SPREADING OF PLASTIC REGION FROM A CYLINDRICAL HOLE

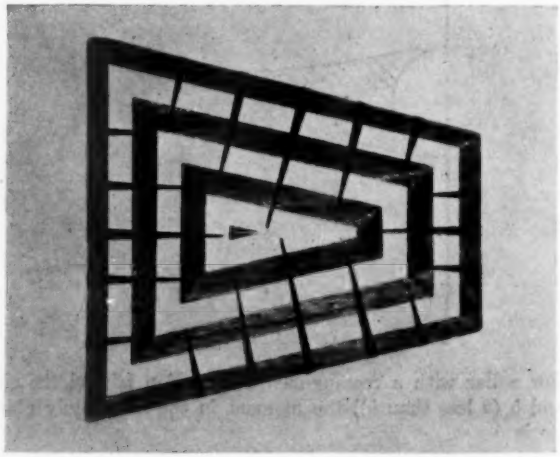


FIG. 56 POSITION OF SLIP LAYERS IN CROSS-SECTION OF TWISTED BAR RELATIVE TO CONTOUR LINES OF PLASTIC SURFACE

of the contour lines in this case is exactly the same as that of the well-known streamlines of a fluid flowing along a straight wall on which the half of a long circular cylinder has been fastened (Fig. 44).

As the inclination of the plane of the thin membrane is increased, all shearing stresses are increased until finally the stress at the point C reaches the yield point k. Then the membrane must be inclined in a plane $F = kx/2$ with respect to the plane of the cross-section. In order to determine the limits of the plastic area as the inclination of the plane of the thin membrane is further increased, we must think of a sloping surface in the form of a cone with radius $r = a$, erected as shown in Fig. 45, the cone being pointed downward. If the slope of the membrane at C (Fig. 44) becomes larger than $k/2$, certain neighboring parts of the membrane will lie upon the cone. These parts lying on the

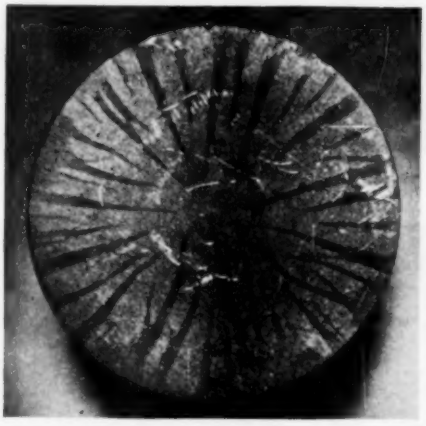


FIG. 57 ETCHING OF CROSS-SECTION OF A TWISTED BAR

cone (the corresponding area is shaded in Fig. 44) represent the plastic area of the cross-section.

At the suggestion of the author, E. Trefftz[6] has determined mathematically the shape of the membrane for this case.

From the standpoint of an experimental comparison, the limiting case (not considered by Trefftz) of what shape the flow areas assume if the shearing stress $s = c$ approaches the value of the yield stress $s = k$ for pure shear is of a special interest. The writer found[7] that in the case of a small semicircular groove, plastic flow penetrates into the stressed material with the increasing shearing stress along a narrow strip the width of which finally approaches a finite value $b = 4a/\pi$.

Thus in a body stressed by pure shear and having a well-defined yield stress in shear or tension, a thin plastic layer starts at a semicircular notch of diameter d. When at greater distances from the notch the shearing stress approaches the yield stress of the material for pure shear, this plastic area extends to infinity and has a width $2/\pi$ times the diameter d of the notch. (Fig. 46.)

The plastic stress function is in this case a circular cone

$$F = k\,(a - r),\ (r > a)$$

These considerations lead to a somewhat paradoxical conclusion; for, according to what has been stated previously, the presence of a small longitudinal groove on the surface of a material strained by pure shear is sufficient to cause the small

[6] Über die Spannungsverteilung in tordierten Stäben bei teilweiser Überschreitung der Fliessgrenze. *Zeit. Ang. Math. Mech.*, vol. 5 (1925), p. 64. In this a second noteworthy case is treated, namely, the origin of the first flow area at the inner reentrant edge of a twisted steel angle iron.

[7] Proc. 2nd Int. Congress for Applied Mechanics, Zürich, 1926.

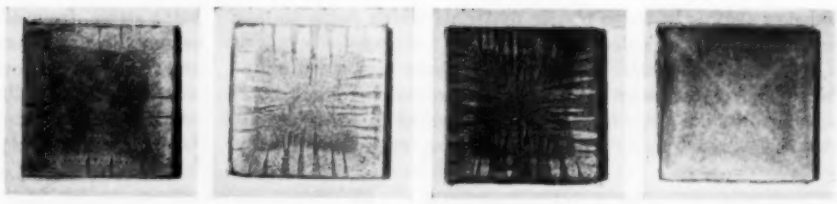


Fig. 58

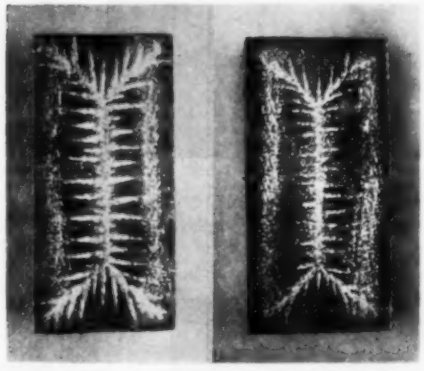


Fig. 59

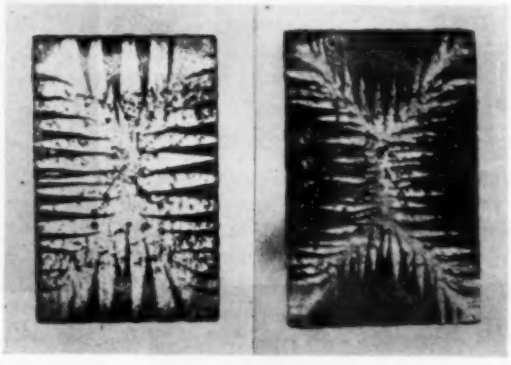


Fig. 60

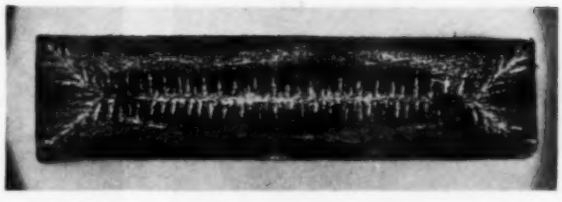


Fig. 61

FIGS. 58–61 ETCHINGS OF CROSS-SECTIONS OF TWISTED BARS

plastic area at the edge of the groove to extend deeply into the material, if the shearing stress at this edge is increased to the yield point. Moreover from this it follows that it is not possible, by increasing the shearing stress, to produce plastic deformation elsewhere than in the narrow flow area. The apparent discrepancy between this conclusion and actual test results which would show that it is possible to raise the stresses higher than the yield point is easily explained in recalling one of the idealizing assumptions, on which it was based; namely, that the yield stress should be considered as independent of deformation. On the other hand, the above conclusions have been confirmed by many observations on the development of the thin flow layers

and flow figures in those plastic materials which begin to flow under a constant or a decreasing stress.

(*b*) *Cylindrical Hole.* In the neighborhood of a small cylindrical hole of diameter d in a region stressed by pure shear, two narrow flow areas of width $2d/\pi$ develop in an analogous way as described under (*a*) if the shearing stress is gradually increased to the yield point. The stress distribution in the neighborhood of a cylindrical hole in a plate of infinite extension, which is subjected to pure shear, corresponds completely with that of the above-mentioned case of a cylindrical groove; the two cases are mathematically the same.

From these considerations, it must be concluded that in a plastic material, which begins to flow under approximately constant stress, from all those places where the stresses were concentrated in the elastic state, narrow plastic regions will radiate and the plastic deformation will be localized to thin layers of a comparatively small thickness, along which it will spread inward into the stressed material.

(*c*) *Mechanical Apparatus for Demonstrating Spreading of Plastic Flow Starting at a Small Notch or Hole.* The preceding conclusions can be confirmed by means of a mechanical apparatus (Figs. 47–49) consisting of a sheet of rubber uniformly stretched in its plane on a rectangular frame of metal, which may be tilted about one of its sides. Above the rubber membrane, a flat truncated semi-cone with a semicircular cross-section, machined from a block of black micarta, is rigidly fastened by means of

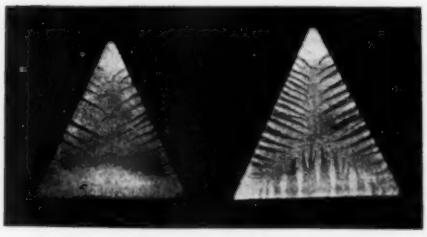


Fig. 62

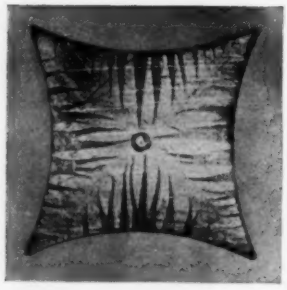


Fig. 63

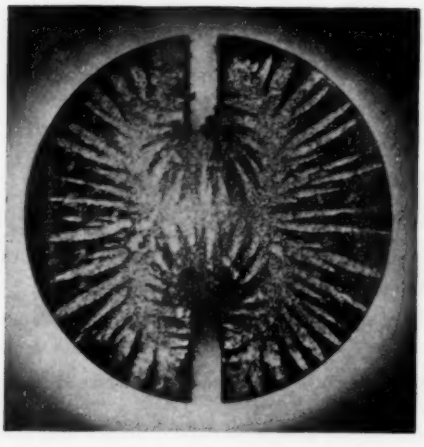


Fig. 64

FIGS. 62–64 FRY ETCHINGS OF CROSS-SECTIONS OF TWISTED BARS

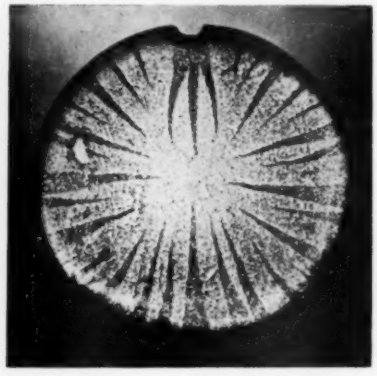


Fig. 65

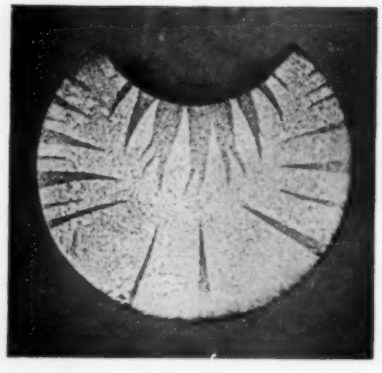


Fig. 66

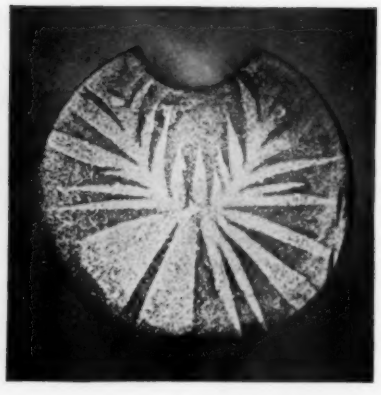


Fig. 67

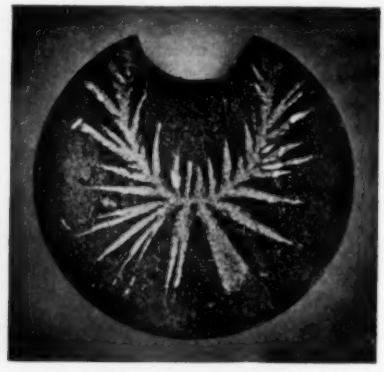


Fig. 68

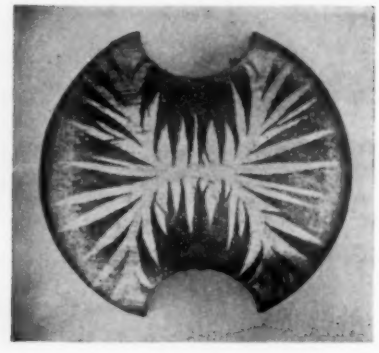


Fig. 69

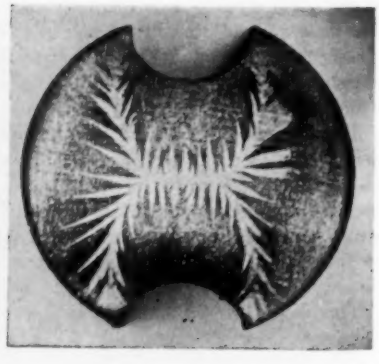


Fig. 70

FIGS. 65–70 ETCHINGS OF CROSS-SECTIONS OF TWISTED BARS

the screws shown in Fig. 47 to the flat stationary plate of the frame in the manner indicated in Figs. 47 and 49. The flat cone of black micarta required for the model test is also shown at the left of Fig. 49 and in a front view in Fig. 50. The rubber membrane was covered with a white powder and the cone with a thin coating of oil. Then the frame with the rubber membrane was tilted through a certain angle. Where the micarta and rubber surfaces came in contact, the white powder adhered to the black micarta. The white areas thus obtained on the surface of the cone could be photographed and illustrate the shape of the plastic region (of the shaded area in Fig. 44) as it spreads inward from the edge of the semicircular notch or groove, as the slope of the tubber sheet is gradually increased.[8]

The spreading of the plastic region from the edge of a semicircular notch as the shearing stress s increases is shown in the set of photographs (Figs. 50 and 51–54) taken by Mr. Friedman. The angles of inclination of the plane of the rubber sheet are proportional to the shearing stress in the undisturbed parts of the body subjected to pure shear.

Slip Layers in Twisted Steel Bars

(*a*) *Flow Layers. Torsion Tests With Iron Bars.*[9] The position of the slip layers in the plastically deformed parts of a twisted bar may be predicted with the help of the plastic stress surface. For a soft metal such as wrought iron, the slip layers coincide approximately with the surfaces of maximum shear or of maximum shearing displacements. These slip layers are therefore approximately perpendicular to each other. At an arbitrary point inside a twisted bar one surface of maximum shearing stress coincides continuously with the plane of the cross-section. The other surface of maximum shear is parallel to the axis of the bar. The traces of the second system of slip layers must remain perpendicular to the cross-section.

After the yield point has been passed in a twisted iron bar, very regular layers or markings may actually be shown to exist. In these the iron is apparently deformed more than in the neighboring layers. These layers may subsequently be made visible in the cross-section of the bar by means of Fry's etching method. In such etched sections the flow layers appear as dark strips and lines. A series of such etchings on soft-iron bars, using Fry's method, are shown in Figs. 57–70: In Fig. 57 for circular cross-sections, in Figs. 58 to 61 for rectangular cross-sections, in Fig. 62 for triangular cross-sections, and in Figs. 63–64 for two other cross-sections. In the etched cross-sections the traces of one system of slip layers are indicated. Traces of the second system of slip layers could only be noted in exceptional cases.

In a twisted iron bar, the flow layers appear uniformly when the twisting moment reaches values corresponding to those along the horizontal portion BC of the moment curve $M = f(\theta)$, Fig. 1.

The first flow layers occur at such values of the twisting moment at which the outermost part of the cross-section has already been plastically deformed and the moment curve begins to bend over into the horizontal branch. With increasing twist new layers form beside the old ones, while the latter become longer and thicker. The flow layers appear in the etched cross-sections of the photographs as wedge-shaped areas, the apexes of the wedges pointing toward the less-stressed parts of the cross-section (Fig. 57).

As a rule, it may be concluded that the black flow lines in the cross-sections run mainly perpendicular to the stress lines (the projections of the contour lines of the plastic stress surface). Moreover these flow lines always remain perpendicular to the edge of the cross-section. Thus they run in the direction of the projections of the lines along which water will run off the flat roof erected above the cross-section and representing the plastic-stress surface (Figs. 55, 56). With the more severe deformations, the black strips finally cover the whole cross-section. In the extreme right etching of the square cross-section of Fig. 58, for example, which corresponds to the most severely twisted bar, the elastic areas may only be recognized as bright areas along the diagonals of the square. In other words, the four ridges of the stress surface F, which is here a four-sided pyramid, are projected on these diagonals.

(*b*) *Bars With Longitudinal Grooves.* In Figs. 65–70 are shown etchings of twisted bars having longitudinal grooves of different shapes and dimensions, obtained by W. Bader. On the flow figures found on the cross-sections of a cylindrical bar with a small semi-circular groove, Fig. 65, it will be noticed that three wide flow layers extend from the groove.

Construction of the contour lines of the plastic-stress surface for complete yielding is in these cases much facilitated by the condition that the surfaces in question consist entirely of either planes or straight circular cones. Therefore the intersections of the surfaces of constant slope which correspond to the edge of the cross-section can be obtained easily by constructing the mutual intersections of these planes and circular cones. In the photographs of the wooden models, the contour lines may be seen.

According to the assumptions made, in the completely plastic condition the elastic area in the cross-section must shrink until it coincides with the ridges (more exactly the projections of the ridges upon the cross-sectional area) of the plastic-stress surface of the sand heap which may be conceived to be erected over the cross-section. It will be seen by comparison of the etchings with the corresponding contour figures that the middle lines of the white, jagged, ridge areas in the photographs of the etchings agree quite well with the curves of the ridges of the plastic-stress surface shown in the photographs of the wooden models or the corresponding sand heaps.

Acknowledgment

The author wishes to acknowledge his indebtedness to S. M. Kintner, assistant vice-president of the Westinghouse Elec. & Mfg. Company, and to J. M. Lessells, manager of the Mechanics Division of the Westinghouse Research Laboratories, for the use of facilities needed to conclude this investigation begun in Germany. Special acknowledgments are due to Dr. W. Bader for having prepared the etchings of the cross-sections of twisted steel bars, and to H. Friedman for having prepared the two sets of apparatus, the sand heaps, and wooden models, and photographed them.

Addendum

Since the paper was presented, three wire models mounted on micarta plates have been prepared from surfaces of constant slope of elliptical cross-sections. These surfaces can easily be represented by wire models since the curve in which the surface of constant slope of an ellipse intersects the vertical plane of symmetry erected above the base ellipse and containing the major axis, is itself an ellipse. Furthermore, if the generatrices of the surface of constant slope are so chosen that they start

[8] E. G. Coker in his valuable paper, "Elasticity and Plasticity" (Thomas Hawsley lecture, Proc. Inst. Mech. Eng., vol. 1, 1926), has published some photographs which show the shape of the plastic stress function by means of sand heaps in a similar case; namely, for a twisted shaft having one or more keyways.

[9] The author is much indebted to Dr. W. Bader, who has carried out these tests—see his Doctor thesis, 1927, Univ. of Göttingen. See also *Zeit. V.D.I.*, No. 10, p. 317, 1927.

in points along the base ellipse which are situated on a set of equally spaced lines perpendicular to the major axis of the ellipse, the generatrices will intersect the vertical symmetry plane in a set of points, which are in their horizontal projection also equidistant from each other. Figs. 71–73 showing three such wire models of surfaces of constant slope erected above ellipses having ratios of major axis to minor axis of 1:2, 1:4, 2:3 were constructed in this way. Fig. 74 shows a wire frame on which

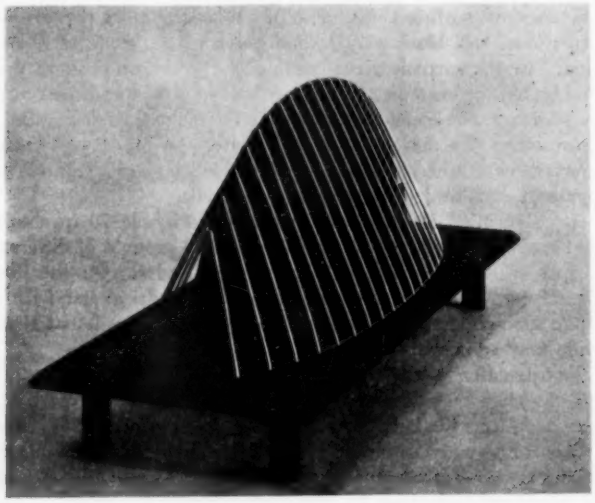


FIG. 71

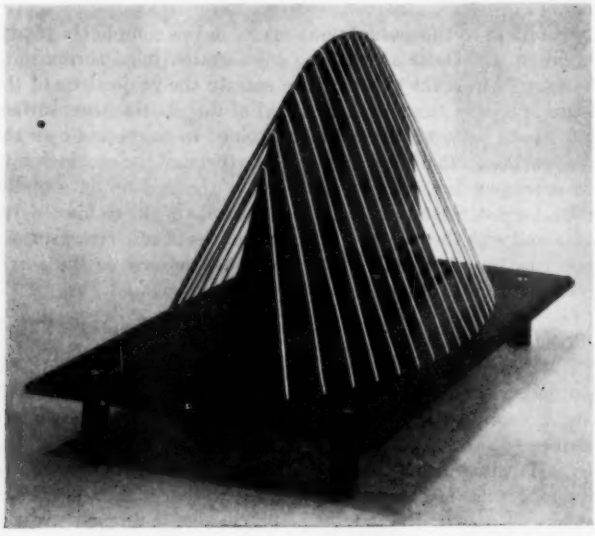


FIG. 72

brass disks of various shapes can be attached and covered by sand for demonstrating sand heaps.

Discussion

E. G. COKER.[10] The author has explained the subject matter so lucidly that there seems little to criticize, but there is one matter which appears to call for some further explanation in the last section dealing with slip layers in twisted steel bars. Referring to the right-hand photograph of Fig. 58, there are faint traces of a square band of light inside and bordering the contour of this specimen, and similarly there are more marked indications of light-colored bands bordering the longer sides of the sections shown in the photographs, Fig. 59 and 61. These ultimately join up with the feathery diagonals connecting the central ridges with the angular points of the sections. It would be of interest if the author would describe in some detail the meaning to be attached to these bands running parallel to the contours and near to them. The author is to be congratulated on a paper of such scientific interest and importance.

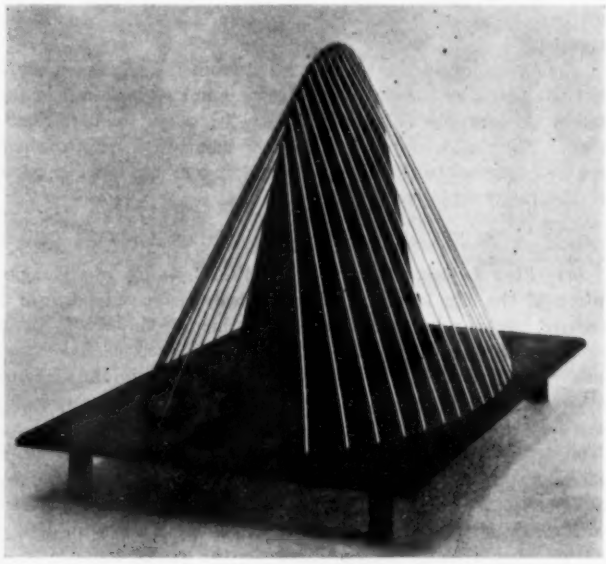


FIG. 73

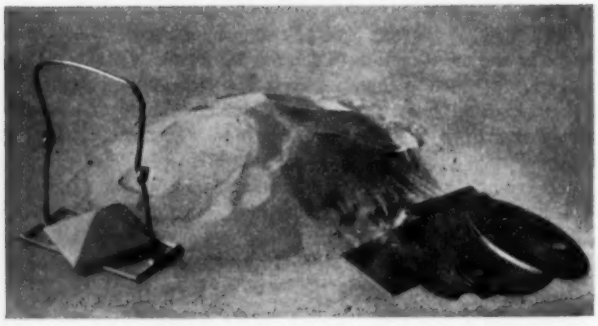


FIG. 74

G. M. EATON.[11] The mathematician to whom a complicated equation conveys a physical picture may consider it almost childish and at least of small practical value to resort to models as the author did in the research which preceded his paper. But to the average mechanical engineer he offers a new tool. He enables us to start developing in our minds a qualitative sense of plastic actions, which we cannot achieve from equations.

If we can sharpen this new sense finely enough to warn us when we are working in a dangerous range, then we shall know when to call for the assistance of the specialist in plastic phenomena.

The writer sincerely hopes that the author will continue to

[10] Professor of Civil and Mechanical Engineering, University of London, University College, London, England. M.Inst.C.E.

[11] Spang-Chalfant & Co., Inc., Ambridge, Pa. Mem. A.S.M.E.

work along this line and give us periodic reports similar in form to the present paper, because we can put this kind of information to work promptly.

This paper is an illustration of one way that applied mechanics can be "sold" to industry.

GLEASON H. MACCULLOUGH.[12] The paper is so excellent and valuable that the writer hesitates to raise a question on even one relatively minor point.

In considering the effect of a small longitudinal groove of semicircular cross-section in a twisted cylindrical bar, the author states in the third paragraph below Fig. 57 that the plastic area extends to infinity in a direction perpendicular to the longitudinal axis and has a width of $2/\pi$ times the diameter d of the notch. From this it follows, so he says, that it is not possible, by increasing the shearing stress, to produce plastic deformation elsewhere than in the narrow flow area. As we interpret this statement, it means that in an overstrained cylindrical bar with a small semicircular keyway, the plastic area never extends outside of the radial region of width $2d/\pi$ mentioned above. The author admits a seeming paradox here.

How to reconcile this view with the picture of the plastic area that would be obtained by the orthodox method, namely, the membrane stretched over the hole with an opening like that of the shaft and inflated until it touches the "roof" of the plastic stress function as per Fig. 13, is a puzzle to the writer. The smaller the hole, the more nearly this membrane (in the so-called orthodox method) touches the roof in equal width zones all the way round the circumference, however large it may be.

To be sure, the author's arguments at a first reading seem convincing. While they may hold for the initial appearance of the plastic area, the writer wonders if they do really hold at the greater distances from the notch. He would add here that he is not questioning the dimension $2d/\pi$ obtained in the tilted-sheet analogy.

It would seem that perhaps the roof which the author implicitly used in the tilted-sheet method is not the same as would be used with the circle. In this latter or orthodox method, the roof consists of a circular cone (no matter how large the radius) and an inverted cone. In the tilted-sheet analogy the circular cone has been transformed into an inclined plane, but the inverted cone remains an inverted cone of the same size.

The point the writer wishes to emphasize is this: Should not this inverted cone have been transformed into some other surface by the same transformation which carried the upright cone over into the oblique plane? It is agreed that at the edge the conditions of the problem are the same in the two methods, but as one penetrates into the shaft, he begins to doubt if they are. In fact, the center of the round bar or the vertex of the upright cone has been changed from a point to a line equal in length to the circumference, yet the inverted cone has not been changed. The distortion is akin to that on a map made according to Mercator's projection: the map of Canada undergoes a distortion; can we leave that of Greenland unchanged?

The radial lines in the etched bar of Fig. 65 appear to bear out the author's conclusion if we fail to note that the plastic area is not confined to lines radiating from the keyway. Thus the writer still wonders if the above views may not explain away the paradox.

M. F. SAYRE.[13] The author is to be congratulated on the success with which he has shown the results both of the "Sand-heap" method and of the "slip-layer" method in his illustrations.

There are only two possible additions which the writer might make to the discussion. One is to emphasize the point that the "sand-heap" results are essentially limited to mild steel and that they will not apply rigorously to non-ferrous metals, for which no yield point can be found, nor will they apply to the full extent to the high-strength, heat-treated or cold-worked materials which are commonly used for springs. The same basic idea, of course, applies to plastic flow in these cases, but due to the absence of a sharply defined yield point, the solution is not quite as simple.

The present paper does not include a discussion of the residual stresses in the bar as a result of previous plastic flow. This, it is believed, the author has treated in a previous paper. The experimental work carried on under the Research Committee for Mechanical Springs has brought out rather clearly that in many cases one effect of plastic flow is to change the modulus of elasticity of the material under future loadings. This change in modulus may be great enough to have a relatively large quantitative effect upon the magnitude of the residual stresses.

C. R. SODERBERG.[14] The general problem of "strength" of engineering structures involves problems in elasticity for the determination of deformations and stresses, and problems connected with the failure of our materials for determining the effects of these stresses. It is only fairly recently that some one has had the courage to ask questions concerning the plastic behavior of our materials as well. The writer believes that the incentive for studies in plasticity actually came from the geologists, and the direct returns have possibly been greater in that science than in engineering. Restricting ourselves to engineering, however, the general problem of strength will include elasticity, plasticity, and failure. The last heading is somewhat unsatisfactory; the German word *Anstrengung* is more to the point, but unfortunately there is no equivalent of this word in the English language. It is a regrettable fact that, among engineers, the problem of strength is frequently approached as a problem in elasticity. When the stresses are known the problem is considered as solved, while in reality the most difficult part of the problem lies in determining the ultimate effects of the stresses.

The work of the author and others in clearing up many difficult problems of plasticity is of great value in this connection, because the most important of our engineering materials are ductile materials having a stress-strain diagram which very closely resembles that of Fig. 2 of the paper. The ultimate outcome of these studies still remains hidden; there are very few positive results which can be shown to have a definite cash value in engineering practice. A disappointment is inevitable, therefore, if the present paper is approached with the object of obtaining ready-made and simple solutions for predicting the failure of ductile materials under torsion. The writer does not by any means consider it necessary to justify every bit of our brain work by the practical value of the results obtained, but industry is not always of the same opinion. The only thing that he can add to the author's interesting paper, therefore, is a plea for further efforts along the path that he and his collaborators have started.

S. TIMOSHENKO.[15] The paper gives a derivation of Prandtl's well-known analogy as well as the application of this analogy in studying plastic torsion. The illustrations and photographs

[12] Professor, Mechanical Engineering, Worcester Polytechnic Institute, Worcester, Mass. Assoc-Mem. A.S.M.E.

[13] Associate Professor of Mechanical Engineering, Union College, Schenectady, N. Y. Mem. A.S.M.E.

[14] Westinghouse Electric and Manufacturing Co., East Pittsburgh, Pa. Assoc-Mem. A.S.M.E.

[15] Professor of Mechanical Engineering, University of Michigan, Ann Arbor, Mich. Mem. A.S.M.E.

give a very clear picture of the distribution of plastic flow in twisted bars. The example of a small groove is especially interesting.

Referring to the apparatus for experimental investigation of stress distribution, a further development is very desirable to show us with what accuracy the magnitude of stresses can be obtained from such experiments. From the experiments made at the University of Michigan by Prof. L. H. Donnell, the writer knows that it is necessary to give to the rubber membrane a very large uniform initial stretching, which is very difficult to do accurately. Another point which must be kept in mind is the necessity of keeping the slope of the surface of the loaded membrane very small. Only for such a surface does the analogy between membrane and torsion problems hold.

If a sufficient accuracy for the rubber-membrane method can be established by further experimenting, this will be an important result which can be utilized in solving our torsional problems.

Author's Closure

The author appreciates very much the further points that have been brought out by the discussion, and will endeavor to answer the various questions raised and to reply to the interesting remarks expressed in connection with the matter treated in the paper.

Prof. E. G. Coker's valuable observations refer to certain bright bands in the etched cross-sections of twisted steel bars. In some of the etchings these light-colored bands run in a parallel direction with the boundaries of the cross-sectional areas. According to what was said in the paper, such bands should not be expected to appear. The undisturbed "fluidal structure" or the changes in the structure of the twisted steel bars due to plastic flow, as they were made visible in the etchings of the cross-sections by using Fry's method, consist, in the case of torsion, of narrow dark strips, which were described in the paper and made visible in many of the etched sections by Dr. Bader. These were found to progress in a direction perpendicular to the boundary line of the cross-sectional areas.

When comparing some of the other etchings (Figs. 57 to 70), however, it will perhaps be noticed that in these figures more or less pronounced differences in the density of color within the darkened areas can be well recognized. The tendency of this darkening effect can especially be seen in the first and third figures (from the left) of the four etched sections of the bar with a square cross-section shown in Fig. 58. In these etchings, in the center of the figure, a square appears which is darker than the surrounding parts. It was shown by former tests that this darkening within a square area could be detected in many rolled bars of both square and circular cross-section. This darkening is due to a non-uniformity of the structure of the steel ingot from which the bars were rolled. In general, the inner portions of cast ingots contain all the impurities (segregations) due to the more rapid solidification of the outermost layers of the ingots, and to the further concentration of the impurities in the inner core. These segregations are well known to the rolling-mill engineer[16] since they can still be detected by suitable methods after a bar has undergone considerable amount of reduction in cross-section by rolling.

That this explanation is correct is further checked by the two etchings in Fig. 62. These are the etched sections of a twisted bar having an equilateral triangle as its cross-section. These bars were machined from a bar with a square cross-section (the same bar as that of the specimens shown in Fig. 58). Parts of the dark square can be seen in the triangular areas of the etchings in Fig. 62, leaving an outer portion brighter, while the fluidal structure, consisting of the dark streaks, runs in these cross-sections perpendicular to the edges, as it should be. In the bars with a flat rectangle as cross-section, the core, later appearing dark, was further deformed by the rolling process into the central longitudinal areas, the border lines of which were the faint traces of a white band running parallel to the long edges of the rectangular cross-sections.

Coming now to Professor MacCullough's remarks, the question which he raises and wishes to answer in arriving at conclusions different from those drawn by the author in the paper, seems to be more a geometrical than a mechanical one and might perhaps, if repeated in a slightly different form than it was stated, be expressed as follows: Given are two circles in a plane, a large one of radius R and a small one of radius a, intersecting each other at right angles (Fig. 75). A thin membrane uniformly stretched in its plane is considered to be fastened to the periphery $ABCDA$ of the figure bounded by the two arcs of circles, and loaded by lateral pressure p until it is slightly disturbed. The inclination of this very flat membrane with respect to the plane of reference along the periphery of the figure $ABCD$ under a given lateral pressure p is to be known. The membrane along the greater part of the periphery of the large circle CDA will obviously have about the same inclination; this latter will, however, decrease toward the corners A and C, where it will vanish. Along the circumference of the smaller circle ABC the membrane will be inclined steeper than along ADC. If now while the radius of the smaller circle a is kept constant, the radius of the larger circle is increased gradually, while at the same time the *average inclination of the membrane along the larger circle and the stresses which hold the membrane stretched are kept constant*, obviously the pressure p must gradually be decreased in order to maintain above the larger span the same slopes along the circumference ADC. If R tends to become larger and larger compared to a, the lateral pressure p will tend to gradually become zero. The membrane will no longer need to be loaded by lateral pressure, but only through the stresses which hold it stretched. The parts of the circle ADC near the corners A and C will become straight lines, when R/a increases indefinitely. If we confine the further discussion only to the portion of the membrane in the neighborhood of the circle ABC, considering, for example, only distances r from the center C_2, which are still small if compared with R but large when compared with a (say, e.g., $r = 20a$, while $R = 100r$), the

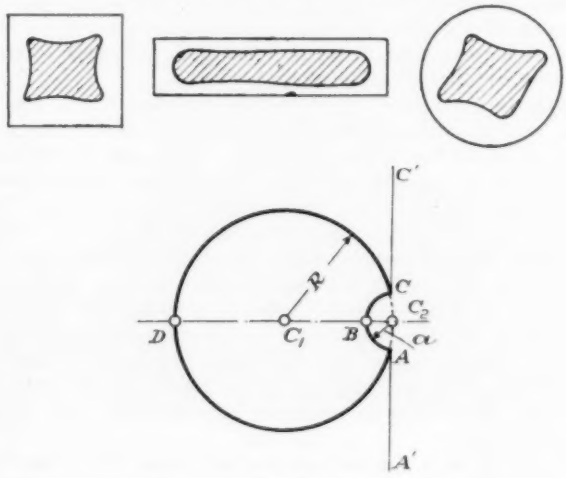


Fig. 75

[16] Cf., for example, some etchings of the sections of rolled bars obtained from steel ingots in Oberhoffer's book, "Das technische Eisen," 2nd Ed., 1925, J. Springer, Berlin. (Fig. 412 on p. 416.)

membrane can be fastened along a "*plane*" or a "rectangular frame" instead of along the circle with "infinite radius R." This frame would consist of four straight lines and contain the semicircular indentation ABC on one side, as indicated in Fig. 45 or as realized by the rubber model, Fig. 49.

Similar reasoning must now also apply to the shape of the limiting roofs or to that of the two cones by means of which the lateral deformation of the free membrane above $ABCDA$ is to be restricted. For the finite figure $ABCDA$ there are two equal straight circular cones of the same angle of inclination to be constructed above the two circles, one with its apex above the plane and the large circle ADC, and the other one, "inverted," above the arc ABC. The arc ABC is not changed when R is increased. Hence the inverted cone does not change. The radius R of the base of the other cone is, however, gradually increased. This means that its axis is displaced toward the left of Fig. 75, its apex C_1 finally moving toward an infinitely distant point on the line BC_1, while B still is kept stationary. The region of this cone, in which one will finally be interested, thus more and more closely approaches a certain plane containing the straight lines CC' and AA' and still having the same inclination as both former cones. It can easily be seen that the inverted cone erected above the small circle ABC intersects the cone erected above the large circle ABC in an ellipse (the line of intersection of both cones degenerates in the special case considered here into an ellipse). In the case where $R = \infty$, while a = constant, the ellipse obviously becomes a parabola, because a plane which is parallel to *one* generatrix of a circular cone intersects this cone in a parabola.

The author does not agree with one of the conclusions arrived at by Professor MacCullough, namely, that the smaller the semicircle ABC, the more nearly the membrane would touch the conical roof, consisting also of the *larger* of both cones along a zone of equal width all around the entire circumference (of the larger circle ADC). If one conceives of an inverted cone being erected above the semicircle ABC together with the plane containing AA' and CC', and inclined by the same angle as the cone, both will intersect in the parabola just mentioned. If he now considers a membrane pressed for a moment against the concave side of a roof which would consist of these two intersecting surfaces, he sees at once that such a membrane, if, *permitted again to deflect freely*, could never cover the regions near the parabola and the two corners A and C. The membrane will touch only the inverted cone along the line BC_1, but not the straight or plane portions of the roof—at least not at finite distances from the semicircular indentation ABC. Employing metaphor to explain these things in a simpler manner, Professor MacCullough asks whether, considering the distortions on a map of Canada made according to Mercator's projection, we can leave those of Greenland unchanged. Might not the question be answered in this manner? "To get a picture of the surroundings of Toronto or Quebec with sufficient accuracy, can we not leave Greenland out altogether since it is too far distant from the part which we are at this moment interested in mapping?"

The author agrees with Professor Sayre that the analogies used to visualize the stress distributions of bars are limited to those materials which have a stress-strain curve like that in Fig. 2. This was clearly stated when enumerating the restrictions under which the conclusions were arrived at.

With reference to the question of residual stresses in a bar which has been twisted above the plastic limit and then unloaded, the author would point out the possibility of getting—at least under certain restricting assumptions—also a first picture of those stresses. All that would be necessary would be to subtract from the "plastic-stress function," as it was represented for the case of complete yielding (by the slope of the sand heaps), the ideal "elastic-stress function" corresponding to the same cross-section and having the same volume as the first one. The differences of the ordinates of the functions would represent the stress function for the residual stresses. However, how far this could be utilized for spring steel is not clear, on account of the considerably different shape of the stress-strain curve of such steels as compared with that of Fig. 2.

Mr. Soderberg points to certain important problems in which mechanical engineers are directly and immediately interested in receiving answers. It must indeed be admitted that immediate applications of torsion cases where plastic flow enters are not often encountered, at least not in machine design. However, the questions raised by Professor Sayre would seem to indicate that some further useful applications of methods, similar to those treated here could be found in springs, where, for example, torsion is often carried to very high stresses and where a deviation from Hooke's law is apparent. On the other hand, knowing about certain cases in which certain important quantities can be more exactly predicted or determined by model tests or by other mechanical means, may perhaps aid in predicting what must be expected in more complicated cases.

The matter of failure brought up by Mr. Soderberg might also perhaps be looked upon from a similar point of view. As far as plastic deformations would precede the final breaking of a member subjected to high stresses—and a more exact discussion of the conditions of rupture would be considered as constituting the main part of the problem—the author thinks that a future analysis of such more or less complicated phenomena might sometime be approached by trying to describe more exactly what processes combine in the act of failure. A discussion of the question of rupture of ductile metals, for example, would then lead automatically to a description of what occurs first during the small plastic deformations and then later during the large ones which precede the rupture. Thus one would be led to considerations similar to those treated in the paper in the case of torsion. In other cases, however, where, for example, fatigue enters, ductility might not be of such importance inasmuch as the change of the actual distribution of stresses while fracture was proceeding might, to a certain extent perhaps, be determined from the theory of elasticity.

The case of the small hole in a material subjected to pure shear treated in the paper might, for example, throw light on a number of phenomena observed in mild steel, which heretofore have been thought to be caused by what might perhaps be termed more metallurgical than mechanical reasons. The conditions under which the Lüders lines (the narrow layers of plastically deformed material) appear in mild steel were formerly thought to depend on metallurgical reasons; however, the author has endeavored to show that the formation of these well-known lines can be explained by considering mainly mechanical reasons: namely, a well-defined yield stress and a local stress concentration combined with a slight instability of the plastic equilibrium.

Referring to certain difficulties encountered if numerical values of the quantities used in the paper were to be obtained by using similar apparatus, as those mentioned here, the author has pointed out on a former occasion that some of these difficulties mentioned by Prof. Timoshenko might be overcome. To obtain, for example, a membrane of rubber with a uniform stress in all directions, it is advisable to hold it stretched above a circular disk or ring by means of a large number of screws spaced at equal distances along the circumference of a circle. If the holes in the membrane are bored along a circle having a diameter that is considerably smaller than the circle on which

the centers of the screws are situated, a uniform stretch will result, if, by stretching the membrane, the holes in the membrane are made to coincide with the centers of the screw holes in the disk.

APM-53-4

Flow of Gases at a Rate Exceeding the Acoustic Velocity

By O. G. TIETJENS,[1] EAST PITTSBURGH, PA.

This paper deals with the flow of gases at high velocities, that is, at velocities exceeding the velocity of sound in the gas under the existing conditions of temperature and pressure. The discussion starts with the development of the fundamental equations of flow and shows why the acoustic velocity assumes such critical importance. In particular, the flow of gas through a De Laval nozzle is considered, the natural assumption of adiabatic flow being made. The so-called compression shock is discussed and the conditions under which these shocks occur, their connection with the acoustic velocity, and their effect on the flow through the nozzle, are pointed out.

IT IS well known that gases and steam change their behavior completely as soon as their velocities exceed the acoustic velocity. This happens in several cases. Not only does the velocity of flow in steam and gas turbines almost always exceed the acoustic velocity, but in internal-combustion engines, especially in the flow through the valves and into the cylinders, this condition also exists. In addition, the behavior of gases in explosions has to be mentioned in this connection. Here the velocities are usually extremely high, in some cases twenty times the acoustic velocity, and even higher. Also many phenomena in the extensive field of ballistics come under this classification.

The theoretical treatment of flow at velocities far below the acoustic velocity is widely known, especially the application of the Bernoulli equation and the equations of the potential theory. However, this cannot be said of the theoretical treatment of high-velocity flow. The statement that the acoustic velocity connotes a sort of limit velocity is often loosely adopted without a clear idea as to just why the acoustic velocity plays this role.

It is hoped, therefore, that this paper may contribute to broadening the knowledge of this interesting and important part of aerodynamics. In this field Dr. L. Prandtl at the University of Goettingen has been one of the outstanding contributors and on his lectures there much of this discussion is based. To make its perusal easier, references to other papers or textbooks, which might somewhat shorten the text, are intentionally omitted. Instead, it is attempted to develop from the very beginning the fundamental formulas with the simplest use of mathematics possible, and in particular to apply these formulas to the flow through a nozzle as an example.

I—Relation Between Pressure and Velocity

We start with the fundamental law of mechanics: namely, that the product of the mass of a particle times its acceleration is equal to the sum of the forces acting on the particle. Applying this fundamental law to an infinitesimal gas particle having the shape shown in Fig. 1 we have, using the notation given in that figure,

$$\underbrace{\rho dF ds}_{\text{mass}} \times \frac{dw}{dt} = \Sigma X \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

where $\rho = \gamma/g$ is the density, i.e., the mass per unit volume.

Since the motion of the particles of a fluid takes place along certain streamlines, it is possible to express the velocity w of the particle at any instant as a function of the time t and the distance s measured along its particular streamline, or $w = f(s, t)$. We can write for the total differential

$$dw = \frac{\partial w}{\partial s} ds + \frac{\partial w}{\partial t} dt$$

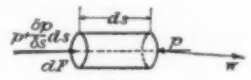


Fig. 1

and therefore for the longitudinal acceleration of the particle,

$$\frac{dw}{dt} = \frac{\partial w}{\partial s} \frac{ds}{dt} + \frac{\partial w}{\partial t}$$

or, since $ds/dt = w$,

$$\frac{dw}{dt} = w \frac{\partial w}{\partial s} + \frac{\partial w}{\partial t}$$

Although for some problems which deal with high-velocity flow of gases, for example, the flow of gas through engine valves, the change of velocity with time is essential and may not be neglected, still for most problems we can make the assumption that the flow is steady, i.e., that the velocity is the same at any point throughout the period of flow but of course generally varies from point to point. Such kind of flow we have, for instance, in the nozzles of steam and gas turbines. Neglecting $\partial w/\partial t$, therefore, we have

$$\frac{dw}{dt} = w \frac{\partial w}{\partial s}$$

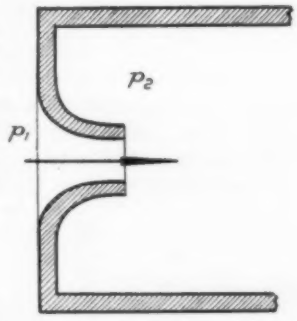


Fig. 2

Regarding the forces summated in the right-hand member of [1], it should be stated that the force of gravity acting on each particle is eliminated by the aerostatic lift.[2]

With respect to the forces due to the internal friction or viscosity of gases, we know that they are generally very small compared with the inertia forces in the left-hand member of [1]. Therefore in most cases it is permissible to neglect the friction forces altogether. Consequently the pressure difference, or,

[1] Research Department, Westinghouse Electric & Manufacturing Company. Mem. A.S.M.E.

Contributed by the Applied Mechanics Division and presented at the Semi-Annual Meeting, Detroit, Mich., June 9 to 12, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] However, in cases of flow where the force of gravity causes differences in density due to large differences in altitude (meteorology), it has to be taken into account.

more exactly, the pressure gradient, remains as the only force in the right-hand member of [1]. However, as the pressure gradient generally has almost the same direction as the velocity—meaning by this that in surfaces perpendicular to the cylinder axis the pressure is constant—the pressure forces acting on the wall of the cylinder can be neglected. The only remaining forces, therefore, are the pressure forces on the two bases on the cylinder.

Thus we have for [1]

$$\rho dF ds \times w \frac{\partial w}{\partial s} = pdF - \left(p + \frac{\partial p}{\partial s} ds\right) dF$$

or

$$w \frac{\partial w}{\partial s} = -\frac{1}{\rho} \frac{\partial p}{\partial s} \quad \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

Integrating this expression along a streamline from point 1 to point 2,

$$\frac{w_2^2 - w_1^2}{2} = \int_2^1 \frac{1}{\rho} dp \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

In those cases where the pressure differences from 1 to 2 (and therefore also the velocity differences) are small enough to permit neglecting the change in density due to them, we have, with ρ = constant, the well-known Bernoulli equation for incompressible fluids:

$$\frac{w_2^2 - w_1^2}{2} = \frac{p_1 - p_2}{\rho}$$

or with $\rho = \gamma/g$ and $p = \gamma h$,

$$\frac{w_2^2 - w_1^2}{2g} = \frac{p_1 - p_2}{\gamma} = h_1 - h_2$$

In this paper, however, we are interested in high-velocity flow of gases, where the changes in density due to pressure differences cannot be neglected. In order to integrate [3] it is therefore necessary to make a particular assumption regarding the manner in which density depends on pressure. The flow of gases in all cases considered in this paper being at very high velocities, we can assume that all changes of state of the gas are adiabatic, or

$$pv^k = \text{const} \quad \ldots\ldots\ldots\ldots\ldots\ldots [4]$$

where $k = c_p/c_v$ depends on the kind of gas, and to a certain extent on its pressure and other conditions. For dry air and normal pressure, $k = 1.405$.

Substituting $\rho = \gamma/g = 1/vg$ in [3],

$$\frac{w_2^2 - w_1^2}{2g} = \int_1^2 v dp$$

and since

$$pv^k = p_1 v_1^k$$

or

$$v = p_1^{1/k} v_1 \times \frac{1}{p^{1/k}}$$

we have

$$\frac{w_2^2 - w_1^2}{2g} = p_1^{1/k} v_1 \int_2^1 \frac{dp}{p^{1/k}}$$

Integrated, this expression becomes

$$\frac{w_2^2 - w_1^2}{2g} = \frac{k}{k-1} p_1 v_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right] \quad \ldots\ldots [5]$$

Equation [5] gives the relation between velocity and pressure along a streamline for a gas in the same way that Bernoulli's equation describes the motion of an incompressible fluid. In this connection, however, it must be borne in mind that Bernoulli's equation applies not only to incompressible fluids but also to gases when their compressibilities can be neglected.

As the specific volume v_1 generally cannot be measured very easily, it is advisable to introduce the temperature T_1 (deg. fahr. abs.), using the relation

$$p_1 v_1 = RT_1$$

where for dry air $R = 53.34$ per deg. fahr. (p_1 in lb. per sq. ft. and v_1 in cu. ft. per lb.); thus

$$\frac{w_2^2 - w_1^2}{2g} = \frac{k}{k-1} RT_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right] \quad \ldots\ldots [6]$$

Considering, for instance, the flow of air through a nozzle into a vessel in which a lower pressure p_2 exists (see Fig. 2), we can calculate the velocity for each pressure p_2 in the vessel. Since the initial velocity of flow is zero, we have $w_1 = 0$ and hence, for the velocity which corresponds to the pressure p_2 in the vessel,

$$w_2 = \sqrt{\frac{2gk}{k-1} p_1 v_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]}$$

$$= \sqrt{\frac{2gk}{k-1} RT_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]} \quad \ldots [7]$$

For $p_2 = 0$, i.e., for flow into a perfect vacuum, the maximum velocity will be

$$w_{2\max} = \sqrt{\frac{2gk}{k-1} RT_1}$$

Assuming, for instance, a temperature $T_1 = 523$ deg. abs. equal to about 63 deg. fahr., $k = 1.405$, and $R = 53.34$, the maximum velocity will be 2500 ft. per sec.

In order to obtain a higher velocity, it would be necessary to start with air of a higher temperature. According to [7], the velocity of a gas is the same whether the gas expands from a pressure of five atmospheres to one atmosphere or from one atmosphere to one-fifth of an atmosphere, assuming the same initial temperatures in both cases. Fig. 3 shows graphically the relation between velocity and pressure ratio for an initial temperature of $T_1 = 523$ deg. abs., etc., and $k = 1.405$.

II—Importance of Acoustic Velocity

In order to give a clear idea of the fundamental importance of acoustic velocity to the flow of gases at high velocities, let us consider the weight per second (W) flowing through the nozzle shown in Fig. 2. From considerations of continuity we see that the weight per second has to be the same for each cross-section A of this nozzle. Thus

$$W = Aw\gamma = A \frac{w}{v} = \text{const} \quad \ldots\ldots\ldots\ldots [8]$$

Assume that the pressure p_1 may be constant (atmospheric pressure) and that the pressure p can be changed. The product $w\gamma$ as a function of the pressure ratio p/p_1 can be easily obtained from Fig. 3 by multiplying together the two curves

$$w = f\left(\frac{p}{p_1}\right) \quad \text{and} \quad \gamma = \gamma_1 \left(\frac{p}{p_1}\right)^{1/k}$$

(adiabatic condition). The curve in Fig. 4 has been obtained in this way and we see from it that the value $w\gamma$ is zero for $p = 0$ and for $p = p_1$. This is due to the fact that for $p = 0$ (absolute vacuum) the specific weight γ is zero and that for $p = p_1$ (no

pressure difference) the velocity w is zero. Consequently $w\gamma$ has a maximum value for a certain intermediate value of p/p_1. This value can be easily determined by multiplying [7] by the specific weight $\gamma = 1/v$ and then obtaining the derivative of this expression with respect to p/p_1. Equating this derivative to zero gives the value of p/p_1 for which $w\gamma$ is a maximum. Thus

$$w\gamma = \frac{w}{v} = \sqrt{\frac{2gk}{k-1}\ \frac{p_1 v_1}{v^2}\left[1 - \left(\frac{p}{p_1}\right)^{\frac{k-1}{k}}\right]}$$

or, remembering that

$$pv^k = p_1 v_1^k, \quad \text{or} \quad \frac{1}{v^2} = \left(\frac{p}{p_1}\right)^{2/k}\frac{1}{v_1^2}$$

then

$$w\gamma = \sqrt{\frac{2gk}{k-1}\ \frac{p_1}{v_1}\left[\left(\frac{p}{p_1}\right)^{2/k} - \left(\frac{p}{p_1}\right)^{\frac{k+1}{k}}\right]} \quad \ldots\ldots [9]$$

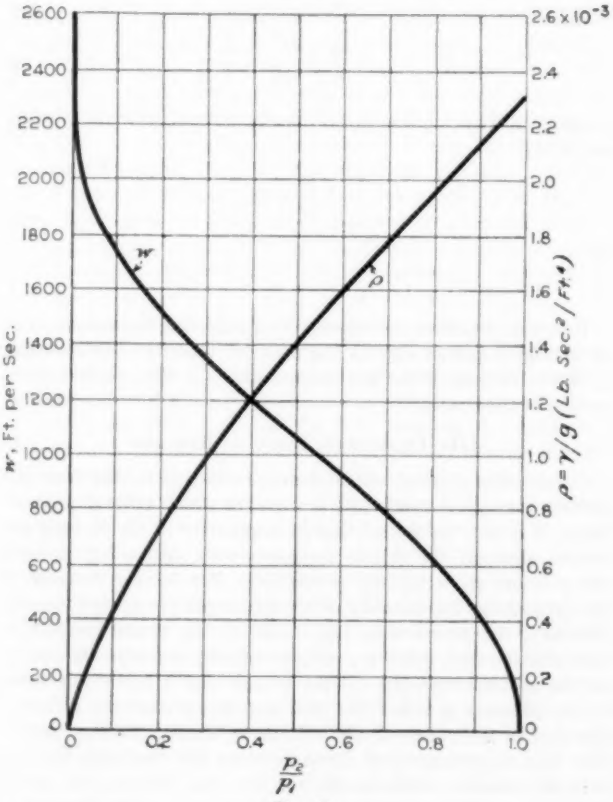


FIG. 3

Also, since

$$\frac{p_1}{v_1} = \frac{p_1^2}{p_1 v_1} = \frac{p_1^2}{RT_1}$$

$$w\gamma = p_1\sqrt{\frac{2gk}{k-1}\ \frac{1}{RT_1}\left[\left(\frac{p}{p_1}\right)^{2/k} - \left(\frac{p}{p_1}\right)^{\frac{k+1}{k}}\right]} \quad \ldots\ldots [9a]$$

According to the [9a], $w\gamma$ is a maximum if $\left[(p/p_1)^{2/k} - (p/p_1)^{\frac{k+1}{k}}\right]$ is a maximum, and so it is sufficient to put the derivative of this expression equal to zero, or

$$\frac{\partial\left[\left(\frac{p}{p_1}\right)^{2/k} - \left(\frac{p}{p_1}\right)^{\frac{k+1}{k}}\right]}{\partial\left(\frac{p}{p_1}\right)} = 0$$

Performing this differentiation and solving the equation for p/p_1, we get

$$\frac{p}{p_1} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} = 0.528 \text{ for, } k = 1.405$$

For this pressure ratio the product $w\gamma$, and therefore, according to [8], also W/A (i.e., the weight of gas per unit of cross-sectional area), becomes a maximum.

Considering now especially the flow in the smallest cross-

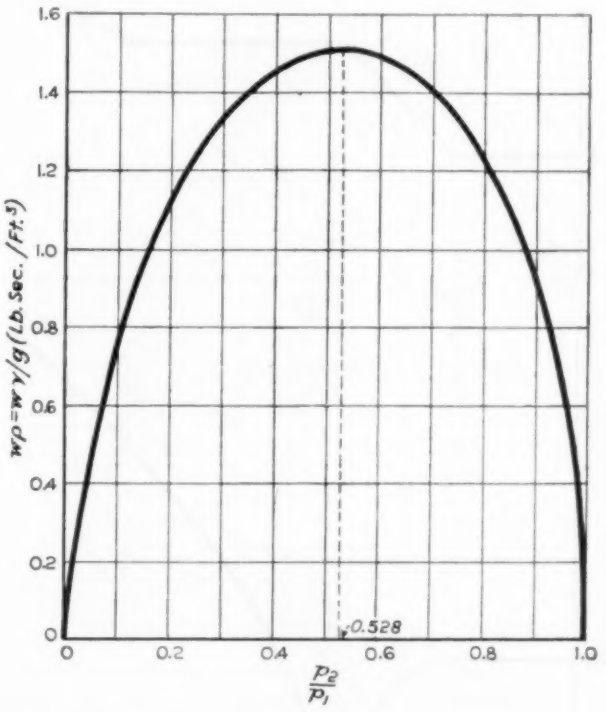


FIG. 4

section A' of the nozzle, let us gradually decrease the pressure p in the vessel (noted in Fig. 2 as p_2). According to Equation [9] and to Fig. 4, the weight of gas flowing per second through the nozzle, i.e., $W = A'w\gamma$, will increase until the pressure p is equal to $0.528p_1$. If, however, we decrease the pressure p in the vessel still further, getting thus pressure ratios smaller than 0.528, the flow of gas per second cannot decrease as indicated by [9] and Fig. 4, but will maintain a constant value corresponding to the maximum value of $w\gamma$. This implies that the velocity in the cross-section A' cannot exceed a certain critical value which corresponds to the pressure ratio $p/p_1 = 0.528$. For if the velocity in A' could exceed this critical value, the pressure at A' would have to be less than $0.528\ p_1$. This would mean that at some cross-section A'' reached before A' and larger or at least equal to A', the pressure $0.528\ p_1$ itself would be established. Since for this critical pressure $w\gamma$ is a maximum, this would imply that more fluid passes through section A'' than through A', which is impossible.

We can now easily demonstrate that this critical velocity is exactly the acoustic velocity. The pressure p_1 being kept constant, $w\gamma$ depends only on p, and as the critical velocity occurs for the maximum of $w\gamma$ we have to write the derivative of $w\gamma$ with respect to p and put this expression equal to zero:

$$\frac{d(w\gamma)}{dp} = w\frac{d\gamma}{dp} + \gamma\frac{dw}{dp} = 0$$

or, since $\gamma = \rho g$,

$$w \frac{d\rho}{dp} + \rho \frac{dw}{dp} = 0 \quad \ldots\ldots\ldots\ldots\ldots\ldots [10]$$

For the second term of [10], we have, according to [2],

$$\rho \frac{dw}{dp} = \frac{1}{w} \quad \ldots\ldots\ldots\ldots\ldots\ldots [11]$$

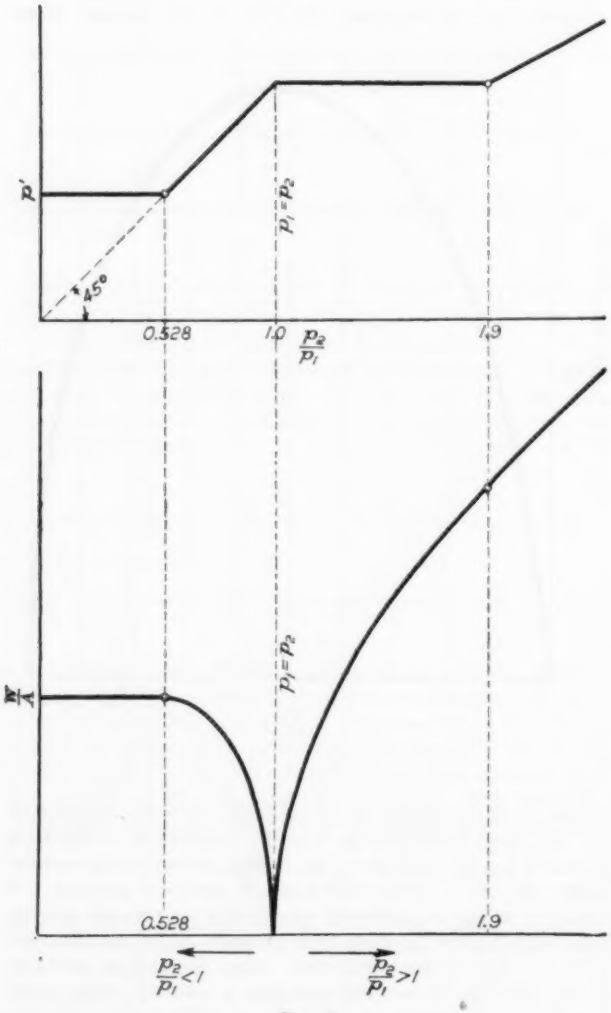


FIG. 5

The first term can be rewritten as

$$\frac{d\rho}{dp} = \frac{\rho}{\frac{dp}{d\rho}\rho}$$

and this can be reduced to the form

$$\frac{d\rho}{dp} = \frac{\rho}{-\frac{dp}{\frac{1}{v}dv}}$$

by virtue of the relations

$$\rho v = \frac{1}{g} = \text{const.}$$

$$\rho \frac{dv}{dp} + v \frac{d\rho}{dp} = 0$$

and

$$\rho \frac{dp}{d\rho} = -\frac{dp}{\frac{dv}{v}}$$

But

$$\frac{dp}{-\frac{dv}{v}} = E$$

where E is the modulus of elasticity, consequently

$$\frac{d\rho}{dp} = \frac{\rho}{E} = \frac{1}{E/\rho} \quad \ldots\ldots\ldots\ldots\ldots\ldots [12]$$

Furthermore the acoustic velocity c is equal to $\sqrt{E/\rho}$, so that

$$\frac{d\rho}{dp} = \frac{1}{c^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots [13]$$

Finally, substituting this expression for $d\rho/dp$ in [10] and making use of [11],

$$\frac{1}{w} - \frac{w}{c^2} = 0$$

or

$$w = c$$

It is thus seen that the velocity of a gas flowing through a nozzle as in Fig. 2 cannot exceed the acoustic velocity corresponding to the conditions of the gas (temperature) in the smallest cross-section of the nozzle.

III—Discharge Through Nozzles

Considering a gas at atmospheric pressure p_1 flowing from the outside through a nozzle into a vessel in which a lower pressure exists (Fig. 2), we found that the quantity of air flowing per second through the nozzle increases with decreasing p_2 until the pressure $p_2 = 0.528 p_1$ is reached. For further decrease of the pressure p_2 the quantity of air discharged per second remains constant, the pressure at the throat of the nozzle remains at the critical value, 0.528 p_1, and the velocity corresponds exactly to the acoustic velocity. Thus we see that a further decrease of the pressure p_2 below the critical pressure does not influence the flow in the nozzle at all. However, due to the fact that in this case a pressure drop exists between the discharge opening and the interior of the vessel, the gas after leaving the nozzle will be accelerated above the acoustic velocity. This is possible since the gas jet now has more room for expansion and can leave the nozzle in diverging streamlines.

Considering now the flow of gas discharging from a vessel of high pressure (p_2) into the atmosphere (p_1), the formulas previously given will hold in case we take account of the fact that the higher pressure is denoted by p_2 and is generally variable, while the lower pressure is denoted by p_1 and is considered as constant (atmospheric pressure). Therefore, for example, in [7] we must write p_1/p_2 instead of p_2/p_1. Increase of the pressure p_2 in the vessel corresponds to an increase of the velocity until the critical pressure ratio $p_2/p_1 = 1/0.528 = 1.895$ is reached and the acoustic velocity in the smallest cross-section of the nozzle is obtained. Up to this critical pressure ratio the pressure at the discharge opening of the nozzle is equal to p_1; with further increase of the pressure in the vessel, however, the pressure and therefore also the specific weight at the dis-

charge opening will increase proportionally to p_2. Due to this fact we have in the case of a pressure $p_2 > 1.895p_1$ an increase of the weight of gas proportional to the increase of p_2. In Fig. 5 it is shown in what manner the pressure in the discharge opening and the weight per second per square foot depends on the pressure ratio. Here $p_2/p_1 < 1$ means flow from the atmosphere into a vessel of lower pressure, and $p_2/p_1 > 1$ means flow from a vessel in which a higher pressure exists into the atmosphere.

IV—Flow Through a De Laval Nozzle

(*a*) *Flow Without Loss of Mechanical Energy.* Let us now investigate the flow in detail within the nozzle itself, assuming at first, as was done earlier, that the flow is without any losses due to friction or other causes. As an example, we shall consider the flow through a so-called De Laval nozzle as shown in Fig. 6(*a*).

It is desired to determine the pressure at different points along the nozzle or in the x direction as in Fig. 6. As the weight of gas W flowing per second through the nozzle must be the same for each cross-section A of the nozzle for reasons of continuity, we can, for a given W and for a given nozzle shape, consider the ratio W/A as a known function of x. Fig. 6(*b*) shows this function for the nozzle shape of Fig. 6(*a*) and for three values of W. On the other hand, the ratio W/A can be determined by using Equation [9a] or from Fig. 4 or Fig. 6(*c*). Making use of the two figures, 6(*b*) and 6(*c*), the variation of pressure along the nozzle can easily be determined for a certain value of W, i.e., a certain curve W/A in Fig. 6(*b*). It is only necessary to determine in Fig. 6(*b*) for different points of the x-axis the corresponding W/A and then to obtain from Fig. 6(*c*) the p/p_1 value, corresponding thereto. Fig. 6(*d*) has been constructed in this way, and shows the variation of pressure along the nozzle for three different weights per second, corresponding to three different pressures p_1 inside the vessel.

The upper W/A curve in Fig. 6(*b*) gives at the same time the curve of the maximum weight per second flowing through the nozzle since the maximum of $W/A = w\gamma$ has been reached at the smallest cross-section. This corresponds to a certain pressure p_I at the discharge opening of the nozzle. For this pressure p_I in the vessel we have the critical pressure in the smallest cross-section, and therefore the acoustic velocity. Nearer the discharge opening the velocity decreases with increasing pressure. In order to obtain in the nozzle a velocity higher than the acoustic velocity it is necessary to have a much lower pressure in the vessel. How low this pressure has to be in order to enable the gas to expand sufficiently depends on the shape of the nozzle. For the nozzle shown in Fig. 6(*a*) we see in 6(*d*) that a very low pressure p_{II} is necessary to fulfil this condition. For a nozzle diverging less than that in 6(*a*), the pressure would not need to be as low as p_{II}. For a nozzle of given shape, therefore, there is but one pressure at the discharge opening that will allow the gas to expand with a velocity above the acoustic velocity. Of course, this holds only as long as the assumption that the process can be considered adiabatic obtains, and that there are no losses due to friction or other causes. The question now naturally arises as to what does happen if the pressure in the vessel lies between p_I and p_{II}. In this case flow without losses is impossible, and only a flow in which part of the mechanical energy is converted into heat can be obtained.

(*b*) *Flow Through Nozzles With Compression Shocks.* Experiments made by Stodola and by Buechner show that where the pressure in the vessel is between p_I and p_{II}, the gas flowing through the smallest cross-section at first expands at a velocity above the acoustic velocity, and then, at some point in the diverging part of the nozzle, is suddenly compressed, suffering a considerable loss in velocity. The loss of kinetic energy occasioned by this sudden compression is much larger than the gain in pressure energy, indicating that part of the kinetic energy has been converted into heat energy.

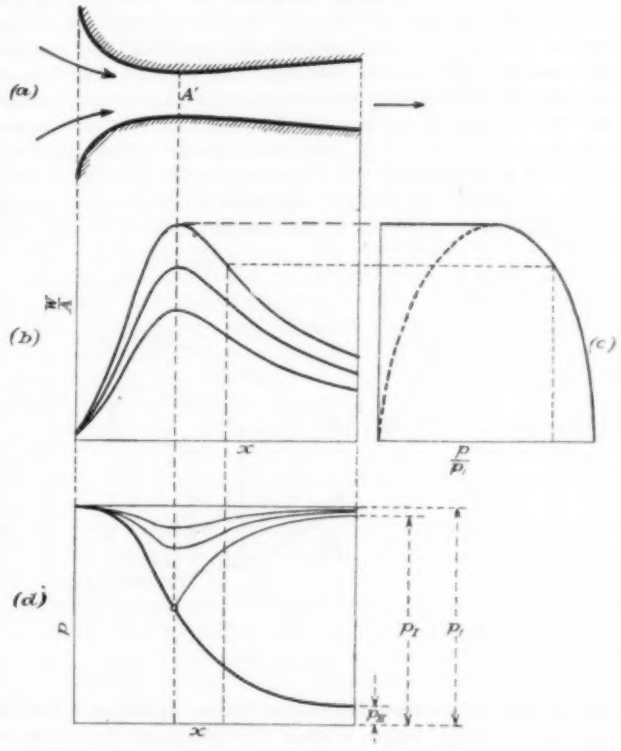


Fig. 6

Without going into the details of these so-called compression shocks now, we shall merely state that this strange behavior of gas flowing with velocities exceeding the acoustic velocity is indicated qualitatively in Fig. 7. The velocity decreases abruptly[3] to a lower value, while the pressure increase is also abrupt and like a shock. As mentioned before, the pressure increase is smaller than it would be under reversible adiabatic conditions, i.e., smaller than the value given by Equation [3]. This difference in pressure energy is just that part of the mechanical energy which is converted after the shock into heat energy.

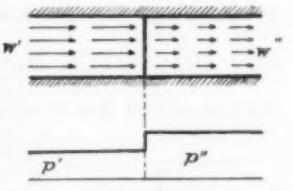


Fig. 7

Having constructed in Fig. 6(*d*) the curve of pressure variation along the nozzle, taking into account that according to [3] the mechanical energy, i.e., the sum of kinetic energy and pressure energy, has to be constant, we have now to construct the curves corresponding to pressures in the vessel between p_I and p_{II} for which the total energy, including the heat energy, is constant. In Fig. 8 the heavy line is the course of pressure variation for adiabatic expansion for the given shape of the nozzle. The three dotted lines are curves of constant total energy for three different pressures at the discharge opening. (The formulas by which those curves can be calculated will be given later.) Calculating now for each point of the expansion curve—but only

[3] Taking into account the conduction of heat, L. Prandtl has shown that this change of velocity and pressure takes place along a length of about $^1/_{20,000}$ in.

for the diverging part of the nozzle—the pressure increase caused by a compression shock and then connecting the resulting points of increased pressure by a dash-and-dot line as in Fig. 8, we obtain a curve which intersects the curves of total energy. The pressure variation along the nozzle for a pressure p in the vessel proceeds now as follows: Up to the smallest cross-section the gas expands adiabatically, reaching the critical pressure and the acoustic velocity. The pressure then follows further the adiabatic-expansion curve according to the increase of the velocity above the acoustic velocity. At the cross-section A_0 the pressure p' increases suddenly to the value p'', involving a considerable decrease of velocity below the acoustic velocity; finally the pressure increases slowly from p'' to p_0 along the curve of constant total energy. The fact that this

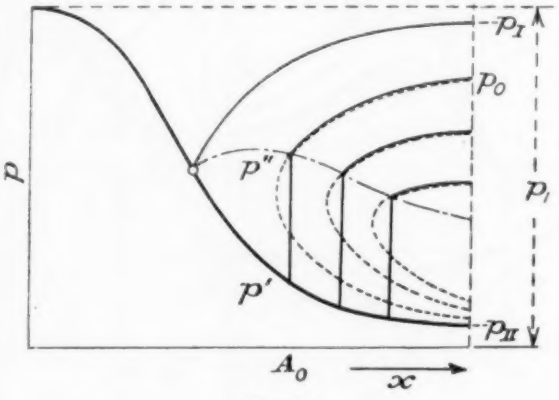


FIG. 8

sort of flow is always accompanied by an appreciable loss of mechanical energy makes it clear that Equation [3], which is based on the constancy of the mechanical energy, cannot be adequate.

In order to obtain an equation which holds also for those cases where portions of the mechanical energy are converted into heat, it is necessary to relate Equation [2] with the heat equation applied to a gas particle moving along a streamline. If u represents the internal energy per pound of the gas and dq the heat supplied, both measured in mechanical units, then

$$dq = du + pdv$$

Or, if we assume that no heat is supplied from outside,

$$du + pdv = 0$$

If we now substitute vg in [2] for $1/\rho$,

$$\frac{w}{g}\,dw + v\,dp = 0$$

Thus

$$du + pdv + vdp + \frac{w}{g}\,dw = 0$$

or

$$du + d(pv) + d\frac{w^2}{2g} = 0$$

which, integrated along a streamline from 1 to 2, gives

$$\frac{w_2^2 - w_1^2}{2g} = u_1 + p_1v_1 - (u_2 + p_2v_2)$$

Assuming the behavior of a permanent gas and considering that

$$u = \frac{1}{k-1}\,pv \quad \text{and} \quad u + pv = \frac{k}{k-1}\,pv$$

we have

$$\frac{w_2^2 - w_1^2}{2g} = \frac{k}{k-1}\,(p_1v_1 - p_2v_2) \ldots\ldots\ldots [14]$$

Eliminating v_1 and v_2 through the continuity equation: namely,

$$W = A_1\frac{w_1}{v_1} = A_2\frac{w_2}{v_2}$$

we obtain

$$p_1w_1 - p_2w_2 = W\frac{k-1}{2gk}\left(\frac{w_2^2}{A_2} - \frac{w_1^2}{A_1}\right)$$

By means of this relation and the assumption of constant entropy for each curve, it is possible to construct the dotted curves of Fig. 8.

Compression shocks can only occur if the velocity is higher than the acoustic velocity and the decreased velocity after the shock is always lower than the acoustic velocity. A remarkable deduction from these considerations is the fact that the product of the velocities before and after the shock is equal to the square of the acoustic velocity. Denoting the velocity and the pressure before the shock by w' and p', respectively, and after the shock by w'' and p'', we may say first of all that the time rate of change of momentum must be equal to the forces acting from outside on the system, and thus that

$$\frac{W}{g}\,(w' - w'') = (p'' - p')A$$

Substituting for W first $\frac{Aw''}{v''}$ and then $\frac{Aw'}{v'}$ according to the continuity equation

$$W = A\frac{w''}{v''} = A\frac{w'}{v'}$$

we obtain

$$\frac{w'w''}{g}\left(\frac{1}{v''} - \frac{1}{v'}\right) = p'' - p' \ldots\ldots\ldots\ldots [15]$$

Now making use of [14], we have, in case the flow starts with zero velocity ($w_1 = 0$),

$$\frac{w'^2}{2g} = \frac{k}{k-1}\,(p_1v_1 - p'v')$$

and

$$\frac{w''^2}{2g} = \frac{k}{k-1}\,(p_1v_1 - p''v'')$$

or, subtracting one equation from the other,

$$p'' - p' = p_1v_1\left(\frac{1}{v''} - \frac{1}{v'}\right) - \frac{k-1}{2gk}\left(\frac{w''^2}{v''} - \frac{w'^2}{v'}\right)$$

We may now use the continuity equation in the form

$$\frac{w''}{v''} = \frac{w'}{v'}$$

since the areas before and after shock are sensibly equal, and get

$$p'' - p' = p_1v_1\left(\frac{1}{v''} - \frac{1}{v'}\right) + \frac{k-1}{k}\,\frac{w'w''}{2g}\left(\frac{1}{v''} - \frac{1}{v'}\right)$$

Also by means of [15] we may eliminate $p'' - p'$ and drop the common factor $\left(\frac{1}{v''} - \frac{1}{v'}\right)$ as follows:

$$\frac{w'w''}{g} = p_1v_1 + \frac{k-1}{k}\,\frac{w'w''}{2g}$$

or simply

$$w'w'' = 2g\,\frac{k}{k+1}\,p_1v_1$$

Returning to Equation [7], substituting therein the critical pressure ratio $\left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}$ for the ratio $\frac{p_2}{p_1}$ and then simplifying, it is easily seen that the right-hand member of this last expression is the square of the acoustic velocity. Thus

$$w'w'' = c^2$$

Considered from a purely mathematical standpoint, the above expression includes the possibility of an expansion shock as well as a compression shock. Such an expansion shock, however, would involve a decrease of entropy instead of an increase, and so would be physically impossible.

Discussion

J. M. Spitzglass.[4] The author presents a vivid demonstration of the following interesting facts about the so-called critical velocity of gases:

First, that the critical, or highest obtainable, velocity of a gas at the smallest section of a nozzle is the same as the velocity of sound through the medium at that section.

Second, that for true adiabatic expansion this critical velocity is reached at a corresponding critical-pressure ratio of

$$\frac{p_2}{p_1} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}$$

which is equal to 0.528 for $K = 1.405$.

Third, that both the critical pressure and the acoustic velocity at the smallest cross-section of a nozzle are maintained the same irrespective of any further pressure reduction on the divergent side of the nozzle.

Fourth, that the critical velocity can be exceeded only on the divergent side of the nozzle if the pressure p_2 is reduced much below the above critical pressure.

Fifth, that the maximum mass rate of flow per unit area is reached at the critical velocity of the gas, and this rate is maintained irrespective of the pressure or velocity changes on the divergent side of the nozzle.

Sixth, that an increase of the initial pressure p_1 after the critical limit is reached increases the critical pressure in the smallest section, maintaining the same ratio of p_2/p_1, and while the critical velocity is still the same as the acoustic velocity, yet the increased density gives a larger rate of mass flow in proportion to the increase of p_1 or of p_2, which increases in the same ratio.

The analysis given in the paper is based upon the usual assumption of adiabatic expansion of the gas and the friction losses neglected. It should be emphasized that Figs. 3 and 4, which are extremely illuminating, are in fact solutions of Equation [7] for the range from zero drop to zero pressure. The part that may actually be obtained in a nozzle is only the first half up to the critical point of maximum flow. As the author shows in Fig. 6, it requires a drop of pressure much below the critical pressure in the nozzle before velocities are increased over the acoustic velocity of the fluid.

The writer would like to ask the author to give an explanation of how in any case velocities are obtained as high as twenty times the acoustic velocity of the fluid.

Water-gas manufacturers have for a long time used nozzles above the critical flow for the supply of steam to gas generators, determining the quantity of flow by the so-called Napier's rule, $W = pA/70$.

Some eighteen years ago the writer had a chance to check this rule at one of the large gas stations in Chicago. There a 6-in. live-steam pipe carrying a pressure of 150 lb. supplied the steam to two gas generators. Each generator had a nozzle of the same diameter in the corresponding branch from the main line. An impact tube was inserted in the main line to indicate the flow of the high-pressure steam in the pipe. Running with two gas generators, one at a time, the differential on the impact tube was 20 per cent higher on one of the machines than on the other. This repeated itself invariably. The nozzles were carefully checked; the lines appeared to be approximately the same, except that one was about ten or fifteen feet farther away than the other, but this did not explain the difference in the flow. Could the author tell us what his experience has been in duplicating the maximum flow at practical applications of nozzles?

It is of interest to apply the author's analysis to the case of the square-edged-orifice plate. In that case, of cou se, there is no apparent reason for a critical, or limiting, velocity either at the section of the orifice or at the *vena contracta* because there is no rigid continuity between any of the sections. That is, in the case of the square-edged orifice, the exit velocity may assume a value determined by an expression similar to Equation [7], or at least a value proportional to it.

Following the demonstration of Figs. 3 and 4, there should be a similar set of curves for the *actual* rate of flow through the orifice, because the velocity is free to increase with the drop of pressure, and the density is free to decrease with the final pressure, and to approach zero as a limit when p_2 approaches zero.

From the experiences that have been published on the subject, the opposite is shown to be the case. John L. Hodgson,[5] for instance, has shown actual test data on orifice plates down to the value of $p_2/p_1 = 0.10$, where the mass rate of flow actually increased with the drop of pressure and by extension the curve indicates a maximum rate of flow at $p_2 = 0$.

The pertinent question is how can there be a definite mass flow with a condition of zero density of the fluid. On the other hand, it is just as hard to conceive of a definite maximum velocity of the fluid at a zero rate of mass flow. Mr. Hodgson's experiments leave no doubt that the mass flow was not diminished at the lower pressure limit. We must conclude, therefore, that after the critical velocity, the density in the streamline is not wholly dependent on the pressure in the space surrounding the fluid stream. This should be equally as true for shaped nozzles as it is for square-edged orifices. In other words, Equation [7] does not seem to apply after the critical velocity is reached in the nozzle.

H. S. Bean.[6] This paper is a timely contribution to the literature on fluid flows. As the author has indicated, most engineers have but a hazy idea of the physical relations involved in gas flows at high velocities. It is a subject that might well be given

[4] Mechanical Engineer and Vice-President, Republic Flow Meters Co., Chicago, Ill. Mem. A.S.M.E.

[5] "The Laws of Similarity for Orifice and Nozzle Flows." Trans. A.S.M.E., vol. 51 (1929), paper no. FSP-51-42-103.

[6] Physicist, Bureau of Standards, Washington, D. C. Mem. A.S.M.E.

much more attention by many of our engineering schools. The importance of gas flows at velocities in the region of the acoustic velocity is not limited to problems of turbine and engine design and ballistics, but also extends to the field of gas measurement. In this work the so-called "critical-flow orifice" is being used both as a limiting demand meter and as a reference meter. The subject has also to be considered in some cases in the design of airfoils.

The phenomenon of gas flows at high velocities through nozzles, either expanding or non-expanding nozzles, is by no means so simple as would appear from the author's analysis. However, the several assumptions which were made are necessary for making a simple analysis of the flow. It would be very difficult, if not impossible, to construct equations to represent the actual flow conditions, for they are so complicated that it is doubtful if any one knows exactly what does take place. A view of a jet of gas issuing from a nozzle at a velocity greater than the acoustic velocity will show various bands across the jet. These bands probably indicate regions of relatively higher or lower density.

It is common practice to assume that the static pressure is constant over any right section of a gas stream flowing along a closed channel such as a pipe or nozzle. Recent experiments by Dr. Prandtl and others have shown that this is not the case. To be sure, the variations are generally small and may be neglected in most engineering computations.

In the paper ρ has been used to represent density expressed in absolute units, while γ is used to represent density expressed in engineering units, often called "specific weight." To some readers it might have been clearer to have used one system of units throughout, and thus to have used but one of these symbols. Also, in section III it might have been clearer to some if p_1 were taken as always representing the pressure at the first or upstream section, and p_2 as representing the pressure at the second or downstream section.

J. M. Naiman.[7] The author has clarified the important distinction between the critical rate of mass flow per unit area W/A, and of linear velocity, w. The former is the one which reaches its maximum value in the minimum cross-section of the flow streamlines, i.e., in the *vena contracta* of the streamlines. The latter, on the other hand, merely reaches in the *vena contracta* the "critical" value of "acoustic velocity," which is not at all its maximum value, since the velocity increases beyond the minimum area or *vena contracta* point, i.e., beyond the throat of the nozzle described, if the pressure beyond this point is less than 52.8 per cent of the initial pressure ahead of the nozzle. The only reason why the mass flow per unit area, W/A, does not increase beyond the nozzle throat from its maximum value at the throat or *vena contracta*, even though the gas velocity w increases above "acoustic velocity," is that the rate of decrease in gas density γ is faster than the rate of increase in gas velocity w. This is very aptly brought out by the two curves of Fig. 3, from which one can see why the product of the two, i.e., $w\gamma$, upon which W/A, the mass flow per unit area, depends, actually decreases beyond the *vena contracta*, in spite of increase of the gas velocity beyond acoustic velocity; i.e., beyond the *vena contracta*, W/A decreases, but the increase in area A keeps W constant.

The effect of temperature changes when the gas velocity exceeds "acoustic velocity," as evidenced by the so-called compression shocks, with resulting decrease of the gas velocity from w_1 (which is more than the acoustic velocity, c) to w_2 (which is less than c), with c being the geometric mean between w_1 and w_2, is very interesting, and should be of value in the design of the outlet end of nozzles, in such a way that the outlet pressure is not too low, and yet the velocity beyond the nozzle throat is above acoustic velocity, if the latter is desirable.

Considering the value of the points brought out by the author it would help to popularize the subject if the mathematical development of the basic equations were more concise and less laborious.

For example, the relation between the change in the kinetic and the change in the potential energy of a gas is as well known as the more basic relation between force and acceleration, and half of the first two pages could be saved by starting with the sufficiently basic relation,

$$\frac{w_2^2 - w_1^2}{2g} = \int v dp \quad \ldots\ldots\ldots\ldots\ldots [1]$$

with the remark that this relation omits the negligible factor of gravity, important only in meteorology, and the effect of friction.

Similarly, the demonstration made by the author from the bottom of page 3 to the lower part of page 4, that the velocity in the *vena contracta* (for the case when p/p_1 is equal to or less than 0.528) is equal to the acoustic velocity c at the pressure and temperature of the gas in the *vena contracta*, could be simplified as follows:

The velocity in the *vena contracta* may be calculated from the author's Equation [7] by noting that the pressure ratio for the case under consideration is, as shown at the top of page 3,

$$\frac{p}{p_1} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}$$

so that the term $\left(\frac{p}{p_1}\right)^{\frac{k-1}{k}}$ in Equation [7] becomes $\left(\frac{2}{k+1}\right)$, giving,

$$w = \sqrt{\frac{2gk}{k-1} RT \left(1 - \frac{2}{k-1}\right)} = \sqrt{\frac{2gk}{k+1} RT} \text{ (ft. per sec.)} \quad \ldots\ldots [2][8]$$

On the other hand, the acoustic velocity at the pressure p and the temperature T in the *vena contracta* is

$$c = \sqrt{\frac{kp}{\rho}} = \sqrt{\frac{kpg}{\gamma}} = \sqrt{kpgv} = \sqrt{kgRT} \quad \ldots\ldots [3]$$

To express this in terms of the initial gas temperature T_1, we note from the gas laws that

$$pv = RT, \quad p_1v_1 = RT_1, \quad \text{and} \quad pv^k = p_1v_1^k$$

from which we may write,

$$\frac{RT}{RT_1} = \frac{pv}{p_1v_1} = \frac{p}{p_1}\left(\frac{p_1}{p}\right)^{\frac{1}{k}} = \left(\frac{p}{p_1}\right)\left(\frac{p}{p_1}\right)^{-\frac{1}{k}} = \left(\frac{p}{p_1}\right)^{1-\frac{1}{k}} = \left(\frac{p}{p_1}\right)^{\frac{k-1}{k}} \quad \ldots\ldots [4]$$

[7] Chicago, Ill. Assoc-Mem. A.S.M.E.

[8] Compare with the value of the maximum velocity, possibly, w_{max} which as shown by the author is $w_{max} = \sqrt{\frac{2gk}{k-1} RT_1}$ (ft. per sec.), or $\sqrt{\frac{k+1}{k-1}} = 2.437$ times greater than the "critical velocity." Thus, for $T_1 = 523$ deg. abs. (63 deg. fahr.), $w_{max} = 2500$ ft. per sec. while $w = 2500/2.437 = 0.410 \times 2500 = 1025$ ft. per sec. only.

But for the case under consideration,

$$\frac{p}{p_1} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}$$

Therefore Equation [4] may be simplified to

$$RT = RT_1\left(\frac{p}{p_1}\right)^{\frac{k-1}{k}} = RT_1\left(\frac{2}{k+1}\right) \ldots\ldots [4a]$$

and Equation [3] for acoustic velocity in the *vena contracta* becomes

$$c = \sqrt{kgRT} = \sqrt{\frac{2kg}{k+1}RT_1} \ldots\ldots [3a]$$

which, when compared with Equation [2] for the gas velocity w through the *vena contracta*, shows it to be equal to the acoustic velocity c through the nozzle throat; i.e.,

$$w = c \ldots\ldots [5]$$

At this point it is interesting to note the simplified equation for the flow rate through the *vena contracta* in pounds per second, when the pressure ratio p/p_1 is equal to or less than $\left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} = 0.528$. Thus, we have from Equation [2], remembering the adiabatic equation $pv^k = p_1v_1^k$, the basic gas law $p_1v_1 = RT_1$ and the relation $\gamma = \frac{1}{v}$,

$$W = Aw\gamma = A\sqrt{\frac{2gk}{k+1}RT_1}\left(\frac{1}{v}\right) = A\sqrt{\frac{2gk}{k+1}RT_1}\left(\frac{p}{p_1}\right)^{\frac{1}{k}}$$

$$\left(\frac{1}{v_1}\right) = A\sqrt{\frac{2gk}{k+1}RT_1}\left(\frac{2}{k+1}\right)^{\frac{1}{k-1}}\left(\frac{p_1}{RT_1}\right) = \frac{0.5\,Ap_1}{\sqrt{T_1}}$$

(lb. per sec.)..... [6]

where W = air-flow rate, lb. per sec.
A = *vena contracta* area, sq. ft.
p_1 = initial air pressure, lb. per sq. ft. abs., and
T_1 = initial air temperature, deg. fahr. abs.

$$\left(\frac{2}{k+1}\right)^{\frac{1}{k-1}}\sqrt{\frac{2gk}{(k+1)R}} = 0.5318$$

For a gas whose specific gravity is S, since the gas law becomes $p_1v_1S = RT_1$ (where $R = 53.34$), R/S should be substituted for R into the Equation [6], which then becomes

$$W = \frac{0.5318\,Ap_1}{\sqrt{ST_1}} \text{ (lb. per sec.)} \ldots\ldots [6a]$$

A research program is now being carried on under the writer's supervision in the laboratories of one of his clients, which is to determine how the effect of friction modifies the last equation, and results so far secured promise to form the basis of valuable progress in the field of the flow of fluids. The results of these tests will, in due time, be published by the writer.

In Equation [1], it is confusing to use the symbol F in the right-hand member of the equation to represent force, and in the left-hand member to represent cross-sectional area.

Author's Closure

In order to answer the question by Mr. Spitzglass relative to the possibility of velocities as high as twenty times the acoustic, the author would like to refer to occurrences such as detonation. It has been shown by different authors[9] that the velocities of the propagation of the pressure wave as well as the velocities of the gases themselves may enormously exceed the acoustic velocity. In the case of detonation of oxy-hydrogen gas, velocities up to 8600 ft. per sec. can be obtained, while in the case of trinitrotoluol, velocities as high as 18,500 ft. per sec. may occur for such cases. The pressure differences and changes of temperature are so large that some of the assumptions, on which the calculations given in the paper are based, no longer hold true; so that the formulas given there do not hold.

To explain why the differential on the impact tube could be 20 per cent higher on one of the machines than on the other, as observed by Mr. Spitzglass, it would be necessary to have more details, especially with regard to the shape of the pipe lines and the velocities of the steam.

Concerning the application of the formula developed in the paper to flow through a square-edged orifice, the author does not agree with Mr. Spitzglass that there should in this case be no critical or limiting velocity either at the section of the orifice or at the *vena contracta*. First of all, there is only a *vena contracta* when the pressure ratio is larger than the critical ratio, the velocity thus being smaller than the acoustic velocity. For pressure ratios smaller than the critical, there is no *vena contracta* at all. In this case, the pressure decreases from atmospheric pressure, p_1, to a value given by $p' = 0.528\,p_1$ in the section of the orifice, with a corresponding increase of the velocity up to the acoustic velocity. In proceeding outward from this section, a further decrease of the pressure takes place down to a value p_2, accompanied by a corresponding increase of the velocity beyond the acoustic velocity, and forming thereby a diverging flow. The fact that the flow is diverging indicates that the pressure in this diverging part is not the same as that in the surrounding gas. It should always be kept in mind that to pass a certain quantity of gas per unit time the smallest area is required if the gas is flowing at the acoustic velocity, while a larger area is required to pass the same quantity of gas at a velocity exceeding the acoustic velocity.

The author agrees with Mr. Bean that the phenomenon of flow of gases exceeding the acoustic velocity is not so simple as would appear from the analysis given in this paper, should the reader assume that our knowledge about this kind of flow is exhausted by that which is given therein. The author only wished to draw the attention of a larger audience to this interesting and important part of aerodynamics, avoiding any higher mathematics. Mr. Bean says that it would be very difficult, if not impossible, to construct equations to represent the actual flow conditions. His statement is not quite correct, because these equations have been known for a long time:

$$\rho\left(\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z}\right) = -\frac{\partial p}{\partial x} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right) + \frac{1}{3}\mu\frac{\partial}{\partial x}\left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z}\right),$$

and two analogous equations for the y and z components.

However, the integration of this system of differential equations, including the fulfilling of the boundary conditions, is very difficult, if not hopeless. Neglecting the viscosity, as is done in the paper, is in many cases justified. The further assumption made in the paper that the flow can be considered as a one-dimensional phenomenon cannot always be maintained.

[9] Berthelot, *Comptes Rendus*, vol. 93 (1881), p. 18. Chapman, *Phil. Mag.*, vol. 47 (1899), p. 90. Becker, R., *Zeit. f. techn. Phys.*, vol. 3 (1922), p. 152, 249; vol. 8 (1922), p. 321. Rüdenberg, R., *Artillerist. Monatsh.* (1916), p. 237.

If we are interested in details of the flow in nozzles, for instance, in the pressure distribution over a cross-section, or in details of compression shocks and pressure waves occurring in air jets at velocities above acoustic velocities, we have to consider the flow as three-dimensional or sometimes as two-dimensional. In the latter case, Prandtl and Steichen succeeded in solving the corresponding differential equations by introducing Legendre transformations.

In this way Prandtl was able to explain and calculate in detail the standing pressure waves that occur in high-velocity air jets. These standing pressure waves, which are described by Mr. Bean as various bands across the jet, are regions of relatively higher or lower density as he supposes, and therefore can easily be made visible by the so-called Schlieren method.

With respect to the static pressure in pipes, it may be stated that this pressure is constant over the cross-section, provided the latter is constant in the direction of flow, but that the static pressure is not constant over the cross-section in diverging or converging pipes.

Referring to the possibility of shortening the paper somewhat by starting with Equation [3] instead of [1] as suggested by Mr. Naiman, the author is of the opinion that by starting with [1] the development becomes clearer for those who are not very familiar with the subject and who need to know why the various assumptions have to be made. This is particularly necessary since the formulas given later hold only if the velocity can be considered as a function of the coordinates only of the time (stationary flow).

The author also does not agree that the way in which Mr. Naiman develops the relation that the maximum velocity in the smallest cross-section is equal to the acoustic velocity, represents a simplification, but rather that it is another way of deriving the same formula. It always depends on what we assume as being known. If we assume, for instance, that the relation $dp/d\rho = c^2$ can be considered as well known,[10] the desired relation can be deduced directly from Equation [10] in connection with Equation [11], thus using only three equations. Mr. Naiman develops an Equation [6] or [6a] for the rate of flow through a nozzle when the pressure ratio is equal to or less than the critical ratio. This expression is only a particular case of the author's Equation [9a], which holds for all pressure ratios and hence also for the critical ratio. In order to show this, by substituting in Equation [9a] $\frac{p}{p_1} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}$ and considering that $\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}} = \left(\frac{2}{k+1}\right)^{\frac{2}{k-1}} \cdot \frac{2}{k+1}$, we obtain directly

$$W = \frac{A p_1}{\sqrt{T_1}} \sqrt{\frac{2}{k+1} \cdot \frac{g}{R} \left(\frac{2}{k+1}\right)^{\frac{2}{k-1}}}$$

which is the same equation as Equation [6a] given by Mr. Naiman. Substituting the values for g, R, and k, we have

$$W = \frac{0.53\, A p_1}{\sqrt{T_1}}$$

[10] This equation simply says that any changes of density caused by changes of pressure are propagated with the acoustic velocity.

APM-53-5

On Problems in the Theory of Fluid-Film Lubrication, With an Experimental Method of Solution

By ALBERT KINGSBURY,[1] PHILADELPHIA, PA.

Analytical treatment of problems in perfect film lubrication is in general very difficult; solutions have thus far been obtained only in a few relatively simple cases. The paper describes methods by which fairly accurate solutions of any stated cases may be obtained experimentally, through the analogy between electrical potential and current flow in a conductor and pressure and volume flow in the lubricating film. Electrical measurements are made upon enlarged models of the lubricating films, the models being bodies of conducting fluid contained in wooden boxes having forms determined by theory. Pressures at points in a film having been deduced from the electrical data, graphical or other means of integration are used to determine the total load, the friction, and the quantities of oil flow at the edges of the film. The method is not limited as regards the shape of the film or variation of viscosity within the film. Solutions are given for a number of cases of plane bearing surfaces and journal bearings, showing close agreement with analytical solutions in cases where the latter have been obtained.

I—INTRODUCTORY

MATHEMATICALLY exact application of the theory of lubrication to practical cases is not generally feasible. Even in the simplest cases it involves difficult mathematical procedure, as well as much tedious work in numerical reduction. Until the work of Michell,(3)[2] on rectangular plane surfaces, no solution had been made that included the important feature of flow transverse to the rubbing motion. Recently, Boswall(4) has obtained approximate solutions for fitted partial journal bearings and for the sector-shaped surfaces of pivoted-block thrust bearings. But in order to avoid excessive mathematical difficulties, all the few solutions thus far obtained have been more or less restricted in the character of assumptions of film forms and of variation in temperature and viscosity of lubricant; factors that among others are not accurately known or determinable under working conditions in practice. Rigorous mathematical solutions, taking into account all the known physical actions involved, are as yet not obtainable; and even if such were obtained, they would necessarily be based upon certain ideal conditions, such as mechanical accuracy of the bearing surfaces, and freedom from distortion by pressure, heat, or wear.

On the other hand, the numerous direct experiments on bearings that have been conducted in the past have not led to general formulas for engineering use. Hersey(5) has shown, by dimensional reasoning, the algebraic relations that would enter into such formulas, but the necessary numerical coefficients, to be derived from experiments, are still wanting. Probably the main reason for this lack lies in the uncertainties as to the form and temperatures of the oil film in experimental bearings, leading to discordant results; differences of fit or finish cause great discrepancies. Experience indicates that extended, systematic, and carefully conducted experiments would be required in order to permit of general formulation with approximate accuracy; and even such experiments would require to be conducted in the light of physical theory. In the absence of such experiments, the fundamentally sound existing theory, omitting no major factor, and covering conditions that may be closely approximated in practice, is in itself a valuable guide to practice; and it would be still more valuable if more readily available. The object of the present paper is to present a method of extending the application of theory to bearing problems, by means of experimental solutions of stated cases, with sufficient accuracy for engineering purposes, and covering cases that have not heretofore yielded to mathematical treatment.

The approximate and partly experimental methods now to be described are not limited as regards form of bearing surfaces, or other assumptions such as viscosity variation; and relatively little mathematical work is involved, and that of a simple kind. As compared with the method of direct experiment upon bearings, the present method has the advantage of exact treatment of the film dimensions, and of permitting separate determinations of the effects due to separate actions, such as the effects of shear, of centrifugal action, and of boundary pressures. If the solutions obtained fail to fairly represent actual cases, it is only by reason of inexact or incomplete statement of the problems, not because of limitations of the method.

II—GENERAL DESCRIPTION OF THE EXPERIMENTAL METHOD

The method, described in brief, consists primarily in direct experiment upon large-scale electrically conducting models, each so designed that the space relations of current and voltage are very nearly the same as those of volume flow and pressure in lubricating films of arbitrarily assumed form and viscosity variation. The observations of current and voltage are readily interpreted in terms of volume flow and pressure; then by means of graphical or other approximate methods of integration, it is easy to determine the total pressure or load, its position and direction, the frictional resistance, and the volume and distribution of the oil flow. The scale of the electrical model is made large in the direction of thickness of the oil film, but not correspondingly large, if enlarged at all, in the other dimensions.

It is possible to draw various physical analogies to the conditions of flow and of stress in the lubricating film. Perry (8) remarks that viscous flow, the flow of heat or of electricity in a

---

[1] President, Kingsbury Machine Works. Mem. A.S.M.E.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Special Research Committee on Lubrication and presented at the Annual Meeting, New York, N. Y., December 1 to 5, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

conductor, stress and strain in an elastic solid, and other physical problems, are all correlated through the "equation of continuity." Heaviside (9) draws an analogy between viscous flow and the flow of electric current in a conductor. Michell (3) has applied the elastic analogy experimentally to the study of lubricating films between plane surfaces; but it does not appear that any of the other possible analogies have been utilized for experimental work in this field. The electric-current analogy offers the greatest possibilities, because of simplicity of the special apparatus, ease of control, and readiness of reasonably accurate measurements.

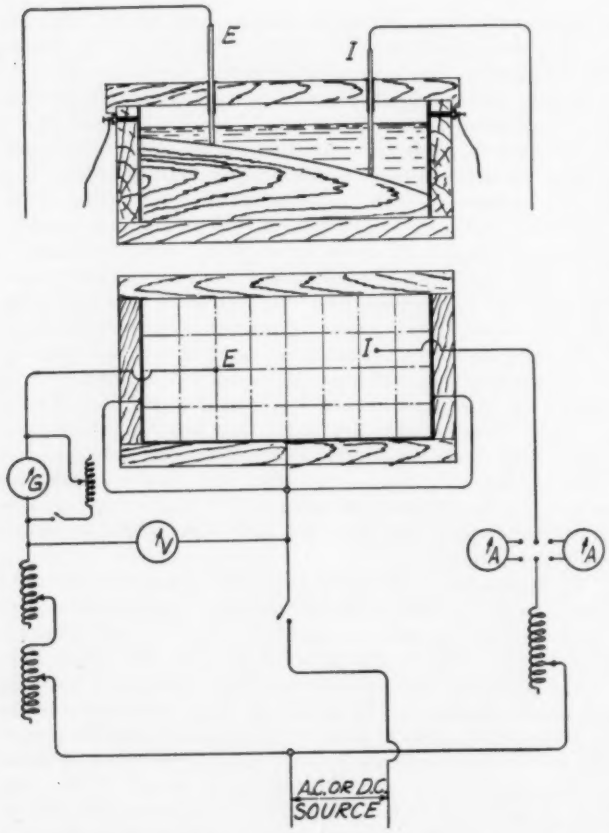


FIG. 1 TYPICAL FORM OF ARRANGEMENT OF TEST MODEL, WITH DIAGRAM OF THE ELECTRICAL INSTRUMENTS AND WIRING

The general arrangement of the model consists of a body of fluid of low electrical conductivity, with free upper surface, contained in a wooden box having the same plan as the developed bearing surface, to any desired scale; the varying depth of the bath is determined by a wooden form fastened in the box, its contours being determined according to the reasoning given in Section III. Metal plates inserted at the sides provide means for establishing potentials corresponding to pressure conditions at the edges of the bearing film. Metal-rod electrodes, inserted vertically to the full depth of the bath at chosen points, furnish means for supplying current, corresponding to volume flow in the oil film, and for measuring voltage, corresponding to pressure in the oil film. Details of some of the constructions will be described later.

The experimental procedure, according to the conditions of the problem, consists either in supplying measured current successively at the centers of small divisional areas of the bath, or of measured voltage at the side plates or of sectional parts thereof, and observing the voltage thereby produced at other points of the bath. The sum of the observed values of voltage at a given vertical in the bath due to all of the separate currents or voltages supplied, when reduced by the method described in Section III, gives the desired value of the pressure in the oil film at the corresponding point.

Fig. 1 shows a typical form of arrangement of test model, with a diagram of the electrical instruments and wiring.

The box and the form are most conveniently made of well-seasoned wood, which has been found serviceable if well coated with an asphalt paint or with a solution of celluloid in acetone. Several other coatings were tried, but none but those mentioned were found to be waterproof. The top edge of the box, made to a plane and set level, forms a datum from which to gage the depth of the bath, measured vertically from the bottom form to the free surface. The projected area is plotted to the desired divisional areas; the wood cover has drilled holes in which close-fitting copper tubes are inserted, for locating and guiding the rod electrodes, at the centers of the divisional areas for the current electrode, and at the corners of the divisional areas for the voltage electrode.

For several reasons, aqueous solutions appear to be the best substances available for the electrical models. As Kohlrausch (6) has pointed out, fluids are the only homogeneous conductors, all solids such as metals being more or less non-homogeneous; and aqueous solutions have true ohmic resistivity, which may readily be varied through a wide range by the use of more or less dissolved chemical. Their specific conductivities are low in all cases; and this is a requisite in the models, in order to permit of the necessary definite control of boundary potentials by means of metal-plate electrodes.

In some of the experiments relating to plane bearing surfaces, the test bath, constituting the model of the lubricating film, was a solution of pure cupric sulphate in distilled water, with a resistivity of about 50 ohm-inch. Copper was used for the side plates and the current and voltage electrodes; and direct current was used, supplied by one or two commercial dry cells. The use of direct current, instead of alternating current as used by Kohlrausch,(6) was due to the relatively great sensitiveness of direct-current instruments, and particularly because there appears to be no quick-reading a.c. voltmeter serviceable for very low ranges (0.002 to 0.005 volt).

The use of direct current in connection with an electrolytic bath for the model has some disadvantages, in that the contact of the electrolyte with the electrodes induces small and variable potentials and currents, independently of the action of the external batteries; and with continued use of the battery currents there are surface changes at the electrodes (polarization), causing surface resistances and counterpotentials that markedly affect the observations. These difficulties are minimized by using a salt of the electrode metal to provide the desired conductivity in the bath; by using a battery current that is barely large enough to make the contact currents of minor importance; by keeping the current flowing only long enough to read the instruments; and by reversing the current for alternate readings of potential at a given point, so that the errors nearly balance out in the summation. A highly important help in reducing the errors consists in frequent withdrawal of the copper parts from the bath and scouring them with emery cloth to remove the dark-colored deposits due to the electrolytic action. A current of the order of one-fourth ampere per square inch of the current electrode has been found suitable; excessive current causes undue local heating of the bath, with consequent local changes of resistance, which are undesirable, since all calculations are based upon uniform resistivity in the bath. With these precautions, check tests for the value of the pressure at a given point generally agree within about 1 per cent.

A more accurate and less troublesome arrangement was used in later experiments. As compared with the direct-current method described above, the changes consisted in the use of alternating current,(6) chromium-plated copper electrodes, and higher resistivity of bath.

The current was taken from the city supply, 110 volts, 60 cycles, and was controlled as required by wound rheostats.

The bath was a dilute solution of potassium dichromate in distilled water, having a resistivity of at least 500 ohm-inch, and in some cases much higher. By thus increasing the resistivity it was possible to raise the minimum measured voltages in the bath to about 0.050–0.060 volt, thus bringing the voltage measurements within the range of available a.c. voltmeters.

The chromium plating of the electrodes greatly reduced the attention required for keeping the surfaces clean. As received from the plating works, the chromium surfaces had been buffed, and apparently had a very thin coating of the stearic acid or other substance used with the abrasive for charging the buffing wheels, so that the bath solution would not wet the chromium, nor would cleaning by ordinary methods secure the necessary wetting. This difficulty was overcome by scouring the surfaces with fine washed emery moistened with a portion of the bath solution, applied with a pad of surgical cotton. This secured immediate and thorough wetting, and the surfaces so cleaned remained in constant condition throughout long series of tests, without evidence of plating, of oxidation, or of polarization. This condition is necessary for accurate results in the experimental work, since surface resistances at the electrodes would result in false indications of voltage at all points in the bath.

The resistivity of the bath solution was determined by direct measurements, using a prism of the fluid of about 6 sq. in. section, contained in a wood box about 24 in. long, with movable plate electrodes, and using the same electrical instruments that were used for the model tests. Resistivities were determined over the range of expected room temperatures, thus obtaining data for corrections for temperature changes that might occur during tests.

Referring to Fig. 1, it will be observed that the potentials were taken by a null method; if the current required to operate the voltmeter were taken from the bath, the distribution of current and of potential in the bath would thereby be altered and thus the true potential would not be read.

The electrical problems involved are closely akin to those of measuring the conductivity of electrolytes. On this subject there is an extensive literature in physical and chemical journals; with very few exceptions, the methods described involve the use of alternating current, and the principles and precautions necessary for precise results are applicable to the present experimental work.

III—THEORY OF THE EXPERIMENTAL METHOD: PRESSURES DUE TO RELATIVE MOTION OF BEARING SURFACES

The differential equation established by Reynolds (1), and now generally used as a basic expression, is

$$\frac{d}{dx}\left(h^3\frac{dp}{dx}\right)+\frac{d}{dz}\left(h^3\frac{dp}{dz}\right)=6\mu\left\{(U_0-U_1)\frac{dh}{dx}+2V_1\right\}\ldots[1]$$

in which

$U_0 - U_1$ is the relative tangential velocity of the surfaces
V_1 is the relative normal velocity of the surfaces
x is parallel to U_0 and U_1
z is at right-angles to x and parallel to both surfaces
h is the film thickness
p is the pressure, above atmospheric or other uniform ambient pressure, and
μ is the coefficient of viscosity, assumed constant by Reynolds.

This equation, derived from the fundamental expressions for stress in a viscous fluid (identical with those for an elastic solid), may be written, by dividing by 12μ,

$$\frac{d}{dx}\left(\frac{h^3}{12\mu}\frac{dp}{dx}\right)+\frac{d}{dz}\left(\frac{h^3}{12\mu}\frac{dp}{dz}\right)=\frac{U_0-U_1}{2}\frac{dh}{dx}+V_1\ldots[2]$$

In this form, the physical meaning of the terms is as follows: Considering as an "element" a volume of the film of unit dimensions in x and z, and of height h (the film thickness), the expression $\left(\frac{h^3}{12\mu}\frac{dp}{dx}\right)$ may readily be shown (2) to be the component of volume of lubricant, per unit time, flowing in the x-direction past the section of unit length in z and of height h, due to the pressure gradient $\frac{dp}{dx}$. The derivative of this quantity with respect to x, that is, the first term of [2], is the difference of the flow past the opposite faces of the element; it is the net volume flowing out from (or into) the element in the x-direction; and this is true whether μ be constant or variable, while [1] is true only if μ is constant. A precisely similar reasoning applies to the second term of [2]; this term expresses the net component rate of volume flow out from (or into) the element in the z-direction. The third term of [2] is the net volume per unit time flowing into (or out from) the element in the x-direction, as determined by a uniform rate of shear over the film thickness h, due to the relative motion of the bearing surfaces in the x-direction, without reference to pressure. The fourth term is the volume-displacement rate due to the relative component of motion normal to the nearly parallel bearing surfaces, also without reference to pressure.

Thus Equation [2] states, in symbols, with reference to a single elementary volume of the film, the equality of the inflow to the outflow; an equality that can exist only if the lubricant is continuous and incompressible, as assumed by Reynolds (1) in deriving Equation [1].

It appears also that if there be any other action than those considered above, by which a volume displacement for the element may be determined independently of pressure effects, a corresponding term may be inserted on the right in [2]; for example, expansion of the fluid by heat, or displacement of volume by centrifugal action. Further, it is important to note that because of the linear relation of pressure and viscous flow, in any solution or integration of [2] with the object of finding an expression for p at any point in terms of the known or assumed volume displacements on the right, it is permissible to find the separate values of p for the separate terms on the right, and the algebraic sum of such values is that due to all the terms on the right, whether the latter be considered together or separately. This is analogous to the addition or superposition of stresses and of strains in the theory of elasticity, the latter theory and that of lubrication being based upon the same fundamental equations. And further, if any or all of the terms on the right be considered as operative at a single element at x_1,y_1, producing there a pressure p_1, then at any other point x_2,y_2 there will be a pressure p_2, less than p_1, and depending for its value upon the ways in which the flow from x_1,y_1 is distributed throughout the film. And if further the terms on the right be operative simultaneously at all the elements, the total pressure at a point x_2,y_2 is the algebraic sum of all the pressures p_2 due to all the elementary volume supplies taken separately. This latter principle is the basis of the present experimental method.

For present purposes, the basic differential equation for the oil film may be written, from [2],

$$\frac{d}{dx}\left(\frac{h^3}{\varphi}\frac{dp}{dx}\right)+\frac{d}{dz}\left(\frac{h^3}{\varphi}\frac{dp}{dz}\right)=12\mu_0 q \quad \ldots\ldots\ldots [3]$$

in which μ_0 is the viscosity at a base point, and φ a non-dimensional function of x and z such that the viscosity at any point is given by $\varphi\mu_0$, and q represents any or all of the terms on the right in [2]. The solution of [3] for p requires that h, φ, and q shall be determinable from the data of the problem, for all parts of the surface, whether or not they can be expressed as continuous functions of the film dimensions and the surface motions; and the boundary values of p are also required data.

In order to establish working equations for the experimental method, we have first to consider that just as viscous flow is proportional to the pressure gradient and inversely as the viscosity, so is current flow proportional to the potential gradient and inversely as the resistivity (Ohm's law). There is, however, this difference: that the viscous flow is proportional to the third power of the film thickness, while the current flow is proportional to the first power only. Thus following the reasoning that has been applied to Equation [2], as applying equally to the electrical conditions, we may at once write the electrical equation analogous to [2], giving the differential relation of current and potential in the model,

$$\frac{d}{dx}\left(\frac{H}{r}\frac{dE}{dx}\right)+\frac{d}{dz}\left(\frac{H}{r}\frac{dE}{dz}\right)=i=(\text{approx.})\ \frac{I}{\alpha} \quad \ldots\ldots [4]$$

in which H is the variable depth of the bath, r is the uniform resistivity, E the potential, i the current supplied per unit area. Like Equation [2], this is an equation of continuity; it is the symbolized statement of the equality of the inflow and the outflow of current, with reference to a single elementary volume of the bath, of unit-area base and of height H. The reason for the expression I/α is that in the experiment it is not generally practicable to deal directly with unit areas; the whole area is divided into an arbitrary number of (preferably equal) areas α, and a current I is supplied at the center of each of these areas successively; thus instead of i we may write I/α, the mean intensity of current for the divisional area, as giving an approximation which is closer as the areas α are smaller and therefore more numerous. Practically, it is desirable to make the number of divisions as small as may be, since for each point where the pressure is to be determined, observations must be taken for each of the divisional areas; but supplying a current I at a point within the area α is not the exact equivalent of the theoretical distributed current i over the same area, and the discrepancy is in general greater as α is greater. Thus a compromise is required, for which rules have not been laid down; but it may be said that in general a very large number of divisions is not required; in some cases a small number, such as eight to twelve, may suffice.

Comparing [3] and [4], it is evident that if we were to make, numerically, $H=\frac{h^3}{\varphi}$ and $\frac{Ir}{\alpha}=12\mu_0 q$ at all points, then a complete experimental solution of [4] for the numerical value of E at any point would give also the numerical value of p at the same point; assuming, of course, that the values of E and p are made the same at all points around the edges. But in the experimental work it is desirable to make H a relatively large dimension, say, of the order of one inch; h may be of the order 10^{-3} inch, hence h^3 is of the order 10^{-9} inch; φ may be of the order unity. Thus we may write $H=K\frac{h^3}{\varphi}$ where K is a constant of the order $10.^9$ Introducing this factor K in [3], we have

$$\frac{d}{dx}\left(K\frac{h^3}{\varphi}\frac{dp}{dx}\right)+\frac{d}{dz}\left(K\frac{h^3}{\varphi}\frac{dp}{dz}\right)=12K\mu_0 q \quad \ldots\ldots [5]$$

Now referring to [4], if we supply any measured current I_1 at a single small divisional part having an area α_1 at x_1,y_1, and measure the potential ΔE_2, thereby produced at a point x_2,y_2, the relationship may be indicated as

$$\Delta E_2=\frac{I_1 r}{\alpha_1}\psi$$

in which ψ is an undetermined function of x_1,z_1,x_2,z_2. Considering [5] in the same way and for the same points, the same value of ψ will apply; hence we may write

$$\Delta p_2=12\,K\mu_0 q_1\psi$$

whence, by elimination of ψ,

$$\Delta p_2=12\,K\mu_0\Delta E_2\frac{q_1\alpha_1}{I_1 r}$$

Thus considering the whole bearing surface, we may write for the pressure at any point, dropping the subscripts for generality,

$$p=\Sigma(\Delta p)=12\,K\mu_0\,\Sigma\left(\frac{\Delta E}{Ir}\,q\alpha\right) \quad \ldots\ldots [6]$$

or, replacing K by its value $\frac{H\varphi}{h^3}$,

$$p=12\,\mu_0\,\Sigma\left(\frac{\Delta E}{Ir}\frac{H\varphi\,q\alpha}{h^3}\right) \quad \ldots\ldots [7]$$

Equations [6] and [7] are the general expressions for the experimental determination of the pressure, in so far as it is produced by the relative motion of the bearing surfaces. If there are, in addition, external actions producing non-uniform pressures in the film, these are most conveniently treated separately, as shown in the following section. In the application of [6] and [7] it is necessary to bear in mind that all the variables on the right relate to the points where the current is supplied, except ΔE, which relates to the points where p is to be determined.

IV—NON-UNIFORM BOUNDARY PRESSURES

If a part of the boundary of the film is at a known pressure p_0 due to external conditions, the remainder of the boundary being at zero pressure, there results a non-uniform pressure field in the film, the pressure Δp at any point depending upon the form and dimensions of the film, but directly proportional to p_0. For the experimental determination of the pressure, the electrical model, proportioned as in III, may be provided with a metal plate at the wall, covering the corresponding part of the boundary of the bath, and kept at a potential E_0, while the remainder of the boundary is held at zero potential; this produces a potential field in the bath exactly similar to the pressure field in the film, and the potential ΔE at any point is directly proportional to E_0. We may then write for the pressure at any point,

$$\Delta p=\frac{\Delta E}{E_0}\,p_0$$

If the pressure p_0 varies from point to point at the edges of the film, the resulting pressure at a point in the film may be found approximately by dividing the wall plate into parts, each representing a part of the film boundary having a mean local

pressure p_0. Dealing with these parts successively as described above, the pressure p at a point in the film is, approximately,

$$p = \Sigma(\Delta p) = \Sigma\left(\frac{\Delta E}{E_0}\, p_0\right)$$

It will be noted that the value of the resistivity of the bath is not involved; the only requisite is that the resistivity be uniform throughout the bath, as in all cases.

The voltages E_0 and ΔE may be measured separately by the use of voltmeters; but since only their ratio is involved, a simpler and more accurate method is to measure their ratios directly by means of a Wheatstone bridge, thus dispensing with all absolute electrical measurements.

V—INDIRECT METHOD

In cases in which h and μ are assumed to be functions of x only, independent of z, a solution for the "infinitely wide" bearing may first be made, resulting in $p_x = f(x)$. This may be done by calculation in the simpler cases, and in any case either by

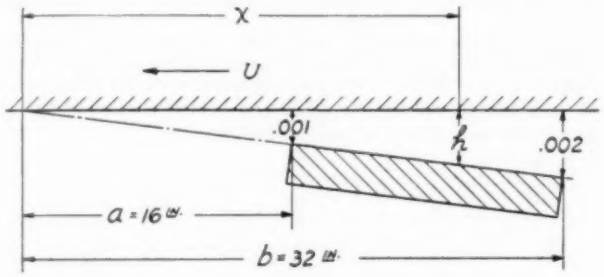


FIG. 2

graphics or by the experimental method of III. Having this solution, the pressures for any finite width may be found by the aid of IV with much less work than by the direct application of III, as follows: Referring to Fig. 3, if the model be treated by the method of III for the case of infinite width, the pressure $p_x = (x)$ exists along the non-conducting side BB as well as elsewhere; if for finite width, the pressure along BB is kept at zero by a conducting plate. We may therefore write in a case of finite width, for any point x, z,

$$p_q = p_x - p_s$$

in which p_q = pressure with side leakage at BB
p_x = pressure without side leakage at BB
p_s = pressure due to pressures $p_0 = p_x = f(x)$ along the side BB, acting alone.

The pressures p_s may be found by method IV with a minimum of time and labor. Thus in the case of the model of Fig. 3, five divisions of the side plate at BB were found fairly sufficient; this required but five observations for each point where the pressure is to be found; while when the method III was applied directly there were 32 observations of voltage and 32 of current for each pressure determination. Obviously the saving by the indirect method is greater as the width of the bearing in the z-direction increases, and as the number of divisions of BB diminishes.

The sole objection to this indirect method is that for narrow bearings, and for points near the boundary in any case, the value of p_q is small compared with p_x, and thus a small experimental error in p_s involves a larger proportionate error in p_q. Nevertheless experience indicates that owing to the superior accuracy of the bridge method of IV, this indirect method is preferable, as regards both simplicity and accuracy, for the cases where it is applicable.

VI—PLANE SURFACES: RECTANGULAR SLIDE BLOCK, OF INFINITE WIDTH AT RIGHT ANGLES TO THE RECTILINEAR MOTION; CONSTANT VISCOSITY; STEADY CONDITIONS

This hypothetical case was chosen primarily as furnishing a first critical test of the experimental method, since the pressures can be calculated independently and exactly.

Fig. 2 shows the assumed dimensions of the bearing block and the oil film, and Fig. 3 the arrangement of the model. The constant K was chosen as 5×10^8; thus the depth of the bath varies from $(0.002)^3 \times 5 \times 10^8 = 4$ in. at the inlet end, to $(0.001)^3 \times 5 \times 10^8 = 0.5$ in. at the outlet end. The metal plates at these ends, connected by wire, serve to determine zero

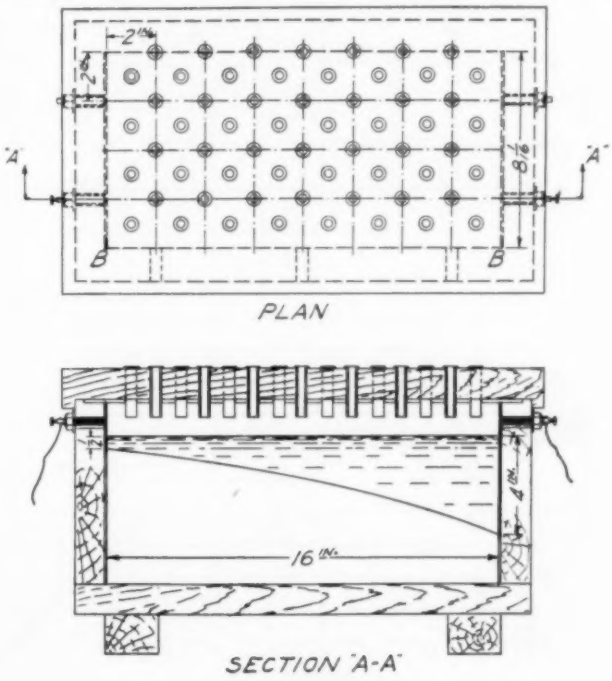


FIG. 3

potential, corresponding to zero pressure; the potentials measured in the bath are differences (+ or —) from the metal-plate potential, just as the pressures in the oil film are differences above or below the external pressure. The bottom form and the sides, being non-conductors, prevent any transverse flow of current. Thus the model represents a section of the oil film in the hypothetical case of infinite width at right angles to the motion.

Applying the method III, the following values are to be substituted in Equation [6] or [7], as required; U being the relative velocity $U_0 - U_1$ of Equations [1] and [2]:

$K = 5 \times 10^8$
r = 49.5 ohm-inch at 50 deg. fahr. with corrections for varying temperature
$q = \frac{U}{2}\frac{dh}{dx} = \frac{U}{32{,}000}$
$\varphi = 1$
α = 2 in. × 2 in. = 4 sq. in.
$h = cx$
$\frac{dh}{dx} = c = \frac{1}{16{,}000}$

Equation [6] thus becomes

$$p = 7.5 \times 10^8 \mu_0 U \Sigma \left(\frac{\Delta E}{Ir}\right) \ldots\ldots\ldots\ldots [8]$$

Equation [7] becomes

$$p = \frac{6\mu_0 U}{c^2} \Sigma \left(\frac{\Delta E}{Ir} \frac{H\alpha}{(x^2)}\right) \ldots\ldots\ldots\ldots [9]$$

Either [8] or [9] may be used for the numerical calculation. There are 32 terms $\frac{\Delta E}{Ir}$ for the pressure summation at each point. Fig. 4 shows the results of a test, made with alternating current. The circled points show the experimental values of p/C and the full curve the theoretical values; the latter being given by the expression, derived by integrating [3] with omission of the second term,

$$p/C = \left[\frac{(x-a)}{(a+b)} \frac{(b-x)}{x^2}\right] \text{ in which } C = \frac{6\mu U}{c^2} \ldots . [10]$$

It will be noted that the experimental results agree very closely with the theory. The pressures shown in Fig. 4 for this simple case relate only to a bearing block of the particular dimensions $a = 16$ in., $b = 32$ in. For any other dimensions with the same general assumptions, the pressures could be calculated from Equation [10]; but in this case, as in others to follow, the specific experimental results are readily convertible to apply to *geometrically similar* bearings, by means of dimensional reasoning.(5) Thus in Equation [9], the factors in $\frac{\mu_0 U}{c^2}$ are independent of the size of the bearing; the group $\frac{H\alpha}{x^2}$ is of zero dimensions; but the group $\frac{\Delta E}{Ir}$ is of dimension L^{-1}. Hence in a bearing of the size $a = 16n$ in., the pressures at similarly located points would be $\frac{1}{n}$ th of those found by the experiment described.

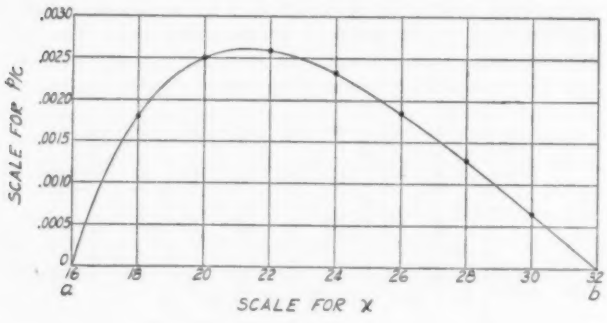


FIG. 4

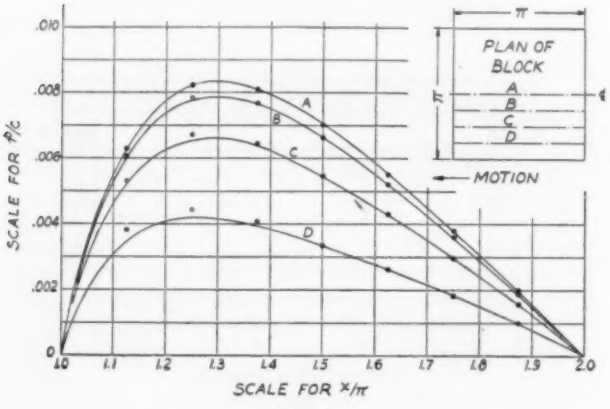


FIG. 5

VII—PLANE SURFACES; RECTILINEAR MOTION; SQUARE SLIDE BLOCK; LEAKAGE AT RIGHT-ANGLES TO THE MOTION; CONSTANT VISCOSITY; STEADY CONDITIONS

This case was chosen as a further test of the applicability of the electrical method, comparison being made with the theoretical results obtained by Michell.(3) As in the preceding case, the conditions are hypothetical, such cases not existing in practice.

The dimensions of the model are shown in Fig. 3. The arrangement differs from that of VI only by the addition of a zero-potential conducting plate at *BB*, corresponding to the condition of leakage along that side of the oil film. The model thus represents one-half of the oil film, either side of the median plane.

The numerical data are the same as in VI, and Equations [8] and [9] and deductions therefrom also apply; the only differences lie in the values $\frac{\Delta E}{Ir}$, and there are, as in the preceding case, $4 \times 8 = 32$ such terms in the summation for the pressure at each point; and for the 28 chosen points, a total of 896 observations each of current and of voltage.

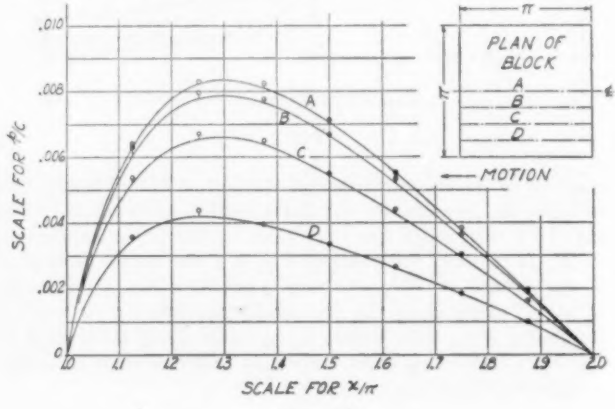


FIG. 5(a)

Fig. 5 shows the comparison, the circled points being the test results by method III, obtained with alternating current; and the full curves, graphical interpolations from the results of Michell's calculations.

In this case, the experimental values average 2.2 per cent higher than the pressures calculated by Michell; thus the agreement of the experiment with the mathematical results is less exact than in VI and in all other cases tested in which no side leakage is involved.

Fig. 5(*a*) shows a further test for this case by the indirect method V. The mean error in this case is 0.9 per cent in excess of Michell's results. A repeat test by method V confirmed the circled points so closely that the differences could not be shown on the diagram.

VIII—PIVOTED-BLOCK THRUST BEARINGS; SECTOR-SHAPED SURFACES; SIDE LEAKAGE; VARIABLE VISCOSITY

Fig. 6 shows the assumed form and dimensions of the pivoted block. The "axis of tilt" is the line of intersection of the extended plane of the block with the surface of the rotating collar,

and the particular choice of this axis is such as to make the oil film geometrically similar along arcs at all radii (a condition not necessarily the best in practice) and the variable thickness at the leading edge twice that at corresponding points at the trailing edge.

The viscosity is assumed to be uniform along any radial line, but varying uniformly with the angular position of the radius, so that at the trailing edge it is one-half that at the leading edge. This assumption is largely arbitrary, in view of the highly complex nature of the conditions of heat equilibrium in the film, and of the meagerness of existing physical data on the subject; yet for conditions of high speed and pressure it is probably closer to the truth than would be, for example, an assumption of constant viscosity.

The data for the specific case, partly shown in Fig. 6, and the substitutions for changing Equations [6] and [7] to polar coordinates are as follows:

$K = 5 \times 10^8$

c = dihedral angle between the bearing surfaces

$h = c\rho \sin\theta$

$\frac{dh}{dx} = c \cos\theta$

$a = b\rho\,\Delta\theta$ (values of ρ being taken at the middle of the widths b)

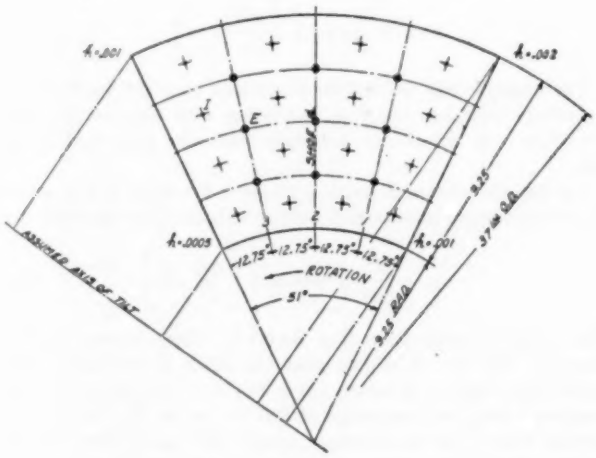


FIG. 6

ω = angular velocity of collar, radians per second

N = angular velocity of collar, revolutions per minute

U = linear speed of collar at any radius $= \omega\rho = \frac{\pi N}{30}\rho$

$q = \frac{U}{2}\frac{dh}{dx} = \frac{\pi}{60} N\rho c \cos\theta$

Equation [6] thus becomes

$$p = \frac{\pi}{5}\mu_0 N c b\,(\Delta\theta) K \Sigma\left[\frac{\Delta E}{Ir}\rho^2\cos\theta\right] \quad \ldots\ldots\ldots [10]$$

Equation [7] becomes

$$p = \frac{\mu_0 N}{c^2}\frac{\pi}{5} b\,(\Delta\theta)\,\Sigma\left[\frac{\Delta E}{Ir}\frac{H\varphi\cos\theta}{\rho\sin^3\theta}\right] \quad \ldots\ldots\ldots [11]$$

Figs. 7 and 8 show the pressure distribution obtained by tests with direct current, for the block of Fig. 6, and the film of maximum thickness 0.002 in., with $N = 100$ revolutions per minute and $\mu_0 = 3.4 \times 10^{-8}$ (inch, lb., sec.) corresponding to about 150 seconds Saybolt.

For extending these experimental values of p to cases of larger or smaller blocks, geometrically similar to the experimental block, it may be noted that in Equation [11] the dimensions of the product $\mu_0 N$ are the same as those of p; namely, $ML^{-1}T^{-2}$, and the remaining factors reduce to zero dimensions. Hence for the same N, μ_0, and c the pressure at any point is independent

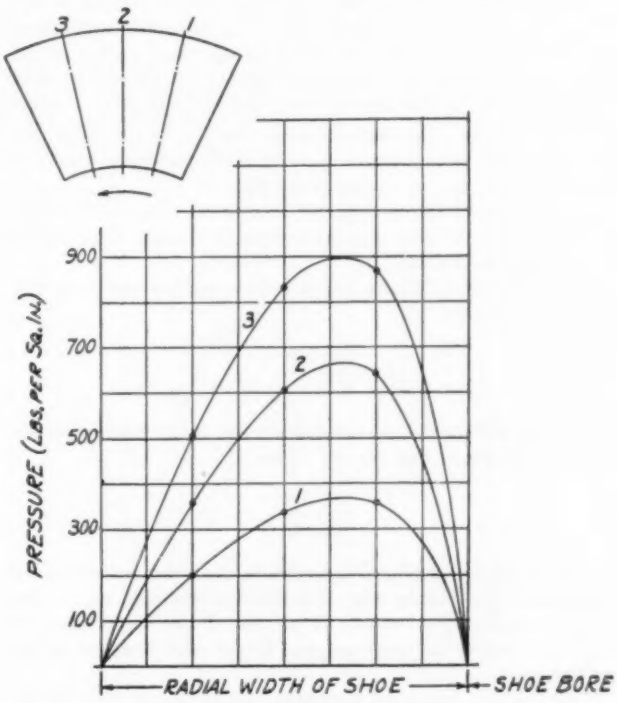


FIG. 7

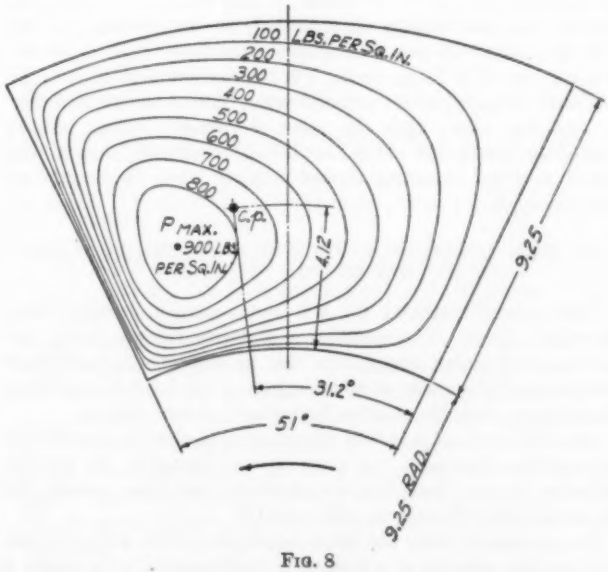


FIG. 8

of the size of the block; but for a given circumferential speed, the pressure is inversely as the size.

The friction torque is determined for the numerical example given above, by means of Reynolds' equation for the shear stress at the collar surface, x being in the direction of the motion.

$$f_s = \frac{\mu U}{h} + \frac{dp}{dx}\frac{h}{2} \quad \ldots\ldots\ldots [12]$$

The values $\frac{dp}{dx} = \frac{dp}{\rho d\theta}$ are found graphically at the selected points, from the pressure curves plotted from the experimental data, and the values of $\mu U/h$ for the same points are readily calculated; then by approximate integration the mean pressure is found to be 352 lb. per sq. in. The friction torque is 508 in-lb., and the power loss 0.805 hp. The torque is equivalent to a friction force of 36.6 lb. acting at the mean radius of the block, corresponding to a mean coefficient of friction of 0.000908. This coefficient agrees closely with the formula $f = 0.00009\sqrt{\text{r.p.m.}}$ derived from tests on large bearings of this form, for the same mean pressure and approximately the same viscosity of the oil at the entrance edge of the film.

The volumes of oil entering and leaving the film, for the same numerical example, were also determined by the aid of the curves of pressure distribution.

The volume of oil flowing per unit time and per unit length of radial edge, is given by

$$\frac{U}{2}h \pm \frac{h^3}{12\mu}\frac{dp}{dx}$$

and for the circular edges, the volume lost is determined by the rate per unit length and per unit time.

$$\frac{h^3}{12\mu}\frac{dp}{d\rho}$$

It was found by graphical integration that the volume of oil entering at the leading edge is approximately 0.98 cu. in. per sec.; of which approximately 27 per cent is lost along the outer arc, 7 per cent at the inner arc, and 66 per cent flows out at the trailing edge.

If all the heat generated were absorbed by the oil, then assuming a specific gravity of 0.91 and a specific heat of 0.45, the mean temperature rise of the oil would be 21.9 deg. cent. Assuming further that the leakage oil is heated, on the average, by one half the maximum rise, the temperature rise of the oil at the trailing edge is 26.5 deg. cent. The original assumption as to the viscosity change implies a temperature rise of about 17.5 deg to 19.5 deg. cent.; thus the assumed viscosity change implies that about one-fourth of the heat is conducted away through the metal, and the remaining three-fourths is taken up directly by the oil supply.

IX—THE PRESSURE EFFECT DUE TO CENTRIFUGAL ACTION IN THE OIL FILM

This was determined for the sector-shaped bearing block described above. The question is of minor interest only; but in view of frequent suggestions that at high speeds centrifugal effects must be considerable in bearings of this kind, it is perhaps desirable to show the relative importance of such effects.

For this purpose, it is first necessary to derive by analysis the appropriate expression for q for use in Equation [6] or [7]; all other factors, including the electrical test data, remain the same as for the Equations [10] and [11].

In accordance with the usual conception, it is assumed that the angular velocity of a lamina of thickness dy, at a height y above the stationary surface, is $\frac{y\omega}{h}$; ω being the angular velocity of the collar, and h the thickness of the film. The density of the oil being m, each such lamina, of volume $\rho d\rho \cdot d\theta dy$ is subject to a "centrifugal force" of the value

$$m\rho^3\omega^2 d\theta \cdot d\rho \cdot \frac{y^2 dy}{h^2}$$

This force is balanced by the difference of the radial components of the shear forces on the upper and lower faces of the element. Applying the established laws of laminar flow, it is readily shown by integration that the volume flowing across a section $h \cdot \rho d\theta$ in unit time is

$$\frac{m\omega^2\rho^2 d\theta h^3}{40\,\mu_0\varphi}$$

Hence the difference of flow past two such sections at an interval of $d\rho$ is

$$\frac{m\omega^2 d\theta}{40\,\mu_0} d_\rho\left(\frac{h^3\rho^2}{\varphi}\right)$$

Substituting for h its value $c\rho \sin\theta$, differentiating, and dividing by $\rho d\rho \cdot d\theta$ to reduce to terms of unit area of bearing surface, we have the desired value

$$q = \frac{m\omega^2 c^2\rho^3 \sin^3\theta}{8\,\mu_0\varphi} = \frac{m\omega^2}{8\,\mu_0}\frac{h^3}{\varphi} \quad \ldots\ldots\ldots\ldots [13]$$

Thus Equations [6] and [7] become, by substitution for q,

$$-p = \frac{3}{2} m\omega^2 K \Sigma\left(\frac{\Delta E}{Ir}\frac{h^3}{\varphi}\alpha\right) \quad \ldots\ldots\ldots\ldots [14]$$

$$-p = \frac{3}{2} m\omega^2 \Sigma\left(\frac{\Delta E}{Ir} H \alpha\right) \quad \ldots\ldots\ldots\ldots [15]$$

The negative sign is used for the reason that the value of q is essentially negative, the flow increasing with the radius; thus the effect is to reduce any pressures otherwise generated in the film.

For the special case in hand, expressing the value of ω in terms of revolutions per minute and with $\alpha = b\rho\Delta\theta$, [15] becomes

$$-p = \frac{\pi^2}{600} N^2 mb\ \Delta\theta \Sigma\left[\frac{\Delta E}{Ir}\rho H\right] \quad \ldots\ldots\ldots [16]$$

The pressure distribution thus found by direct current tests is shown in Fig. 9. It is seen that the effect is very small, the maximum negative pressure being but 0.11 lb. per sq. in., as compared with the maximum of 900 lb. per sq. in. due to tangential shear. At the assumed speed, 100 r.p.m., the ratio is about 1/8000; the ratio would vary directly as the speed, the value of c being held constant; but even at the very high speed of 1000 r.p.m., the ratio would be but 1/800. Thus for all practical purposes, the centrifugal effect is negligible.

X—EFFECTS OF MOMENTUM OF OIL AT FILM ENTRANCE

If the channel through which oil is supplied to the leading edge of the pressure film is of large section compared with the film thickness, the oil adjacent to the moving surface has substantially the same speed as that surface, throughout a layer that is of fairly large thickness compared to the thickness of the film. Hence the oil enters the film at a pressure approximately the same as that at center of a jet of velocity U directed against a fixed plate; and if the bearing block is square-edged, the entry pressure may be assumed to have approximately the value established by experiments with jets directed normally against fixed plates, $p_e = mU^2/2$.

The volume that actually enters the film is small compared to the total jet volume, and its mean velocity of entrance is reduced in general to less than one-half of the jet velocity. Hence the entry pressure is presumably not greatly affected by the flow into the film. In practice, however, the edge of the bearing

block is at least slightly rounded or beveled; and it appears that this should cause the entrance pressure to exceed the above value. Only direct experiment could determine the true entrance pressure; and for want of such tests, the pressure is taken as $mU^2/2$ for present purposes.

The pressure so calculated is not important in the case of moderate speed; but in recent practice, particularly in pivoted-block thrust bearings, the speed in some cases exceeds 200 ft. per sec. At this speed, with oil of specific gravity 0.9, the calculated entry pressure is 242 lb. per sq. in.; this obviously has an appreciable influence upon the pressure distribution in the film, since in high-speed bearings the mean pressure rarely exceeds 500 lb. per sq. in.

In case the oil is supplied under pressure, the electrical method may be used for determining the combined effects of supply pressure and jet pressure, by treating their sum as the initial pressure at the leading edge; or the effects of each upon the film pressure may be determined separately, as may be desired.

For the determination of the film pressures due to any assumed entry pressure, the general Equations [6] and [7] do not apply directly, since the values of q at the several divisional areas, due

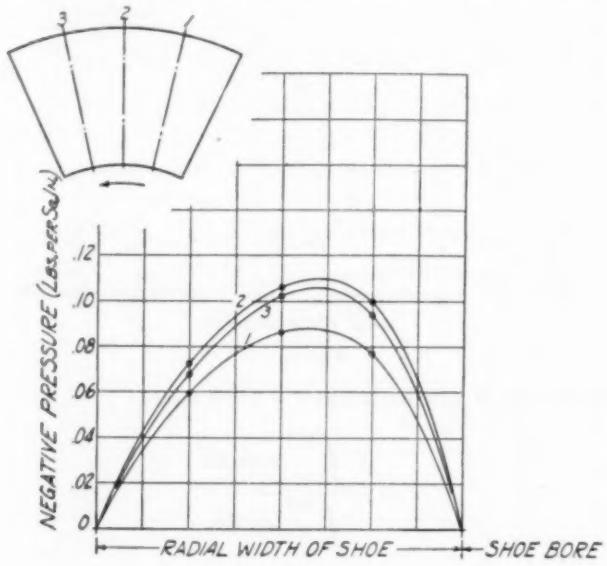


FIG. 9

to the entry pressure, are not determinable in advance. The case is treated on the basis of boundary pressures only.

The simplest case is that in which the entry pressure is uniform along the leading edge, as in journal bearings. The electrical analogue is established according to the principles of IV, by keeping the plate electrode at the leading edge of the bath at a potential E_0 above the nominal zero potential of the other edge plates; the potential E is then observed at any desired point x, y, in the bath. Then p_0 being the assumed jet pressure, and p the resulting pressure at x, y.

$$p = p_0 \frac{E}{E_0} = 1.37 \times 10^{-3} D^2 N^2 m \frac{E}{E_0} \ldots\ldots [17]$$

In the case of non-uniform pressure along the leading edge, as exemplified by the jet pressure in thrust bearings, the electrical arrangement is provided according to method IV, by substituting for the single plate electrode at the leading edge a number of narrow vertical strips, slightly separated from each other and from the side plates. In the test, a voltage E_0 is applied to one of the strips, the remaining strips and plate electrodes being kept at the nominal zero potential. The voltage ΔE is then observed at a point x,y of the bath. The calculated or assumed mean pressure for the position of the strip being p_0, and Δp the resulting pressure in the film at x,y,

$$\Delta p = p_0 \frac{\Delta E}{E_0}$$

All the separate strips being treated similarly, the pressure total at the point x,y is, approximately,

$$p = \Sigma (\Delta p) = \Sigma \left(p_0 \frac{\Delta E}{E_0} \right)$$

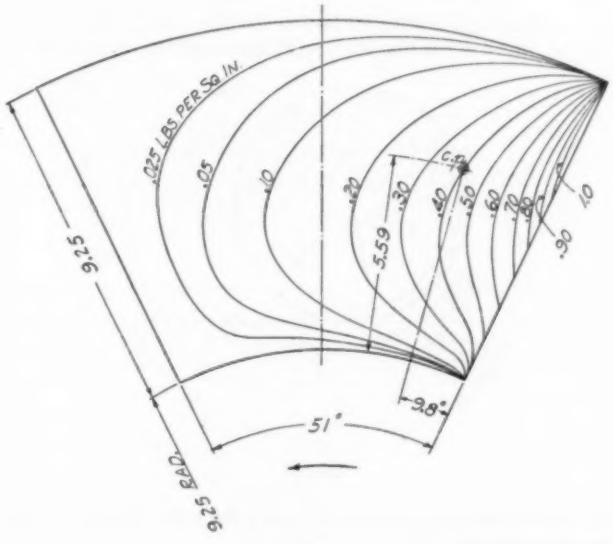


FIG. 10

On the assumption that $p_0 = \frac{mU^2}{2}$, and with ρ the mean distance of a strip from the center of rotation, it follows that

$$p = \frac{\pi^2 N^2 m}{1800} \Sigma \left(\frac{\Delta E \rho^2}{E_0} \right) = 5.49 \times 10^{-3} N^2 m \Sigma \left(\frac{\Delta E \rho^2}{E_0} \right) \ldots [18]$$

Fig. 10 shows the results of this method as applied to the thrust bearing (Fig. 6) with $N = 100$, for which the pressures produced by shear are shown in Figs. 7 and 8. In this case four voltage strips separated $^1/_8$ in. were provided at the leading edge, and direct current was used. A combined pressure diagram could be drawn by direct addition of the shear pressure and the inertia pressure, for a given film form and rotative speed; but this would require a separate combined diagram for each rotative speed, since the inertia pressure varies as the square of the speed, while the shear pressure varies as the first power if the film thickness is held constant, or is constant if the total load is constant.

XI—JOURNAL BEARINGS: CYLINDRICAL SURFACES, AXES PARALLEL. METHOD III

Fig. 11 shows the geometrical notation.

a = radius of journal
$a + \eta$ = radius of bearing
η = radial clearance
c = eccentricity factor

$e = c\eta$ = distance between the axis of the journal and that of the bearing
θ = angle measured from line of centers, in direction of U
ϕ = angle of inclination of journal load
U = linear velocity of journal surface = $\frac{\pi a N}{30}$
N = revolutions per minute
$dx = a d\theta$
z = distance parallel to axes, from middle of width
p = pressure in film at any point θ, z
β = mean value of p in Z-direction at any value of θ
R = mean load per unit width of bearing surface
$h = \eta(1 + c \cos\theta)$ = film thickness
$\frac{dh}{dx} = -\frac{\eta c \sin\theta}{a}$
l = length of bearing surface in the direction of the motion
w = width of bearing surface at right angle to the motion.

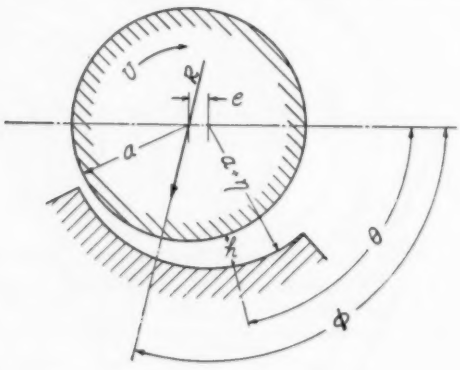


FIG. 11

For substitution in Equation [6] or [7] the value of q for journal bearings thus becomes:

$$q = \frac{U}{2}\frac{dh}{dx} = \frac{\pi}{60} N \eta c \sin\theta$$

Equation [6] becomes:

$$p = \frac{\pi K}{5} N \mu_0 \eta c \Sigma \left(\frac{\Delta E \alpha}{Ir} \sin\theta\right) \ldots\ldots\ldots [19]$$

and [7] becomes:

$$p = \frac{\pi}{5} N \mu_0 \eta c \Sigma \left(\frac{\Delta E}{Ir}\frac{H \varphi \alpha}{h^3} \sin\theta\right) \ldots\ldots\ldots [20]$$

If the values of $\sin\theta$ are taken at the centers of the areas α, Equations [19] and [20] are nearly exact if these areas are relatively small; but more precisely, the value of $\sin\theta$ for each area α should be the mean value; the further, the current should be supplied, not at the center of the area, but at the "center of gravity" of the values of $\sin\theta$ within the area. These latter considerations may be taken into account when the areas α are relatively large, and particularly for any area having a large variation of $\sin\theta$. Thus if the areas are rectangles of sides Δx and Δz, and the variation of h in the distance Δx is Δh, the exact value of $q\alpha$ in [6] is:

$$q\alpha = \frac{U}{2}\frac{\Delta h}{\Delta x} \Delta x \Delta z = \frac{U}{2} \Delta h \Delta z$$

Thus [6] becomes, as the most suitable form for use,

$$p = \frac{\pi}{5} K N a \mu_0 \Sigma \left(\frac{\Delta E}{Ir} \Delta h \Delta z\right) \ldots\ldots\ldots [21]$$

As regards the location of the current electrode within an area $\alpha = \Delta x . \Delta z$: since q varies as dh/dx, the best experimental approach to the theoretical distribution of q is obtained by placing the current electrode at the "center of gravity" of the variable q for the the given area. This center is in the middle of the width Δz, and its position in the x-direction may readily be estimated from a curve $y = dh/dx$. Or passing to the analytic form, if $\Delta x = x_2 - x_1$, then the x-location is given, by the usual method for centers of gravity,

$$\bar{x} = \frac{\int_{x_1}^{x_2} \frac{dh}{dx} x dx}{\int_{x_1}^{x_2} \frac{dh}{dx} dx} \ldots\ldots\ldots [22]$$

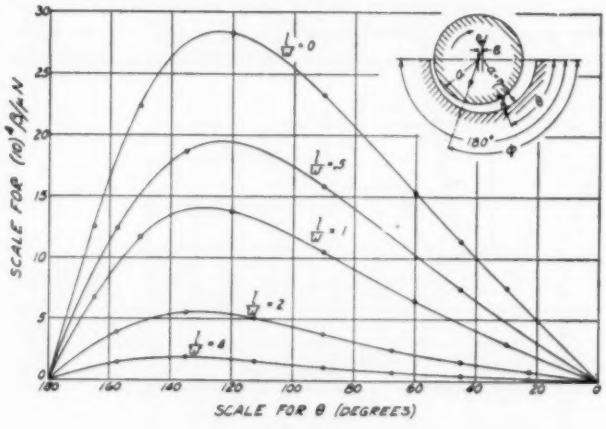


FIG. 12

For example, if Δx extends from $\theta = 0$ to $\theta = \pi/4$,

$$\bar{x} = \frac{a \int_0^{\pi/4} \theta \sin\theta d\theta}{\int_0^{\pi/4} \sin\theta d\theta}$$

$$= \frac{a \left[\sin\theta - \theta\cos\theta\right]_0^{\pi/4}}{\left[\cos\theta\right]_0^{\pi/4}} = 0.518a = 0.66\,\Delta x$$

Thus in this case the current electrode is advanced by $0.16\,\Delta x$ in the direction of decreasing h, beyond the center of figure of the divisional area under consideration. For an area in which Δx extends from $\theta = \pi/4$ to $\theta = \pi/2$, the corresponding correction is much smaller, being but $0.0267\,\Delta x$. The quantity β, which as defined above represents $\int_{z=0}^{z=w/2} \frac{pdz}{w/2}$ at any chosen value of θ, affords a convenient simple basis for comparing visually the load capacities of bearings of a given length l with varying widths w as in Figs. 12 and 15. Further, the curves of β plotted against x or θ, may be used as if β were actually a uniform pressure over the width w, in all calculations for the total load and its direction; for the friction; and for the quantities of oil entering and leaving the film. This is, however, true only with h independent of z, as determined by the assumed condition of parallel axes.

XII—180-DEG. JOURNAL BEARING. METHOD III

Referring to Fig. 12, the dimensions of the models were made to conform to the following bearing data:

a = 3.82 in. (24 in. circumference)
η = 0.001 a = 0.00382 in.
c = 0.4

The viscosity μ was assumed constant.

The test results using alternating current, for five ratios l/w are shown in Fig 12, the circled points being the experimental values of $\beta/\mu N$. For the case of l/w = 0 (infinitely

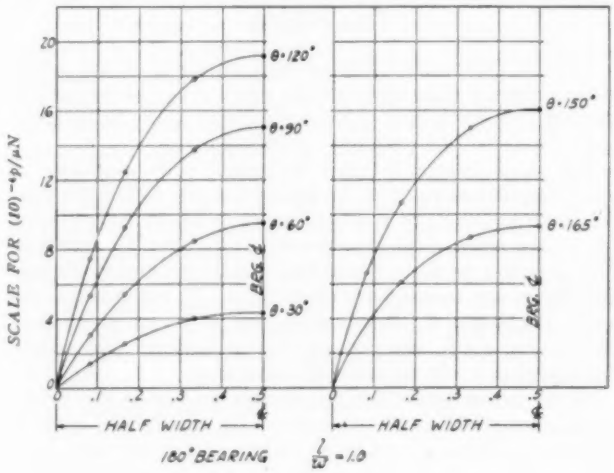


FIG. 13

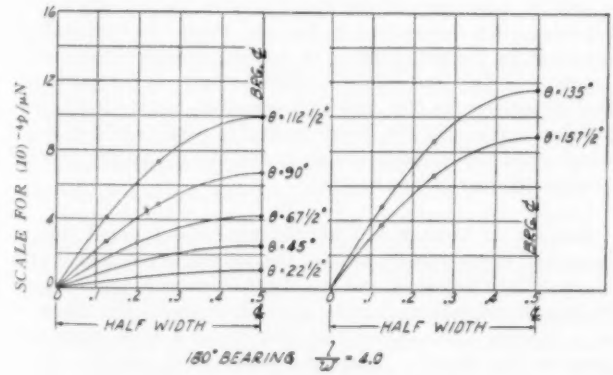


FIG. 14

wide bearing) the curve shows the theoretical values of $\beta/\mu N$, calculated from the equations of Harrison,(10) identical with those of Sommerfeld(7); the experimental results are seen to be very close to the theoretical values. For the remaining values of l/w, the curves are drawn through the experimental points, no theoretical check being available. The curves are drawn with reference to the developed bearing surface.

Fig. 13 shows the experimental values of the pressure p at various values of z and of θ, for the case l/w = 1. Similarly, Fig. 14 shows how p varies with z and with θ for the case l/w = 4. The latter case being a relatively narrow bearing, these transverse pressure curves are very nearly parabolas; this conforms to the hydrodynamical theory, as may readily be shown by making dp/dx = 0 in Equation [1] and integrating the remaining terms.

Fig. 16 shows how the load R is related to the values of l/w, the value of R for the case of no side leakage being taken as unity. The value of R in each case is found by the usual method of resolving the value of β (Fig. 12) with reference to two arbitrary axes, and recombining; the same process yields the direction angle ϕ of the load line. (Figs. 12 and 17.)

Fig. 18 shows the variation of the friction coefficient (tangential resistance at journal surface ÷ load) with variation of l/w.

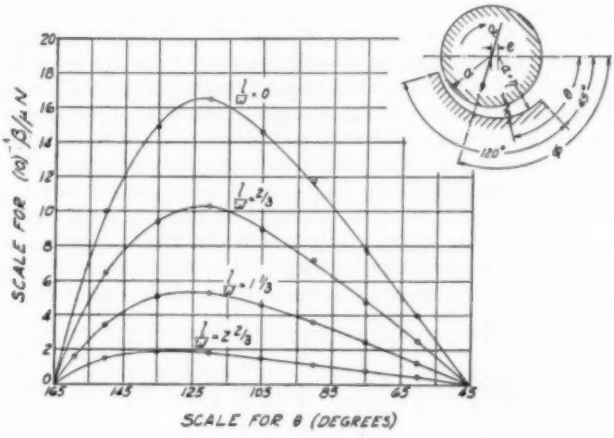


FIG. 15

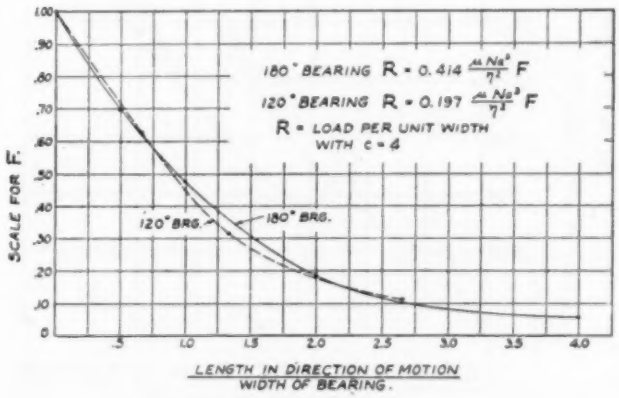


FIG. 16

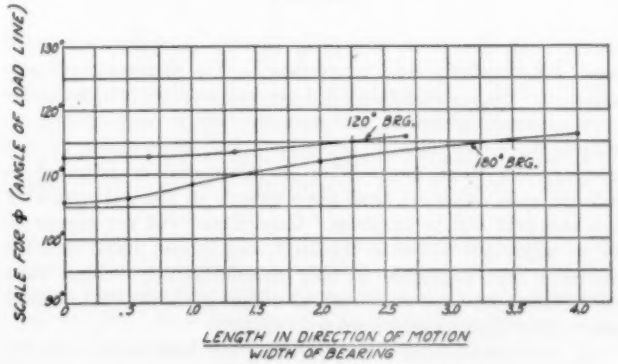


FIG. 17

XIII—120-DEG. BEARING. METHOD III

Fig. 15 shows the assumed arrangement of bearing and journal; the data are otherwise the same as for the 180-deg. bearing. Thus the oil film considered is a part of the same film treated in section XII; it extends from θ = 45° to θ = 165°. The experimental treatment is similar to that described for the 180-deg.

bearing. In Fig. 15, the curve for $l/w = 0$ is derived from the graphical work of Howarth; (11) the circled experimental points show close agreement with the graphical solution. For the remaining values of l/w, the curves are drawn through the experimental values. Fig. 16, 17, and 18 show further derived results, in connection with those for the 180-deg. bearing.

XIV—CONCLUDING REMARKS

The results of the tests of the method indicate a degree of accuracy sufficient for engineering purposes; the experimental results agree very nearly with the theoretical values in all cases where the latter are available, the greatest discrepancy being those shown in Fig. 5 and 5(*a*) where the test results average respectively, 2.2 per cent and 0.9 per cent higher than the values calculated by Michell.

It is probable that a greater degree of precision could be attained by the use of alternating current of high frequency, such as 1000 cycles per second, as used by Kohlrausch(6), and with a vibration galvanometer or a telephone receiver as a null indicator. In the tests described, pointer galvanometers were used, for simplicity and convenience. The direct-current galvanometer has no objectionable features; but the dynamometer-type alternating-current galvanometer, as is well known, is subject to errors due to inductance and capacitance in the circuits, and these must either be eliminated or adjusted to balance each other so that the currents in the galvanometer coil and field will be in phase. Capacitance did not appear to be an important factor in the tests, and it was found that inductance was negligible if only non-inductively wound voltmeters and ammeters were used, as well as in the indirect method where these instruments were not required.

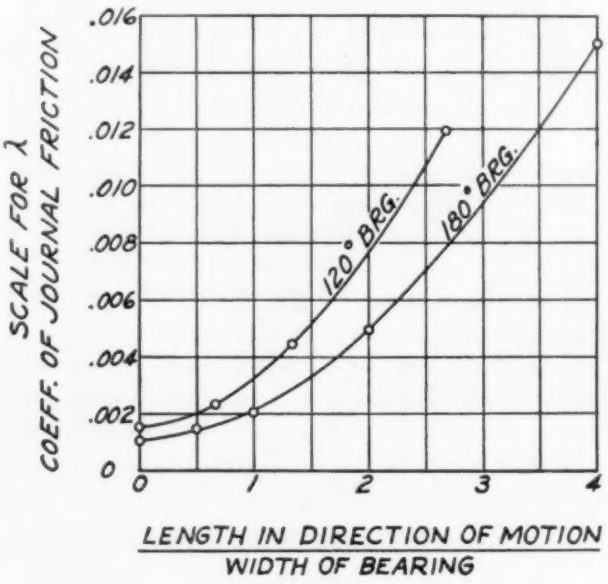


FIG. 18

All the experimental work described has been carried out by Mr. S. J. Needs, including the making of the special apparatus, taking observations, reduction of data, and preparation of the diagrams.

BIBLIOGRAPHY

(1) Osborne Reynolds, "On the Theory of Lubrication," Phil. Trans., 1886.
(2) H. Lamb, "Hydrodynamics."
(3) A. G. M. Michell, "The Lubrication of Plane Surfaces." *Zeitschrift für Mathematik u. Physik*, 1905.
(4) R. O. Boswall, "The Theory of Film Lubrication." Longmans.
(5) Mayo D. Hersey, "On the Laws of Lubrication of Journal Bearings." Trans. A.S.M.E., 1915.
(6) Kohlrausch und Holborn, "Das Leitvermögen der Elektrolyte." Teubner, Leipzig.
(7) A. Sommerfeld, "Zur Hydrodynamische Theorie der Schmiermittelreibung." *Zeit. f. Math. u. Physik*, vol. 50, 1904.
(8) John Perry, *The Electrician* (London), Jan. 26, 1894.
(9) O. Heaviside, "Electromagnetic Theory," vol. I, p. 236.
(10) W. J. Harrison, "The Hydrodynamical Theory of Lubrication." Trans. Cambridge Phil. Soc., vol. xxii, no. iii, 1913.
(11) H. A. S. Howarth, "A Graphical Analysis of Journal-Bearing Lubrication." (Reprint) A.S.M.E., N. Y. 1923–1925.

Discussion

A. G. M. MICHELL.[3] The author's experimental method of solving lubrication problems will be welcomed by all workers in the field, and the fullness of the explanations which he has given of its details will be gratefully appreciated.

The correspondence between the results of the experimental and theoretical methods in the cases which the author has published is gratifying, and it is therefore to be hoped that extended use will be made of the experimental method for the investigation of similar problems not amenable to theoretical treatment. By this means comparisons might be made, for instance, between the merits of thrust-bearing pads with various positions of the axis of tilt, and between journal bearings with wholly converging and with partially diverging films.

As the author points out, the experimental (as is also indeed the mathematical) method is based on the principle of additive volume displacements. The limitations of the principle, however, require to be recognized; for instance, in cases where the momentum of the fluid becomes of importance. For this reason it may be suggested that the momentum of the entering fluid is not adequately represented in the case treated in section X of the paper, by the mere addition of an entry pressure to the viscous equations of flow, but that it involves a change in the pattern of flow throughout the film, making it essentially three-dimensional; in other words, that the kinetic energy is possibly not so much transformed into pressure energy in the film as directly into intensified laminar motion and heat.

ADOLPH A. MOLINE.[4] With the present-day trend toward larger and higher-speed machines, the designer is confronted with many difficult problems, and one of the foremost is the bearings. He is searching continuously for methods to predict the operation of a bearing. The author has contributed a very interesting paper on this subject. There are many items, however, which cannot be considered by the author's method and which affect the bearing operation materially, such as changes in viscosity, thermal distortion of the bearing shell, probable misalignment, and finish of the bearing and journal surfaces.

It would be interesting to know the author's opinion as to the minimum oil-film thickness for slow-speed and for high-speed cylindrical bearings; also what he considers a good value for bearing clearance based on the bearing diameter.

Referring to Fig. 18, it would appear that a ratio of bearing length to bearing diameter of about unity would result in the lowest friction coefficient. No doubt, for safe operation, a slow-speed bearing should have a higher ratio of bearing length to diameter than a high-speed bearing, as the end leakage is greater on a short bearing than on a long bearing, which necessarily results in a thinner oil film. What does the author consider as the minimum ratio of bearing length to diameter?

[3] Canton, Mass.

[4] Power Engineering Department, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.

H. A. S. Howarth.[5] The art of making fluid-film bearings has been very long in its development. It started no doubt when the first hot box occurred in the wheeled war chariot of some prehistoric king. Clay or fat may have been applied to cure the trouble. The bearing parts were surely of wood.

It has taken uncountable centuries to carry the art down to us, through the adoption of metal axles and bearings and the use of animal oils as lubricants until the high-speed machine age was reached.

It is not so long ago that Morin's friction coefficients were the sole dependence of machine designers. He would turn in his grave if he should hear of the low coefficients we encounter today.

Fluid-film bearings, with no metallic contact, were scarcely dreamed of until discovered accidentally by Tower in 1883. Physicist Reynolds was so intrigued by Tower's discovery that he set out to bring physical theory to bear upon the problem. In 1886 Reynolds presented to the small world about him the hydrodynamical analysis of bearing lubrication.

Reynolds' work was abstruse and few were interested. Because modern publicity did not come to its rescue it lay for half a generation buried in the Philosophical Transactions.

In the meantime Mr. Kingsbury, before graduation from Cornell in 1889, had acquired from Professor Thurston an interest in bearings that has followed him to this day. Mr. Kingsbury's most important early published work was that describing his experiments on the air-lubricated journal bearing, in the course of which he was led to several of the conclusions which Reynolds had reached by analytical mathematics.

Before the publication of his experiments Mr. Kingsbury learned of Reynolds' analysis and proceeded to study it thoroughly. He must have been gratified to see how closely each confirmed the other.

Thus fortified by his experimental and mathematical knowledge of the art, Mr. Kingsbury in 1898 made the first construction and tests of the thrust bearing that is now known by his name. But it was his other work in experimental engineering that led to his giving up his teaching of applied mechanics at Worcester in 1901 to devote his time to engineering for the Westinghouse companies.

Once embarked in the business world Mr. Kingsbury stayed therein, and since 1914 he has been concerned solely with the development of his thrust-bearing business. His interest in bearing problems has continued keen, and it was but natural that he should have been selected as chairman of the Lubrication Research Committee of this Society.

Recognizing the importance of the mathematical analysis of Reynolds, Mr. Kingsbury has continually urged engineers to master it. This has led to the further development of the art by others. But no amount of urging will give others the mental capacity to clear away the intricate problems which Reynolds himself had left unsolved.

Although much valuable work has been done by others since Reynolds' time, it has remained for Mr. Kingsbury himself to add the finishing touches to the analytical study of the lubricating film. He has now, figuratively speaking, placed the capitals on the columns, so that construction may now proceed upon a temple dedicated to fluid-film bearings.

L. J. Bradford.[6] The author has outlined an experimental method for determining the pressures existing in the oil film of a bearing and of evaluating the effect of each of the various factors which contribute to this pressure. He has shown that the values obtained by this method agree remarkably well with the theoretical values obtained by applying the equations of Reynolds and Michell to the cases dealt with. There is, however, one other step that should be taken before this method can be accepted with complete confidence. The agreement between the pressures actually developed in the oil film of a bearing and those predicted by the application of the hydrodynamic theory should be demonstrated.

Some years ago Mr. Kingsbury designed and built a machine for investigating the film pressures in a journal bearing $2^1/_2$ in. in diameter and 5 in. long. The essential parts of this machine were a cylindrical sleeve bearing without grooves, lubricated by oil supplied under pressure of 9 or 10 lb. per sq. in. supplied to one end of the bearing. Holes were drilled through the shell of this bearing and connected to pressure gages, thus permitting the pressure in the oil film at the point where the holes entered the bearing to be determined. By rotating the bearing, traverses covering the entire 360 deg. could be made.

Mr. Kingsbury generously loaned this machine to the Pennsylvania State College, and work with it has been going forward for the last two years. Fig. 19 shows a typical curve of pressure found at the center of the bearing, together with a theoretical curve plotted from Harrison's equation, using the theoretical eccentricity. It will be noted that the positive pressure begins on the "on" side at right angles to the direction of the load, and in this case extends for about 140 deg. in the direction of rotation, then drops to a negative pressure of a maximum of about 4 or 5 in. of mercury, then after about 10 deg. rises to a value of about atmospheric, and continues at that value for the remainder of the circumference.

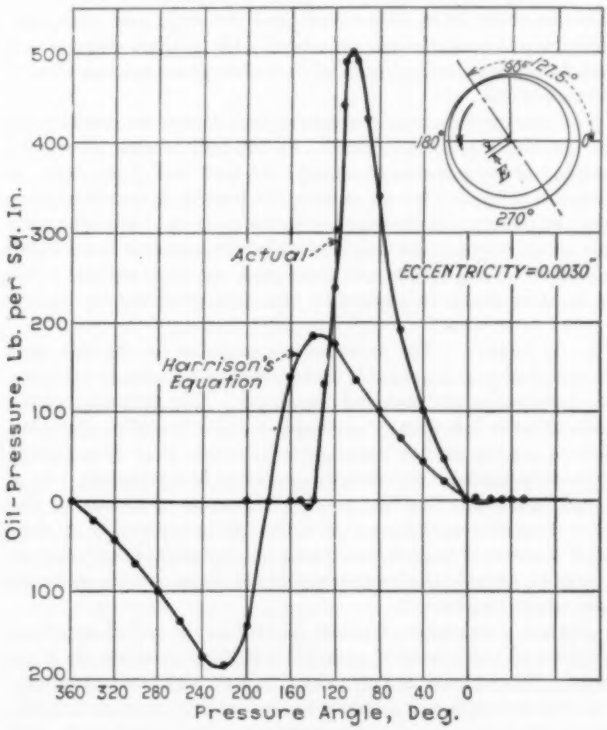


Fig. 19 Typical Curve of Pressure at Center of Bearing (Load = 200 lb. per sq. in. projected area; speed 500 r.p.m.; temperature, 107.5 deg. fahr.)

The pressure distribution actually found differs considerably from that predicted by the theory. This can be ascribed to four causes:

1 Zero absolute pressure is approximately 15 lb. per sq. in.

[5] Vice-President and General Manager, Kingsbury Machine Works, Philadelphia, Pa. Mem. A.S.M.E.

[6] Professor of Machine Design, Pennsylvania State College, State College, Pa. Mem. A.S.M.E.

below atmospheric, and therefore the very low pressures called for by theory are impossible of attainment.

2 The oil is supplied at one end only, and is therefore non-symmetrical with respect to the bearing.

3 There is considerable in-leakage of air at the outlet end of the bearing in the region of negative pressures.

4 Oil carries a considerable quantity of air, both in solution and in suspension. This air is liberated at pressures below atmospheric and serves to break the vacuum that would otherwise be formed. This is shown by the pressure's attaining a negative value of 4 or 5 in. of mercury and then rising to about atmospheric after an interval of only about 10 deg.

All of these factors interrupt the continuity of the oil film, which is a condition essential to the theory.

An effort was made to stop the in-leakage of air by sealing the end of the bearing with a stuffing box so that the space between the journal and bearing would be submerged in oil. Observations made after this was done showed that the eccentricity was decreased about 10 per cent, the line of centers moved in the direction of rotation about 12 deg., the negative pressure's was raised to about 15 in. of mercury, and the angle over which the pressure was positive was increased. All of these changes are such as to bring the experimental curve into closer agreement with the theoretical.

The bearing has been redesigned with a view to profiting by the experience we have gained. In the new bearing oil will be supplied under pressure to a groove at each end of the bearing. This will be equivalent to running the bearing in an atmosphere equal in pressure to that carried in the grooves. We hope with this arrangement to be able to develop the negative loop, which is at present impossible, and thus bring the experimental curve much more nearly in agreement with that predicted by theory.

L. A. Terry.[7] The paper makes available for the first time an electrical analogy suitable for experimental studies of problems of considerable mathematical complexity. The generality of the method, since it is equally applicable to other branches of applied physics, as the author has suggested, makes it of considerable value to individuals not directly interested in lubrication. Thus it will be recalled that the problem of torsion of a shaft of circular cross-section and variable radius can be reduced to an electrical problem of finding the current and potential distribution in a strip of variable thickness much in the same manner as in the lubrication problem.

There are a number of points relative to the technique of application of the described experimental method which are of interest. The pressure at any point x_1, z_1 is obtained experimentally from the summation of pressure due to point (or, more accurately, small-volume) sources at chosen places in the model bath, with current corresponding to the volume flow in the oil film. It is therefore necessary to take a very large number of readings to obtain the pressure distribution for any model. It appears that a much more convenient method would be to cause currents to flow at all points at once in such a way as to fulfil the conditions of the right-hand side of Equation [5], and thereby permit a single set of observations to suffice for the complete determination of the pressure. The currents could be properly adjusted by having a large amount of resistance external to the bath so that any changes in the bath potential at various points during the adjustment of the proper current values would not appreciably affect the flow from an electrode. It would be appreciated if the author would indicate any additional complications this might involve.

Usually one desires to know the total load which can be carried by a bearing and the friction torque or loss. Therefore a very useful addition to the model would be apparatus which would make possible the direct observation of these two quantities.

The principle of superposition has been used freely throughout the paper, and the separate effects of the pressures due to centrifugal action and momentum have been discussed and shown to be, by themselves, negligible. In general, superposition must be used with a good deal of caution due to non-linear characteristics of the complete differential equation. In Section X of the paper, attention has been given to the effects of momentum of the fluid as a whole at a chamfer, in which case the method of superposition seems justifiable. However if the effect of the inertia forces inside the film itself at the chamfer is to be considered, then pressure variations in the y direction must be included in Equation [5], and it is also no longer linear. Likewise the complete equations defining the motion when variable viscosity is considered are non-linear. By considering these effects in a much different manner from that of the present paper, H. Brillié[8] has shown that the resulting forces are indeed far from negligible.

One of the most fundamental requirements for the study of lubrication by experimental models is the necessity of knowing the dimensions of the oil film. The bearing surfaces bounding the film are quite accurately known, within limits of course of manufacturing tolerances. But usually if the relative position of the surfaces is determined analytically, then one has, in more or less convenient form, the additional data required for obtaining the pressure. The relative position can be obtained from the experimental method of solution by trial and error, but may involve a large amount of experimentation. This appears to be a very serious limitation on the general applicability of the developed method of solution.

It is to be noted that the author has compared the results obtained from the experimental method with values calculated on the basis of the classical theory, and therefore not necessarily those obtained in actual bearings. The agreement shown in the paper between the two is noteworthy.

Robert O. Boswall.[9] The author's electrical method of dealing with fluid-film lubrication problems is of particular interest in view of the complicated mathematical and arithmetical processes required to obtain even approximate solutions when the effect of side leakage, viscosity changes, and other relevant phenomena are taken into account.

That the method is capable of giving accurate results is obvious from the close comparison it gives the Michell's data, but the writer regrets that the position adopted for the axis of tilt in the case of the sector-shaped pads does not enable a direct comparison to be drawn between the results obtained by the electrical method and those obtained by mathematical processes and given in his book.

The author's apparatus should be of special use in connection with journal bearings which do not readily lend themselves to mathematical treatment on account of additional complications resulting from surface curvature.

In Fig. 12, which refers to pressure conditions for a bearing subtending an arc of 180 deg., the line of centers is shown in a horizontal position, and the direction of the resultant film pressure obviously points to the necessity of eccentric loading. It would be interesting to know how decrease in viscosity would affect the pressure conditions and whether such a bearing could

[7] A.-C. Engineering Department, General Electric Company, Schenectady, N. Y.

[8] References 72 and 73 in Fifth Annual Report of A.S.M.E. Special Research Committee on Lubrication, presented at Annual Meeting, December, 1930.

[9] College of Technology, Manchester, England.

be centrally loaded, i.e., whether the effect of reasonable changes in the viscosity would enable the line of action of the resultant film pressure to pass through the geometrical center of the brass. There is another point—how does the line of centers get into the horizontal position shown?

When the journal is at rest, the line of centers will be coincident with the line of action of the load, and it is presumed that as the journal commences to rotate and the speed increases, the line of centers will swing round until it assumes a position such as that indicated in Fig. 12.

This point is mentioned as the writer has always experienced difficulty in obtaining perfect film conditions with journal bearings having arcs exceeding about 120 deg., and he has been under the impression that this difficulty can be attributed to the fact that in the early stages of the motion, when the line of centers will only have swung round through a small angle, a relatively large extent of film at the outlet end will be divergent. This leads to the production of negative pressures which reduce the effective length of the film and interfere with the proper motion of the line of centers.

With an arc of 120 deg. this divergence will not be so extensive, and the formation of negative pressures is not likely to have such a serious effect.

With the apparatus described it should be possible not only to determine the motion of the journal center for both central and eccentric loading conditions, but also to ascertain the extent of the negative pressures produced at each stage of the motion. It might be assumed that the viscosity was halved as the lubricant passed between the surfaces, and it would be necessary, of course, to use, say, three eccentricity positions for each position of the line of centers, and then interpolate to obtain the same position for the resultant film pressure in each case.

It must be borne in mind that the apparatus will presumably indicate negative pressures in excess of those actually possible in practice and that there such conditions are indicated the continuity of the film may be expected to break down.

The momentum effect discussed in section X may possibly be of importance as the author suggests, but it would be interesting to know why this should be regarded as a jet effect since at first sight it would appear to be rather a question of viscous flow along a channel with the bearing block as an obstruction and the film opening as a narrow orifice.

In conclusion, the writer would like to take this opportunity of congratulating the author for a very valuable and interesting paper which should do much to stimulate interest in this important and difficult branch of engineering science.

M. Stone.[10] The subject of this paper is one of considerable interest to the engineering profession, and possibly of particular interest to the writer. This latter is because of his active engagement in the application and operation of the author's bearings and because the type of solution that he has resorted to duplicates the solution of a parallel problem[11] that the writer has worked with in the past—the torsion of prismatical bars.

The present high-pressure thrust bearing had its origin in America and Europe practically simultaneously—by the author here and by Mr. Michell abroad. Whereas the author came upon the idea from a physical or experimental point of view (the writer thinks this is true), Michell arrived at the same conclusion from an entirely mathematical approach. By solving the flow and pressure distribution over a rectangular plate, under certain idealized, simple boundary assumptions, Michell came accurately upon the idea of a pivoted surface that is allowed to assume the position of least friction.

However, as the bearing has been finally evolved, Michell's solution is not accurate enough to determine pressure, temperature, and flow distributions, although it did point the way quite definitely to the ideal type of bearing. The indefinite conditions of viscosity and temperature, side leakage, influx and efflux of oil, segmental shape of the present bearing shoe, and the like are obstacles in the way of a purely analytical solution. However, as far as the purely mathematical difficulties are concerned, the author's solution can be said to be satisfactory, but there still remains the indefiniteness of certain boundary conditions which vitiate his solution as well as the purely mathematical one. However, with certain assumed conditions, the author, Mr. Kingsbury, is able to obtain solutions that present insurmountable mathematical difficulties.

Even analytically, it was shown by Michell that the ideal location of the pivot was at the center of pressure of the bearing segment as determined for its optimum position (see Fig. 8, for instance). However, due to practical considerations, which mainly revolve about the hydrodynamical conditions at the leading edge, and are influenced very much by the form of the bevel or rounding of the edge, the center of pressure may actually be located behind the symmetry axis of the segment, instead of ahead. (This is shown in Figs. 8 and 10.) So that, practically, one does not know exactly where to place the pivot, i.e., from the analysis alone. However, for purposes of using a segment in bearings that may operate in either direction, we have always located the pivot on the symmetry axis, practically at the center of gravity of the plate. Tests that have been made here and in Europe do not reveal any change of bearing efficiency by such a change; in fact, it is quite conceivable that such a position is better than the theoretical one for many conditions of operation.

One of the most important problems that still remains unsatisfactory in the operation of these thrust bearings concerns the prediction of bearing temperatures. This involves an accurate knowledge of how and where the heat is generated over the bearing surface, the effect of equalization of temperature through heat conduction in the shoe itself, and dissipation of heat through the coolers, by conduction, etc. At present, the Westinghouse Company is conducting a series of experiments on several field installations of such thrust bearings for large waterwheel generators, and also in the testing department, with the solution of this problem in view. The best idea of heat generation must follow from such a solution as Mr. Kingsbury has carried out.

Concerning the solution itself, when the hydrodynamic differential equation:

$$\frac{d}{dx}\left(\frac{h^3}{12\mu}\frac{dp}{dx}\right) + \frac{d}{dz}\left(\frac{h^3}{12\mu}\frac{dp}{dz}\right) = \frac{U_0 - U_1}{2}\frac{dh}{dx} + V_1 \quad [23]$$

is analyzed physically for a given position of the pressure plate, we obtain a modified Poisson's partial differential equation with a variable right-hand number:

$$\frac{\partial}{\partial x}\left(f_1\frac{\partial \phi}{\partial x}\right) + \frac{\partial}{\partial z}\left(f_1\frac{\partial \phi}{\partial z}\right) = f_2(x,z) \quad [24]$$

where

$\phi(x,z)$ = scalar potential function
$f_1(x,z)$ = variable-flow resistance
$f_2(x,z)$ = strength of sources or sinks.

The left-hand side of the equation is really the divergence of the flow. The right-hand side represents the density distribution of sources or sinks of the flow medium.

[10] Mechanical Engineer, Westinghouse Elec. & Mfg. Co., East Pittsburgh; at present on special investigations in Europe.

[11] "The General Torsion Problem, Solution by Electric Flow Analogy," M. Stone. Third International Congress of Applied Mechanics, Stockholm, August, 1930.

Electrically this means that flow takes place in a two-dimensional plate of geometrical contour similar to the bearing segment, and with variable conductivity, obtained by varying the thickness of the plate. The right-hand side of the equation represents the current density supplied or taken away from the plate. The next important conditions to impose are the boundary conditions, where electric potential is the analogue of pressure. The problem is simple where the function ϕ is zero, but in the case of the effects of oil at the entrance, the boundary pressure must be assumed before the problem becomes soluble.

This same type of problem has been treated in exactly the same way as the author has done, in the previously-referred-to article. For the case of torsion, the general equation can be written:

$$\frac{\partial^2 P}{\partial x^2} + \frac{\partial^2 P}{\partial y^2} = \nabla^2 P = -2cG \quad [25]$$

In the same sense, the left-hand side is the divergence, and $-2cG$ is the flow-intensity distribution. The boundary conditions are P = constant. This equation can be solved directly, in the same manner as Mr. Kingsbury has done, but there always is the necessity of having to supply current at distinct points, which leads to inaccuracy. Equation [25] was thus transferred to an equation of this type:

$$\frac{\partial}{\partial x}\left(f_1 \frac{\partial \phi}{\partial x}\right) + \frac{\partial}{\partial y}\left(f_1 \frac{\partial \phi}{\partial y}\right) = 0 \quad [26]$$

This allows a solution of the problem *without* the necessity of supplying current as Mr. Kingsbury has done. This presents a much simpler and experimentally more desirable procedure. As far as the experimental procedure is concerned, the writer agrees that the electric flow through an electrolyte method (using a.c., however) is probably the easiest and most accurate. A very comprehensive treatment, however, is given of practically all such methods known, by B. Hague,[12] which article should be consulted by any one interested in carrying out such solutions.

Author's Closure

The suggestion is made by Mr. Michell and by Mr. Terry that there may be appreciable pressure and flow effects from accelerations *within the film*. We have at this time neither an adequate theoretical treatment of this question nor any direct experimental data; but approximate calculations indicate that even at the highest speeds occurring in practice, the pressure effects due to accelerations within the film are negligible for practical purposes. The only case thus far solved is that of radial acceleration in thrust bearings, for which an approximate treatment is given in Section IX. The results there given may be reduced to the following formula, applicable to films of any size geometrically similar to Fig. 6, independent of the film thickness and of the viscosity:

$$-p = 1.64 \times 10^{-4} v^2 \text{ (approx.)}$$

in which $-p$ is the mean centrifugal pressure in pounds per square inch, and v is the rim speed in feet per second. Taking v as 200 ft. per sec., $-p$ is 6.6 lb. per sq. in. The mean working pressure for such a bearing would be about 500 lb. per sq. in.; thus it would be affected but 1.3 per cent by the centrifugal action in this case of very great speed. All other internal accelerations appear to be of relatively less effect, as far as can be estimated by approximate calculation. It therefore seems that a fair approximation is obtained by taking the internal pressure effects of local boundary pressures as directly additive, as was done in Section X.

Mr. Moline refers to the vexatious facts that viscosity changes with temperature, and bearing parts are distorted by heat and by pressure. The methods given in the paper are nevertheless applicable, requiring only that the models be made to conform to any assumed conditions, however bad they may be; the main difficulty lies in the infinite variety of conditions, both bad and good.

A categorical statement regarding permissible film thicknesses is not possible. Bearings under test, with great unit pressure and very high speed, may operate satisfactorily with a film thickness which no reasonable estimate would make as great as 10^{-4} in.; but this is far removed from the limitations imposed by the wear that occurs when starting and stopping under load, and by the presence of solid particles in the oil. The current practice of basing bearing areas and loads upon unit pressures is the readiest method of recognizing these latter limitations. In thrust-bearing practice approximate calculations from loads, speeds, and friction indicate minimum film thicknesses ranging from 0.0008 in. to 0.002 in., the smaller value being for low speed and high pressure, the larger value for very high speed and moderate pressure. There is very little exact information on film thicknesses in journal bearings in practice, and on film forms and friction losses from which the thickness might be deduced by calculation.

Theoretical considerations of journal bearings, including side leakage, indicate that if the width of the bearing be fixed at any value not exceeding one and one-half diameters, the bearing-arc length should be approximately equal to the width, for a minimum of power loss by friction; but if the arc length be fixed there is no limit for the increase of the width except that imposed by distortion of the bearing parts by heat and pressure, however this distortion may be assumed to necessitate increasing the minimum film thickness.

Professor Bradford's diagram shows an effective film of only 140 deg. in the complete bearing; this is similar to the condition described by Mr. Boswall as regards the incidence of zero or negative pressures, and other experimenters have found like results. Professor Bradford's further experimental results will be awaited with interest; but the Harrison curve could be approximated experimentally only by absolutely tight sealing of the sides of the film, with an oil supply at not less than atmospheric pressure at the minimum-pressure point.

Negative pressures depend in large part upon the side leakage and the eccentricity ratio. If the minimum film thickness is small compared with the radial clearance, the effective film occupies but a small angle, and the pressure falls to zero at a short distance beyond the point of nearest approach. The conditions shown in Fig. 12 do require unsymmetrical loading, as noted by Mr. Boswall; for an assumed uniform viscosity, the load line cuts the brass about 16 deg. beyond the center, if the width of the bearing is great; reducing the width increases this divergence, as shown in Fig. 17; but on the other hand, increase of temperature of the oil as it passes through has the opposite effect. Practically nothing is known as to temperature range in the film; one cannot be assured that the film temperature is given by the temperature of the metal near the surface. The load would be nearly central in Fig. 12 if we could, for example, assume that the viscosity varied as the cube of the film thickness, but this would imply a viscosity change of 12.6 to 1 in the film, which is far beyond credibility. The conditions in Fig. 12 are substantially those required for a minimum coefficient of friction with a given minimum film thickness; if the load be made central, the minimum film thickness being the same, the load must be reduced about 10 per cent, while at the same time the power

[12] B. Hague, "Experimental Methods of Determining the Distribution of Electric and Magnetic Fields," *Electrician*, Feb., 1929.

loss increases about 12 per cent, and negative pressure is probable near the off-end of the brass. If with central load the minimum film thickness be still further reduced, the negative pressure area extends. This may account in principle for the facts noted by Mr. Boswall, it being inferred that he was working with central loads.

APM-53-6

Stresses in Retaining and Centering Rings for Turbine-Generator Rotors

By R. PATTERSON[1] and D. H. HARMS[2]

In this paper the authors deal theoretically with the distribution of stresses throughout the system of parts enclosing the rotor coil ends of a turbine generator.

This system of parts consists of a disk of uniform cross-section, a cylinder, and a part of the shaft. These parts are fastened together by means of fairly heavy shrink fits and therefore, in the development of formulas for the stresses, the effect of this shrinkage, as well as that of rotation, must be considered.

Since these parts must remain integral for the design considered, the importance of being able to predetermine the speeds at which the various parts may become free is apparent. Therefore formulas were developed for these speeds subject to the boundary conditions which were set up.

An example is given for which the distribution of stresses for two cases is plotted. These two cases are, respectively, for the generator at standstill and at operating speed.

A summary of the significant results is given at the beginning, and a discussion of the main points of interest at the close of the paper.

DESIGNERS of turbine generators are constantly confronted with the demand for increased output and consequently larger units. This increase in size necessarily calls for a more complete analysis of the mechanical problems involved. A great deal of progress has been made in this direction, but a number of features still remain under discussion. One of these, for which the authors will endeavor to develop a solution in the present paper, is that of the distribution of stresses in the rings enclosing the rotor-coil ends and that portion of the shaft beneath the centering ring. In addition to the stresses, it is also important to be able to predetermine the speeds at which the various members of the system become loose from each other.



R. Patterson



D. H. Harms

There are several designs for holding in place the rotor-coil ends of turbine generators, each of which has its advocates and all of which apparently perform satisfactorily. It is not the authors' intention to discuss in the present paper the merits of the various designs, the only one which they analyze being shown in Fig. 1.

The rings referred to are the centering ring and the retaining ring. The centering ring is shrunk on the shaft beyond the rotor coils and provides a fit for one end of the retaining ring, which in turn must carry the centrifugal force of the coil ends when the generator is in operation.

In order to obtain a practical solution for this problem the retaining and centering rings and that portion of the shaft which encloses the coil ends must be considered as a system of parts dependent upon each other rather than as individual parts. Since these parts are fastened together by shrinkage, the solution must have regard to the combination of stresses due to shrinkage and rotation.

Centrifugal forces of the order of 12,000,000 lb. are common values for the retaining-ring and coil ends. These tremendous forces alone, not considering the heavy shrink fits, necessitate high working stresses, and therefore, in order to be assured of adequate factors of safety for known materials, we must know more accurately than ever the stress distribution.

The authors believe, without wishing to raise any controversy, that this is the first time a complete analysis of the problem as set forth has been published, and also that it is one of considerable practical importance to the designers of turbine generators.

Summary of Significant Results

Referring to Fig. 1, the tangential stress in the retaining ring has its maximum value at the fit between the retaining and centering rings.

A change in the thickness of the centering ring does not materially alter this stress.

It is also apparent from the graphs given later that the fatigue range is small.

The bending stress in the retaining ring reaches an appreciable value at zero speed, while the shearing stresses are very small.

With a heavy shrink fit the retaining ring at zero speed may contract to a smaller diameter for approximately one-half its length from the free end than it had in the unstressed state.

The tangential stress in the centering ring is a maximum at the fit between the centering ring and shaft, and this maximum occurs at the operating speed.

The predominating stresses are tangential, with the exception,

---

[1] Mr. Patterson was born in Rensselaer, N. Y., Aug. 8, 1896. He entered the blueprint department of the General Electric Co. as an apprentice draftsman in 1916. Enlisting in the United States Navy in 1917 he served in the aviation branch for the duration of the war. He returned to the General Electric Co. and was employed in the drafting section of the turbine-engineering department; at present is in charge of mechanical calculation pertaining to turbine-generator design.

[2] Mr. Harms was born in Aachen, Germany, Dec. 29, 1897. After serving in the German Navy during the World War he was graduated, receiving an E.E. degree from an institute of technology in 1921. He entered the testing department of the General Electric Co., Dec. 1, 1925. From Sept. 26, 1927, to May 31, 1930, he was employed in the turbine-generator engineering department, after which he left the General Electric Co. to return to Germany. He was employed by the Bergmann Co. in Germany and at present is working in Russia.

Contributed by the Applied Mechanics Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

perhaps, of the radial compression stress in the centering ring, due to shrinkage alone.

The stresses in the shaft at its joint with the centering ring are small, at the operating speed, but the tangential stress at the bore for zero speed may reach an appreciable value.

The following assumptions will be made in order to avoid complications in the solution:

1 The retaining and centering rings are of uniform cross-section
2 The centrifugal force due to the coil ends is uniformly distributed over the retaining ring in the axial as well as in the peripheral direction
3 The length of the shaft has no effect upon the deformation due to shrinkage
4 The axial stresses in the centering ring and the radial stresses in the retaining ring are negligible
5 No force can be transmitted between the parts of the coils resting in the rotor slots and those lying underneath the retaining ring.

Considering first the shaft and centering ring, let u be the

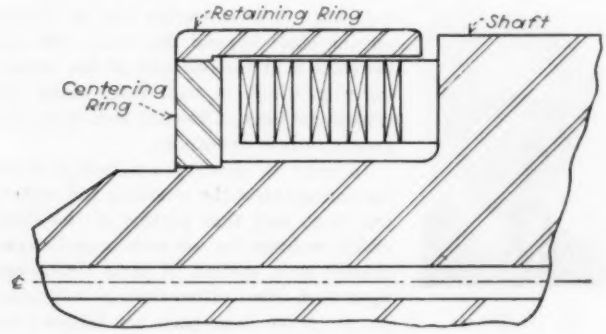


FIG. 1

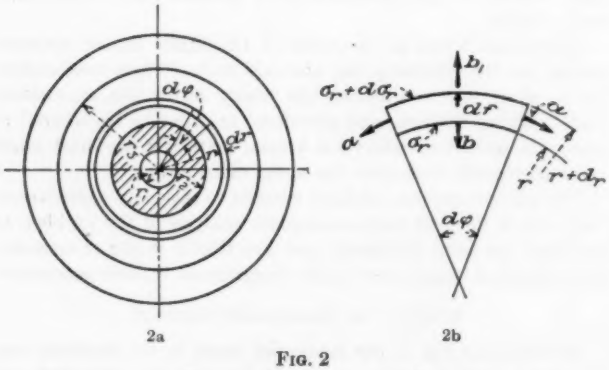


FIG. 2

radial displacement of a point r distant from the center, so that r becomes $r + u$.

The radial elongation is then

$$\epsilon_r = \frac{du}{dr} \quad [1]$$

and the tangential elongation becomes:

$$\epsilon_t = \frac{u}{r} \quad [2]$$

The radial displacement u is to be considered positive if the original radius is increased. We shall furthermore call all forces positive which tend to increase the original radius.

The relation between stress and strain from the theory of elasticity can be written as follows:

$$\epsilon_r = \frac{1}{E}\left(\sigma_r - \frac{1}{m}\sigma_t\right) \quad [3]$$

and

$$\epsilon_t = \frac{1}{E}\left(\sigma_t - \frac{1}{m}\sigma_r\right) \quad [4]$$

where σ_r and σ_t are the radial and tangential stresses, respectively, $1/m$ is Poisson's ratio, and E denotes the modulus of elasticity.

Since all the force shown in Fig. 2 (b) must be in equilibrium, we can write the following equation:

$$b_1 - b + df = ad\varphi \quad [5]$$

where

$$b_1 = (\sigma_r + d\sigma_r)(r + dr)d\varphi \quad [6]$$

$$b = \sigma_r r d\varphi \quad [7]$$

$$a = \sigma_t dr \quad [8]$$

and

$$df = rd\varphi dr \frac{\gamma}{g}\frac{(2\pi rN)^2}{3600r} \quad [9]$$

In Equation [9] γ represents the weight of the material per unit volume, g is the intensity of the earth's field, and N the number of revolutions per minute.

Substituting Equations [6]–[9] in Equation [5] and remembering that $drd\varphi$ is negligible (being a differential of the second order), we obtain:

$$\sigma_r dr d\varphi + d\sigma_r r d\varphi + r^2 \frac{\gamma 4\pi^2 N^2}{3600g} dr d\varphi = \sigma_t dr d\varphi \quad [10]$$

From [3] and [4] we obtain directly:

$$\sigma_r = \frac{mE}{(m^2-1)}(m\epsilon_r + \epsilon_t) \quad [11]$$

and

$$\sigma_t = \frac{mE}{(m^2-1)}(m\epsilon_t + \epsilon_r) \quad [12]$$

Substituting the values for ϵ_r and ϵ_t from Equations [1] and [2] in [11] and [12] gives:

$$\sigma_r = \frac{mE}{(m^2-1)}\left(m\frac{du}{dr} + \frac{u}{r}\right) \quad [13]$$

and

$$\sigma_t = \frac{mE}{(m^2-1)}\left(m\frac{u}{r} + \frac{du}{dr}\right) \quad [14]$$

Differentiating Equation [13],

$$d\sigma_r = \frac{mE}{(m^2-1)}\left(m\frac{d^2u}{dr} + \frac{rdu - udr}{r^2}\right) \quad [15]$$

Calling

$$\frac{\gamma 4\pi^2 N^2}{3600g} \cdot \frac{(m^2-1)}{m^2 E} = q \quad [15a]$$

then, by substituting Equations [13], [14], and [15] in [10], the following differential equation is obtained:

$$r^2\frac{d^2u}{dr^2} + r\frac{du}{dr} - u = -qr^3 \quad [16]$$

the solution of which is

$$u = Ar + \frac{B}{r} - q\frac{r^3}{8} \quad \ldots\ldots\ldots\ldots\ldots [17]$$

This equation gives the elongation in terms of the radius, and calling u_1 and u_2 the elongations of the shaft and centering ring, respectively, we have

$$u_1 = A_1 r + \frac{B_1}{r} - q\frac{r^3}{8} \quad \ldots\ldots\ldots\ldots\ldots [18]$$

and

$$u_2 = A_2 r + \frac{B_2}{r} - q\frac{r^3}{8} \quad \ldots\ldots\ldots\ldots\ldots [19]$$

In the equations that follow, the subscript 1 will always refer to the shaft, 2 to the centering ring, and 3 to the retaining ring.

We shall now investigate the stresses in the retaining ring, a cross-section of which is shown in Fig. 3. Since the difference

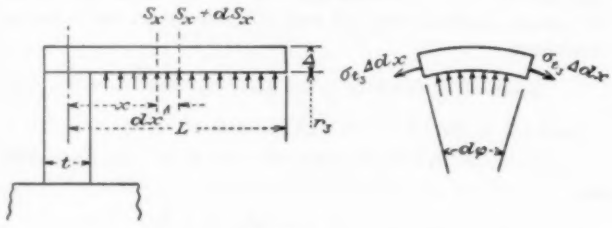


FIG. 3

between the outer and inner radius of this ring is small, we can consider it as a thin ring.

Considering the forces on a ring element $\Delta r_3 d\varphi dx$, we have at the distance x a shearing force S_x and at the distance $x + dx$ the shearing force $S_x + dS_x$. These forces act in the radial direction. In the tangential direction there is a force of $\sigma_{t3} \Delta dx$ acting on both surfaces. The resultant of these two forces is $\Delta\sigma_{t3} d\varphi dx$ acting in the radial direction. Calling the centrifugal force per unit volume due to the ring and its coil load P, we have one more radial force, which is $P \Delta r_3 d\varphi dx$. P is approximately given by the following relation:

$$P = \frac{(W_r + W_c)}{2\pi r_3 L \Delta g} \cdot \frac{4\pi^2 r_3^2 N^2}{3600\, r_3} = \frac{(W_r + W_c)}{L \Delta g} \pi \frac{N^2}{1800} \ldots [20]$$

where W_r = total weight of the retaining ring, lb.
W_c = total weight of the coil load, lb.
L = length of the retaining ring, inches
Δ = thickness of the retaining ring, inches
N = revolutions per minute
g = gravitation constant, inch $\times$ sec.$^{-2}$

Since all the forces on the ring element must be in equilibrium against translational motion, the following equation holds:

$$S_{x+dx} - S_x + P \Delta r_3 d\varphi dx - \sigma_{t3} \Delta d\varphi dx = 0 \ldots\ldots [21]$$

or

$$\frac{dS_x}{dx} + P \Delta r_3 d\varphi - \sigma_{t3} \Delta d\varphi = 0 \ldots\ldots\ldots\ldots [22]$$

Another condition of equilibrium is that the moments on the ring element must vanish. Referring to Fig. 3 and calling counterclockwise moments positive, we obtain, by taking moments about point x,

$$S_{(x+dx)}\, dx + M_{(x+dx)} - M_x + P \Delta r_3 d\varphi \frac{(dx)^2}{2} - \Delta\sigma_{t3} d\varphi \frac{(dx)^2}{2} = 0 \ldots\ldots [23]$$

Neglecting differentials of a higher order, [23] may be reduced to

$$S_x = -\frac{dM_x}{dx} \quad \ldots\ldots\ldots\ldots\ldots\ldots [24]$$

Differentiating [24] and substituting it in [22] gives

$$\frac{d^2 M_x}{dx^2} = P \Delta r_3 d\varphi - \sigma_{t3} \Delta d\varphi \ldots\ldots\ldots\ldots [25]$$

We may now express the value of $\frac{d^2 M_x}{dx^2}$ by the following equation:

$$\frac{d^2 M_x}{dx^2} = \frac{E}{\left(1 - \frac{1}{m^2}\right)} I \frac{d^4 u_3}{dx^4} \quad \ldots\ldots\ldots\ldots [26]$$

It is necessary to use $\frac{E}{1 - \frac{1}{m^2}}$ instead of E because no lateral extension or contraction of the ring element under consideration is possible. The value of I for the ring element becomes

$$I = \frac{\Delta^3 r_3 d\varphi}{12} \quad \ldots\ldots\ldots\ldots\ldots [27]$$

We can also express σ_{t3} in Equation [25] in terms of the elongation u_3:

$$\sigma_{t3} = u_3 \frac{E}{r_3} \quad \ldots\ldots\ldots\ldots\ldots [28]$$

Substituting Equations [26], [27], and [28] in [25], we obtain the differential equation of the retaining ring in the following form:

$$\frac{d^4 u_3}{dx^4} = -u \frac{\left(1 - \frac{1}{m^2}\right) 12}{\Delta^2 r_3^2} + \frac{12}{\Delta^2 r_3^2}\left(1 - \frac{1}{m^2}\right) u_{30} \ldots [29]$$

where u_{30} represents the elongation the retaining ring would have if it were rotating freely at N revolutions per minute. The actual value of this elongation is:

$$u_{30} = \frac{P r_3^2}{E} \quad \ldots\ldots\ldots\ldots\ldots [30]$$

The solution of Equation [29] is:

$$u_3 = \beta_1 \cos \lambda x \cosh \lambda x + \beta_2 \sin \lambda x \sinh \lambda x + \beta_3 \cos \lambda x \sinh \lambda x + \beta_4 \sin \lambda x \cosh \lambda x + u_{30} \ldots [31]$$

λ is determined from the equation:

$$\lambda = \frac{\sqrt[4]{3\left(1 - \frac{1}{m^2}\right)}}{\sqrt{\Delta r_3}} \quad \ldots\ldots\ldots\ldots [32]$$

β_1, β_2, β_3, and β_4 are constants of integration to be determined from the boundary conditions. That [31] is a correct solution of [29] may easily be verified by carrying out the differentiations indicated in [29].

Boundary Conditions

Thus far we have derived the differential equations for the extensions of the shaft, centering-ring and retaining-ring radii. We shall now consider the boundary conditions which have to be imposed upon these equations in order to determine the

constants. Altogether we must have eight conditions, since there are eight constants to be evaluated.

Consider first the boundary conditions of the retaining ring. It is necessary that the moment as well as the shear force must vanish at $x = L$. This condition gives the two equations:

$$\frac{d^2u_3}{dx^2_{(x=L)}} = \frac{d^3u_3}{dx^3_{(x=L)}} = 0 \quad [33]$$

Another condition is that the bending moment at $x = 0$ has to vanish. In actual practice this is not true since the finite width of the centering ring causes a small moment at this point. With our assumptions, however, we may write

$$\frac{d^2u_3}{dx^2_{(x=0)}} = 0 \quad [34]$$

A fourth condition to be fulfilled is that the reaction at the end of the retaining ring has to be equal to the radial pressure

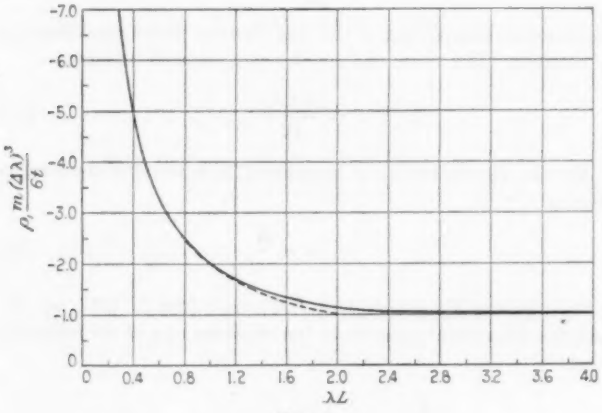


FIG. 4

on the outer periphery of the centering ring. The reaction of a retaining-ring segment of the peripheral length $r_3 d\varphi$ is:

$$S_{(x=0)} = -\frac{E\,\Delta^3 r_3 d\varphi}{12\left(1-\frac{1}{m^2}\right)} \cdot \frac{d^3u_3}{dx^3} \quad [35]$$

Since we have defined a shearing force as positive if acting upward, and a radial stress as negative when compressive, we must have:

$$S_{(x=0)} = \sigma_{r2\,(r=r_3)}\, t r_3 d\varphi \quad [36]$$

By equating [35] with [36],

$$\frac{d^3u_3}{dx^3_{(x=0)}} = -\sigma_{r2\,(r=r_3)} \frac{12\,t\left(1-\frac{1}{m^2}\right)}{E\,\Delta^3} \quad [37]$$

The fifth condition is that the algebraic sum of the extensions of the retaining ring and centering ring at the junction have to be equal to the shrink allowance. The same is true for the centering ring and shaft, giving us the sixth condition. Calling the allowances based on the radii δ_2 and δ_1, respectively, we get the following equations:

$$u_{3\,(x=0)\,(r=r_3)} - u_{2\,(r=r_3)} = \delta_2 \quad [38]$$

and

$$u_{2\,(r=r_3)} - u_{1\,(r=r_3)} = \delta_1 \quad [39]$$

The seventh condition is that the radial stresses of the centering ring and shaft have to be equal to each other at $r = r_3$. Expressing this by an equation,

$$\sigma_{r2\,(r=r_3)} = \sigma_{r1\,(r=r_3)} \quad [40]$$

As the last condition we can state that the radial stress at the bore of the shaft has to vanish. Hence

$$\sigma_{r1\,(r=r_1)} = 0 \quad [41]$$

We have thus stated the eight conditions which are necessary and sufficient for determining the eight constants of integration.

Differentiating [31] twice as indicated by [33], we immediately find that $\beta_2 = 0$. Likewise carrying out the triple differentiation as also indicated in [33] and [37], we obtain the following equations:

$$\beta_1 \tan \lambda L \tanh \lambda L + \beta_3 \tan \lambda L - \beta_4 \tanh \lambda L = 0 \quad [42]$$

$$\beta_1 [\tanh \lambda L + \tan \lambda L] + \beta_3 [\tan \lambda L \tanh \lambda L + 1] + \beta_4 [\tan \lambda L \tanh \lambda L - 1] = 0 \quad [43]$$

and

$$\beta_4 - \beta_3 = \frac{-\sigma_{r2}\, 6t\left(1-\frac{1}{m^2}\right)}{E(\Delta\lambda)_3} \quad [44]$$

Treating these equations simultaneously we obtain:

$$\beta_1 = \sigma_{r2} \frac{6t\left(1-\frac{1}{m^2}\right)}{E(\Delta\lambda)^3} \left[\frac{\sinh 2\lambda L - \sin 2\lambda L}{2 - [\cos 2\lambda L + \cosh 2\lambda L]}\right] \quad [45]$$

$$\beta_3 = -\sigma_{r2} \frac{6t\left(1-\frac{1}{m^2}\right)}{E(\Delta\lambda)^3} \left[\frac{\sinh^2 \lambda L}{\sin^2 \lambda L - \sinh^2 \lambda L}\right] \quad [46]$$

and

$$\beta_4 = -\sigma_{r2} \frac{6t\left(1-\frac{1}{m^2}\right)}{E(\Delta\lambda)^3} \left[\frac{\sin^2 \lambda L}{\sin^2 \lambda L - \sinh^2 \lambda L}\right] \quad [47]$$

Now call

$$\frac{6t}{m(\Delta\lambda)^3}\left[\frac{\sinh 2\lambda L - \sin 2\lambda L}{2 - [\cos 2\lambda L + \cosh 2\lambda L]}\right] = \rho_1$$

or

$$\rho_1 \frac{m(\Delta\lambda)^3}{6t} = \left[\frac{\sinh 2\lambda L - \sin 2\lambda L}{2 - [\cos 2\lambda L + \cosh 2\lambda L]}\right] \quad [48]$$

Equation [48] is plotted on Fig. 4, and we find that for large values of (λL) the trigonometric functions and the constant value 2 may be neglected as compared with the hyperbolic functions, giving

$$\rho_1 \frac{m(\Delta\lambda)^3}{6t} = -1 \text{ (for } \lambda L > 2.2) \quad [48a]$$

For small values we may develop all the functions contained in the foregoing expression into a series, and by neglecting the higher powers obtain the following:

$$\rho_1 \frac{m(\Delta\lambda)^3}{6t} = -\frac{2}{\lambda L} \text{ (for } \lambda L < 1.5)$$

Also call

$$\frac{6t}{m(\Delta\lambda)^3}\left[\frac{\sinh^2 \lambda L}{\sin^2 \lambda L - \sinh^2 \lambda L}\right] = \rho_3$$

or

$$\rho_3 \frac{m(\Delta\lambda)^3}{6t} = \frac{\sinh^2 \lambda L}{\sin^2 \lambda L - \sinh^2 \lambda L} \quad \ldots\ldots\ldots [49]$$

This equation is plotted on Fig. 5, and we find for large and small values of λL, respectively, the following values:

$$\rho_3 \frac{m(\Delta\lambda)^3}{6t} = -1 \text{ (for } \lambda L > 2.2)$$

$$\rho_3 \frac{m(\Delta\lambda)^3}{6t} = -\frac{1.5}{(\lambda L)^2} \text{ (for } \lambda L < 1.5)$$

Likewise call

$$\frac{6t}{m(\Delta\lambda)^3}\left[\frac{\sin^2 \lambda L}{\sin^2 \lambda L - \sinh^2 \lambda L}\right] = \rho_4$$

or

$$\rho_4 \frac{m(\Delta\lambda)^3}{6t} = \frac{\sin^2 \lambda L}{\sin^2 \lambda L - \sinh^2 \lambda L} \quad \ldots\ldots\ldots [50]$$

Equation [50] is plotted on Fig. 6, and we find for large and small values of λL, respectively, the following values:

$$\rho_4 \frac{m(\Delta\lambda)^3}{6t} = 0 \text{ (for } \lambda L > 2.2)$$

$$\rho_4 \frac{m(\Delta\lambda)^3}{6t} = -\frac{1.5}{(\lambda L)^2} \text{ (for } \lambda L < 1.5)$$

For intermediate values of (λL), Equations [48], [49], and [50] must be used.

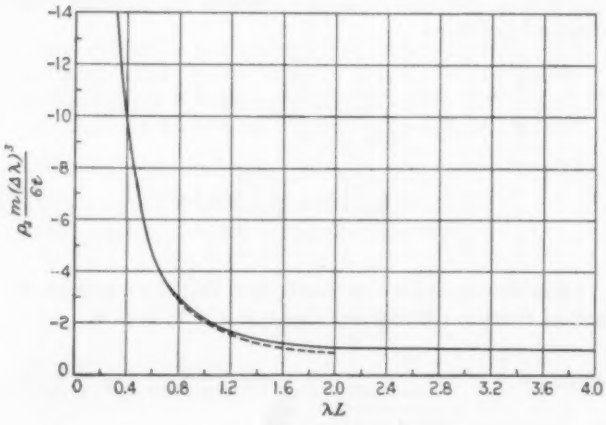


FIG. 5

The present design of turbine generators with which this paper deals has a value of (λL) greater than 2.2. Therefore the constants β_1, β_2, β_3, and β_4 can be written as follows:

$$\beta_1 = -\underset{(r=r_3)}{\sigma_{r2}} \cdot \frac{(m^2-1)}{m^2} \cdot \frac{6t}{E(\Delta\lambda)^3} \quad \ldots\ldots\ldots [51]$$

$$\beta_2 = 0 \quad \ldots\ldots\ldots [52]$$

$$\beta_3 = \underset{(r=r_3)}{\sigma_{r2}} \cdot \frac{(m^2-1)}{m^2} \cdot \frac{6t}{E(\Delta\lambda)^3} \quad \ldots\ldots\ldots [53]$$

$$\beta_4 = 0 \quad \ldots\ldots\ldots [54]$$

There are still four constants to be determined, namely, A_1, B_1, A_2, and B_2. Substituting Equations [18] and [19] in [13] and [14], we are able to write the radial and tangential stresses of the shaft and centering ring as follows:

$$\sigma_{r1} = \frac{mE}{(m^2-1)}\left[A_1(m+1) + \frac{B_1}{r^2}(1-m) - q\frac{r^2}{8}(3m+1)\right] \quad \ldots\ldots [55]$$

$$\sigma_{t1} = \frac{mE}{(m^2-1)}\left[A_1(m+1) + \frac{B_1}{r^2}(m-1) - q\frac{r^2}{8}(m+3)\right] \quad \ldots\ldots [55a]$$

$$\sigma_{r2} = \frac{mE}{(m^2-1)}\left[A_2(m+1) + \frac{B_2}{r^2}(1-m) - q\frac{r^2}{8}(3m+1)\right] \quad \ldots\ldots [56]$$

$$\sigma_{t2} = \frac{mE}{(m^2-1)}\left[A_2(m+1) + \frac{B_2}{r^2}(m-1) - q\frac{r^2}{8}(m+3)\right] \quad \ldots\ldots [56a]$$

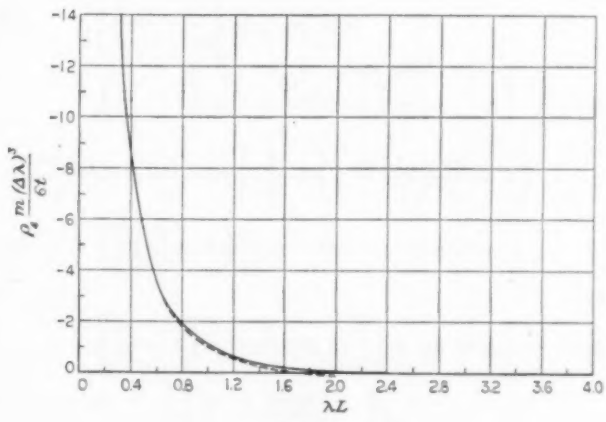


FIG. 6

Now from the boundary conditions which have already been given we can write the following:

For $u_4 - u_2 = \delta_2$ at $x = 0$ and $r = r_3$,

$$\beta_1 + u_{30} - A_2 r_3 - \frac{B_2}{r_3} + q\frac{r_3^3}{8} - \delta_2 = 0 \quad \ldots\ldots [57]$$

and for $u_2 - u_1 = \delta_1$ at $r = r_2$,

$$A_2 r_2 + \frac{B_2}{r_2} - A_1 r_2 - \frac{B_1}{r_2} - \delta_1 = 0 \quad \ldots\ldots [58]$$

Also for $\sigma_{r2} = \sigma_{r1}$ at $r = r_2$,

$$A_2(m+1) + \frac{B_2}{r_2^2}(1-m) - A_1(m+1) - \frac{B_1}{r_2^2}(1-m) = 0 \quad [59]$$

and for $\sigma_{r1} = 0$ at $r = r_1$,

$$A_1(m+1) + \frac{B_1}{r_1^2}(1-m) - q\frac{r_1^2}{8}(3m+1) = 0 \quad [60]$$

We also know that $\sigma_{r2} = \frac{1}{\rho_1}\beta_1$ at $r = r_3$, consequently

$$\frac{\beta_1}{\rho_1} = \frac{B_2}{r_3^2}(1-m) + A_2(m+1) - q\frac{r_3^2}{8}(3m+1) \quad [61]$$

From the foregoing equations the constants A_1, A_2, B_1, and B_2 are found to be as follows:

$$A_1 = \frac{(m-1)}{(m+1)} \cdot \frac{1}{\psi} \left\{ \delta_2 - u_{30} + q\frac{r_3^2}{8}\left[\rho_1\left(1-\frac{r_1^4}{r_3^4}\right)(3m+1) - r_3\left(1+\frac{r_1^4}{r_3^4}\cdot\frac{3m+1}{m-1}\right)\right] + \frac{\delta_1}{2m}\left[m\frac{r_3^2+r_2^2}{r_2r_3} - \frac{r_3^2-r_2^2}{r_2r_3} - (m^2-1)\frac{\rho_1}{r_2}\left(1-\frac{r_2^2}{r_3^2}\right)\right]\right\} \quad [62]$$

where

$$\psi = \rho_1(m-1)\left(1-\frac{r_1^2}{r_3^2}\right) - r_3\left(\frac{m-1}{m+1}+\frac{r_1^2}{r_3^2}\right)$$

$$A_2 = \left(\frac{m-1}{m+1}\right)\cdot\frac{1}{\psi}\left\{\delta_2 - u_{30} + q\frac{r_3^2}{8}\left[\rho_1\left(1-\frac{r_1^4}{r_3^4}\right)(3m+1) - r_3\left(1+\frac{r_1^4}{r_3^4}\frac{3m+1}{m-1}\right)\right] + \delta_1\frac{(m+1)}{2m}\left[\frac{r_2}{r_3}\left(1-\frac{r_1^2}{r_2^2}\right) + \frac{\rho_1}{r_2}(m-1)\left(\frac{r_2^2}{r_3^2}-\frac{r_1^2}{r_3^2}\right)\right]\right\} \quad [63]$$

$$B_1 = \frac{r_1^2}{\psi}\left\{\delta_2 - u_{30} + \frac{q}{8}\left[r_3^2(3m+1)\rho_1\left(1-\frac{r_1^2}{r_3^2}\right) - r_3^3\left(1-\frac{3m+1}{m+1}\cdot\frac{r_1^2}{r_3^2}\right)\right] + \frac{\delta_1}{2m}\left[m\frac{r_3^2+r_2^2}{r_2r_3} - \frac{r_3^2-r_2^2}{r_2r_3} - (m^2-1)\frac{\rho_1}{r_2}\left(1-\frac{r_2^2}{r_3^2}\right)\right]\right\} \quad [64]$$

and

$$B_2 = \frac{r_1^2}{\psi}\left\{\delta_2 - u_{30} + \frac{q}{8}\left[r_3^2(3m+1)\rho_1\left(1-\frac{r_1^2}{r_3^2}\right) - r_3^3\left(1-\frac{3m+1}{m+1}\cdot\frac{r_1^2}{r_3^2}\right)\right] - \delta_1\frac{(m-1)}{2m}\left[1-\frac{r_1^2}{r_2^2}\right]\frac{r_2}{r_1^2}\left[r_3-\rho_1(m+1)\right]\right\} \quad [65]$$

For the large turbine generators which are considered here, the value for (λL) exceeds 2.2, and therefore we can use the simplified value for ρ_1 as found in Equation [48a]. Also the ratio $(r_1/r_3)^2$ is very small compared with unity. Therefore the foregoing constants can be written as follows:

$$A_1 = \left(\frac{m-1}{m+1}\right)\frac{1}{\chi}\left\{u_{30} - \delta_2 + q\frac{r_3^2}{8}\left[\frac{6t}{(\Delta\lambda)^3}\frac{(3m+1)}{m} + r_3\right] - \frac{\delta_1 r_3}{2mr_2}\left[m\left(1+\frac{r_2^2}{r_3^2}\right) - \left(1-\frac{r_2^2}{r_3^2}\right) + \frac{6t}{r_3(\Delta\lambda)^3}\frac{(m^2-1)}{m}\right]\right\} \quad [66]$$

where

$$\chi = (m-1)\left[\frac{6t}{m(\Delta\lambda)^3} + \frac{r_3}{m+1}\right]$$

$$A_2 = \left(\frac{m-1}{m+1}\right)\cdot\frac{1}{\chi}\left\{u_{30} - \delta_2 + q\frac{r_3^2}{8}\left[\frac{6t}{(\Delta\lambda)^3}\frac{(3m+1)}{m} + r_3\right] - \frac{\delta_1(m+1)}{2mr_3}\left[r_2\left(1-\frac{r_1^2}{r_3^2}\right) - \frac{6t}{(\Delta\lambda)^3}\frac{(m-1)}{mr_3^2}\left(1-\frac{r_1^2}{r_3^2}\right)r_3^2\right]\right\} \quad [67]$$

$$B_1 = \frac{r_1^2}{\chi}\left\{u_{30} - \delta_2 + q\frac{r_3^2}{8}\left[\frac{6t}{(\Delta\lambda)^3}\frac{(3m+1)}{m} + r_3\right] - \frac{\delta_1 r_3}{2mr_2}\left[m\left(1+\frac{r_2^2}{r_3^2}\right) - \left(1-\frac{r_2^2}{r_3^2}\right) + \frac{6t}{r_3(\Delta\lambda)^3}\frac{(m^2-1)}{m}\left(1-\frac{r_2^2}{r_3^2}\right)\right]\right\} \quad [68]$$

and

$$B_2 = \frac{r_1^2}{\chi}\left\{u_{30} - \delta_2 + q\frac{r_3^2}{8}\left[\frac{6t}{(\Delta\lambda)^3}\frac{(3m+1)}{m} + r_3\right] + \delta_1\frac{(m-1)}{2m}\left[1-\frac{r_1^2}{r_2^2}\right]\frac{r_2}{r_1^2}\left[r_3 + \frac{6t}{(\Delta\lambda)^3}\frac{(m+1)}{m}\right]\right\} \quad [69]$$

The relations between [66], [67], [68], and [69] are shown by the following three equations:

$$\left.\begin{aligned} A_2 - A_1 &= \frac{\delta_1(m-1)}{2mr_2} \\ B_2 - B_1 &= \delta_1 r_2\frac{(m+1)}{2m} \\ A_1 &= q\frac{r_1^2(3m+1)}{8(m+1)} + \frac{B_1(m-1)}{r_1^2(m+1)} \end{aligned}\right\} \quad [69a]$$

The radical elongation of the retaining ring can now be written as follows:

$$u_3 = u_{30} - \underset{(r=r_3)}{\sigma_{r2}}\frac{6t}{E(\Delta\lambda)^3}\left(1-\frac{1}{m^2}\right)\cos\lambda x e^{-\lambda x} \quad [70]$$

Remembering that $\sigma_{t3} = u_3\frac{E}{r_3}$ and substituting Equations [67] and [69] in [70], the tangential stress in the retaining ring is obtained as follows:

$$\sigma_{t3} = \frac{E}{r_3}\left\{u_{30} + \frac{1}{1+\frac{r_3m(\Delta\lambda)^3}{6t(m+1)}}\left[\delta_2 - u_{30} + q\frac{r_3^3}{4}\frac{m}{(m+1)} + \delta_1\frac{r_2}{r_3}\left(1-\frac{r_1^2}{r_2^2}\right)\right]\cos\lambda x e^{-\lambda x}\right\} \quad [71]$$

From this equation it is readily seen that the maximum tangential stress in the retaining ring is at $x = 0$; that is,

$$\sigma_{t3(\max)} = \frac{E}{r_3}\frac{1}{1+\frac{r_3m(\Delta\lambda)^3}{6t(m+1)}}\left\{\delta_2 + u_{30}\frac{r_3m(\Delta\lambda)^3}{6t(m+1)} + q\frac{r_3^3}{4}\frac{m}{(m+1)} + \delta_1\frac{r_2}{r_3}\left(1-\frac{r_1^2}{r_2^2}\right)\right\} \quad [72]$$

The preceding equations have dealt with the effects due to rotation and shrinkage combined. Now considering the turbine generator at rest, we are left with only the shrinkage effects.

The radial and tangential stresses due to shrinkage alone can readily be written by putting the terms q and u_{30} equal to zero.

Equations [71] and [72] are then, respectively,

$$\underset{(N=0)}{\sigma_{t3}} = \frac{E}{r_3}\frac{1}{1+\frac{r_3m(\Delta\lambda)^3}{6t(m+1)}}\left[\delta_2 + \delta_1\frac{r_2}{r_3}\left(1-\frac{r_1^2}{r_2^2}\right)\right]\cos\lambda x e^{-\lambda x} \quad [73]$$

and

$$\underset{\substack{(x=0\\N=0)}}{\sigma_{t3}} = \frac{E}{r_3} \cdot \frac{1}{1 + \dfrac{r_3 m(\Delta\lambda)^2}{6t(m+1)}} \left[\delta_2 + \delta_1 \frac{r_2}{r_3} \left(1 - \frac{r_1^2}{r_2^2} \right) \right] \ldots [74]$$

However, the stresses in the retaining ring are almost exclusively tangential, due to the heavy shrink fit provided at one end, and the ring takes what has been termed as an "hourglass" shape at standstill. As a result of this, we have a bending and shearing stress which will now be developed, although they are both of much smaller magnitude than the tangential stress.

The bending moment is

$$M_x = \frac{E}{\left(1 - \dfrac{1}{m^2}\right)} I \frac{d^2 u_3}{dx^2} \ldots\ldots\ldots\ldots [75]$$

Therefore the bending stress in the outer fiber is:

$$\sigma_B = \frac{E}{\left(1 - \dfrac{1}{m^2}\right)} \cdot \frac{\Delta}{2} \cdot \frac{d^2 u_3}{dx^2} \ldots\ldots\ldots\ldots [76]$$

Differentiating Equation [31] twice, remembering that β_2 and β_4 are equal to zero, and substituting for β_1 and β_3 their values from [51] and [53], gives

$$\sigma_B = \frac{6tEm}{(\Delta^2\lambda)(m-1)\left[r_3 + \dfrac{6t\,(m+1)}{(\Delta\lambda)^2 m} \right]} \left\{ \delta_2 - u_{30} + q \frac{r_3^3}{4} \frac{m}{(m+1)} + \delta_1 \frac{r_2}{r_3} \left(1 - \frac{r_1^2}{r_2^2} \right) \right\} \sin \lambda x e^{-\lambda x} \ldots\ldots [77]$$

By differentiation, it can be shown that the function $\sin \lambda x e^{-\lambda x}$ has its maximum (for positive values of x) at $\lambda x = \pi/4$. Hence the maximum bending stress occurs at $x = \pi/4\lambda$.

We see from the Equation [77] that since u_{30} is for practical cases always considerably larger than $q \dfrac{r_3^3}{4} \dfrac{m}{(m+1)}$, the bending stress under rotation is always smaller than at rest.

From Equation [24] we know that the shearing force is

$$S_x = -\frac{dM_x}{dx} = -\frac{E}{\left(1 - \dfrac{1}{m^2}\right)} I \frac{d^3 u_3}{dx^3} \ldots\ldots\ldots [78]$$

Hence the average shearing stress becomes

$$\sigma_{s(\text{AVE.})} = -\frac{E}{\left(1 - \dfrac{1}{m^2}\right)} \frac{\Delta^2}{12} \frac{d^3 u_3}{dx^3} \ldots\ldots\ldots\ldots [79]$$

Differentiating Equation [31] three times, remembering that β_2 and β_4 are equal to zero and also that β_3 is equal to β_1, and substituting their values in Equation [79], gives the average shearing stress as follows:

$$\sigma_{s(\text{AVE.})} = -\underset{(r=r_3)}{\sigma_{r2}} \frac{t}{\Delta} e^{-\lambda x} \sqrt{2} \sin \left(\lambda x - \frac{\pi}{4} \right) \ldots\ldots [80]$$

The function $\sqrt{2}\, e^{-\lambda x} \sin \left(\lambda x - \dfrac{\pi}{4} \right)$ has its maximum value (for positive values of x) at $x = 0$ and becomes equal to -1. Then

$$\sigma_{s(\text{AVE.})} = \frac{-mEt}{(m-1)\Delta \left[r_3 + \dfrac{6t(m+1)}{(\Delta\lambda)^2 m} \right]} \left\{ \delta_2 - u_{30} + q \frac{r_3^3}{4} \frac{m}{(m+1)} + \delta_1 \frac{r_2}{r_3} \left(1 - \frac{r_1^2}{r_2^2} \right) \right\} \sqrt{2}\, e^{-\lambda x} \sin \left(\frac{\pi}{4} - \lambda x \right) \ldots\ldots [81]$$

We have already mentioned the importance of being able to predetermine the speeds at which the various members of the system become loose from each other. This will enable us to provide proper allowances, dependent upon the entire system, instead of on merely adjacent parts.

The speed N_3 at which the retaining ring becomes free from the centering ring can be determined by substituting Equations [63] and [65] in [56], by making $\sigma_{r2} = 0$ for $r = r_3$, and remembering that u_{30} and q depend upon the speed N as indicated by Equations [15a], [20], and [30]. Then, calling m (for steel) $= 3.33$ and solving for N_3,

$$N_3 = \frac{10^5}{r_3} \sqrt{\frac{\delta_2^1 + \delta_1^1 \left(\dfrac{r_2^2 - r_1^2}{r_3^2} \right)}{9.48\gamma^1 - 0.47}} \ldots\ldots\ldots [82]$$

where

$$\delta_2^1 = \frac{1000\delta_2}{r_3} = \text{allowance in mils per inch of diameter}$$

$$\delta_1^1 = \frac{1000\delta_1}{r_2} = \text{allowance in mils per inch of diameter}$$

and

$$\gamma^1 = \frac{W_r + W_c}{2\pi r_3 L \Delta} = \gamma \left(1 + \frac{W_c}{W_r} \right) (\gamma = 0.283)$$

Equation [82] gives the correct value, provided the centering ring has not yet started to separate from the shaft. For such a condition, the value would have to be modified to the extent that Equation [39] is fulfilled.

The speed N_2 at which the centering ring becomes free from the shaft can be determined by setting σr_2 at $r = r_2$ in Equation [56] equal to zero, and substituting Equations [63] and [65] in [56].

Solving for N_2, we obtain

$$N_2 = \frac{10^5}{r_3} \sqrt{\frac{\delta_2^1 + \delta_1^1 \left[0.65 \dfrac{r_2^2}{r_3^2} + 0.35 + 2.73 \dfrac{t}{(\Delta\lambda)^2 r_3} \left(1 - \dfrac{r_2^2}{r_3^2} \right) \right]}{9.48\gamma^1 + 0.305 \left[1 + \dfrac{20t}{r_3(\Delta\lambda)^2} \left(1 - \dfrac{r_2^2}{r_3^2} \right) - 2.53 \dfrac{r_2^2}{r_3^2} \right]}} \ldots [83]$$

It must be remembered here that limitations apply to Equation [83] similar to those already explained for Equation [82].

Numerical Example

Calling

$r_1 = 2.5$ $L = 18.0$ $N = 1800$ $\delta_1 = 0.005$ $\delta_1^1 = 0.385$
$r_2 = 13.0$ $\Delta = 1.75$ $W_r = 1400$ $\delta_2 = 0.040$ $\delta_2^1 = 1.667$
$r_3 = 24$ $t = 2.5$ $W_c = 1500$

the following values are obtained:

$u_{30} = 0.02576$ $\chi = 263.943$ $A_1 = 0.0000102$ $B_2 = 0.04236$
$q = 0.79056 \times 10^{-6}$ $(\Delta\lambda)^3 = 0.0418$ $A_2 = 0.000145$
$\lambda = 0.1983$ $\gamma^1 = 0.585$ $B_1 = 0.00010$

The stresses are plotted in Figs. 7 and 8.

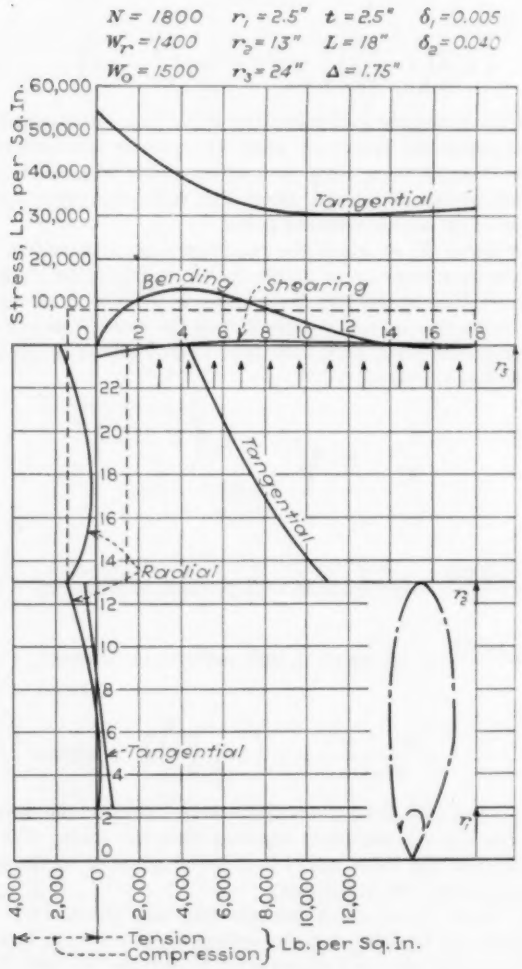


FIG. 7 EXAMPLE A
(N = 1800; Wr = 1400; W_c = 1500; r_1 = 2.5 in.; r_2 = 13 in.; r_3 = 24 in.; t = 2.5 in; L = 18 in.; Δ = 1.75 in.; δ_1 = 0.005; δ_2 = 0.040.)

DISCUSSION OF THE RESULTS

Considering first the tangential stresses in the retaining ring, we may learn from Equations [71] and [72] the following interesting facts:

σ_{t3} has its maximum value at ($x = 0$) and for large values of x it approaches the value which would be obtained if the ring were rotating freely. The tangential stress at any distance x is given by the sum of the latter value and a superimposed stress which varies as a damped trigonometric function as we proceed in the x-direction. This superimposed stress consists of four terms, two of which are proportional to the shrink allowances between the centering ring and shaft and between the retaining and centering rings. The third term depends upon the centrifugal force exerted by the centering ring, which is indicated by the factor qr_3^3, and the fourth term is proportional to u_{30} (the elongation of the retaining ring if it were rotating freely). In calculating an example it will be found that the value of δ_2 contributes the largest amount to the maximum tangential stress in the retaining ring. Any change that may be made in the thickness t of the centering ring does not materially alter this stress.

We shall note at this place that from Equation [72] it might appear that the maximum tangential stress in the retaining ring would reach a finite value for an infinitely thin ring. It is to be remembered, however, that for such thin rings the approximations no longer apply. The value of λx in Equation [71] for this instance is indeterminate, so that the expression $\cos \lambda x \varepsilon^{-\lambda x}$ is no longer unity for $x = 0$. For small distances of x, Equation [71] already gives the value $\sigma_{t3} = \infty$, as it has to be for infinitely thin rings.

The tangential stresses in the retaining ring at standstill are given by Equations [73] and [74]. By inspection, we learn that the tangential stress at zero speed becomes zero for large values of x, as was naturally to be expected. The maximum stress, which depends only on the shrink allowances δ_1 and δ_2, occurs at $x = 0$.

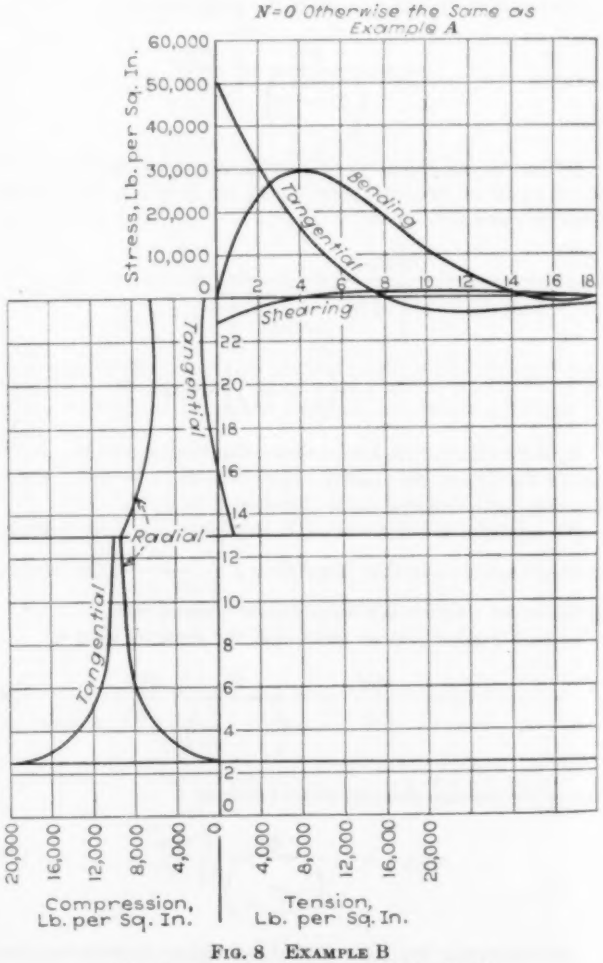


FIG. 8 EXAMPLE B
(N = 0; other values as in Example A, Fig. 7.)

Examples *A* and *B*, showing the stresses at normal and zero speeds, are plotted on Figs. 7 and 8, respectively. Comparing these examples, it is found that the maximum tangential stress in the retaining ring has a fatigue range of less than 10 per cent of the maximum stress at zero speed. It is also apparent that the shrink allowance affects only about one-half of the retaining ring. At standstill, rings of this class actually contract to a smaller diameter than they have in the unstressed state.

The maximum bending stress in the retaining ring occurs at $x = \pi/4\lambda$, and is considerably larger at zero speed than at normal speed. At this point the bending causes compression in the outer and tension in the inner fibers of the ring.

In order to get a true conception of the total stresses in the outer and inner fibers of the ring we must use the following well-known relation:

$$\left.\begin{aligned} \sigma &= \sigma_t - \frac{1}{m}\sigma_B \\ \text{or} \\ \sigma &= \sigma_B - \frac{1}{m}\sigma_t \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots [84]$$

the equation to use being the one which gives the largest value for σ. In Equation [84] tensile stresses have to be considered as positive and compressive stresses as negative. From that equation we learn that the reduced stresses in the outer fiber are larger, and in the inner fiber are smaller, than the tangential stress σ_t.

The average shear stress, whose magnitude is always small, has its maximum value at $x = 0$, and decreases to zero for large values of x. The shear stress is parabolically distributed over the cross-section and is zero at the inner and outer fibers.

The tangential and radial stresses in the shaft and centering ring may be calculated from Equations [55], [55a], [56], and [56a], respectively, where the constants A_1, B_1, A_2, and B_2 have a definite relation to each other as indicated by Equation [69a]. It is therefore only necessary to calculate B_2 from [69], when the remaining three constants may be obtained from their relations in [69a].

From the example given, the tangential stress in the centering ring at zero speed has a positive value at the bore, passes through zero, and becomes negative at the retaining-ring fit. The position of this curve with respect to its ordinate is largely determined by the shrink allowances δ_1 and δ_2. A slight decrease of the allowance δ_2 would put the entire ring in tension as it is at the normal speed. The radial stress at the bore of the centering ring is negative at zero speed, decreases with increasing velocity, and becomes of course zero when the separating speed is reached. Beyond this speed the equations developed here no longer apply because the boundary conditions which have been used for the determination of the integration constants change.

The tangential stress in the shaft at normal speed is negative at the shrink fit and positive at the bore, while at standstill it is negative throughout. The radial stresses in this portion of the system are of course zero at the bore. At normal speed this stress is positive for small and negative for large values of r. At zero speed the radial stress in the shaft is negative, and increases from zero to a maximum at the shrink fit.

Equation [82] gives the separating speed between the retaining and centering rings. It is to be noted that this speed is independent of the thickness and length of the retaining rings, as is to be expected.

From Equation [83] the separating speed between the centering ring and shaft may be calculated. In this case we learn that the dimensions of the retaining ring have quite an influence upon the separating speed. By choosing definite values of N_2 and N_3 it would also be possible to solve for the shrink allowances $\delta_1{}^1$ and $\delta_2{}^1$. One would then find that by increasing N_2 with a constant value for N_3 the shrink allowance $\delta_2{}^1$ between the retaining ring and the centering ring could be reduced, which would in turn reduce the maximum tangential stress in the retaining ring. It must be remembered, however, that this would at the same time increase the tangential stress in the centering ring.

Discussion

C. Richard Soderberg.[3] The authors have presented an interesting discussion of a problem which is of utmost importance to the designers of large, high-speed turbine generators. After reviewing the paper somewhat hastily the writer wishes to comment on the following points:

In treating the deformation of the retaining ring, the well-known theory of a beam on an elastic foundation may be used directly, obviating most of the complicated mathematics of the paper. Considering an element of unit circumferential extension of the ring, the differential equation for the radial deformation is easily found to be

$$\frac{EI}{1 - \dfrac{1}{m^2}}\frac{d^4u_3}{dx^4} + ku_3 = 0 \quad \ldots\ldots\ldots\ldots [85]$$

where $EI/(1 - 1/m^2)$ is the flexural rigidity of the element and k is the modulus of the foundation. For a thin ring of mean radius r_m and thickness Δ, k is easily shown to be

$$k = E\frac{\Delta}{r_m{}^2} \quad \ldots\ldots\ldots\ldots [86]$$

The solution of the differential equation [85] for all possible types of boundary conditions is already available[4] in the engineering literature. The constant term in the differential equation [29] is easily disposed of by a linear change in the variable u_3. Obviously, this criticism concerns only the method of arriving at the results; it does not in any way detract from the value of the final results obtained by the authors.

There is another aspect of the retaining-ring problem which is not mentioned by the authors, but which is of great importance in constructions somewhat different from the one illustrated in Fig. 1. This is the bending reactions transmitted by the centering ring and the retaining ring in those cases where the latter is shrunk on both the rotor body and the centering ring. This construction has been used by several foreign and American firms, and troubles due to movements on the shrink fits have been encountered in several instances. The reason for these troubles is that the bending moment on the rotor is almost wholly taken up by the very stiff ring structure. The arrangement shown in Fig. 1 would certainly obviate this trouble, but other satisfactory ones have also been proposed and are in use.[5] Fig. 9 shows a construction which is sponsored by the company with which the writer is associated. The retaining ring is shrunk on at A and B, but the "centering ring" is entirely free from the shaft. This permits the passage of a maximum of cooling air at C under the centering ring, and obviates all bending reactions. This construction has proved to be very satisfactory.

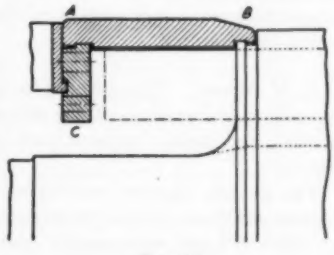


Fig. 9[6]

The writer must confess to a certain disappointment at the

[3] Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa. Assoc-Mem. A.S.M.E.

[4] See Timoshenko and Lessells, "Applied Elasticity," p. 133 et seq.

[5] J. A. Kuyser, *Jl.A.I.E.E.* 1929, "Recent Developments in Turbo Generators."

[6] S. L. Henderson and C. R. Soderberg, A.I.E.E. 1928, "Recent Developments in Turbine Generators." Fig. 9 above is a reproduction of Fig. 29 of this paper. See also Figs. 28 and 30 of same paper.

complete absence of any discussion on the important subject of working stresses in turbo retaining rings. No exaggeration is involved in the statement that these working stresses frequently influence the size of the entire machine. In general, these working stresses were established before the knowledge regarding the actual distribution of the stresses was very complete, and every new contribution to the subject brings the actual maximum stress closer and closer to the yield point of the material. In Fig. 7 of the paper the maximum and mean tangential stresses of the retaining ring at 1800 r.p.m. are given as approximately 54,000 and 36,000 lb. per sq. in., respectively. The corresponding figures for 20 per cent overspeed would be considerably greater. Any one who is familiar with the design of large turbine generators will see nothing extraordinary in these figures, but it would be interesting, nevertheless, to have the authors' comments on this important question.

S. TIMOSHENKO.[7] This paper contains a solution of a very important practical problem. It seems to the writer that the solution can be simplified very much by using the well-known theory of bending of bars on an elastic foundation, together with the numerical tables of functions entering in this theory and which have been calculated by A. Zimmermann in "Die Berechnung des Eisenbahn Oberbaues," Berlin, 1888. (See also Hayashi, "Theorie des Trägers auf elastischer Unterlage," Berlin, 1921.)

In designing a retaining ring it is desirable to begin with the determination of the overspeed at which the ring will become loose. At this speed of separation there are no forces acting between the retaining and the centering rings, and there is no difficulty in calculating, by known formulas, the expansion of both rings under the action of centrifugal forces. Knowing the difference between the radial expansions of the two rings, the reactive forces between the rings necessary to remove this difference can easily be calculated by using the tables for beams on an elastic foundation. This manner of calculation will give in a very simple manner a comparison of the stresses of various initial shrink-fit pressures.

A. M. WAHL.[8] The authors have given a very complete and comprehensive analysis of the stress distribution in a very important and highly stressed part of modern turbine-generator rotors.

The general theories underlying the calculation include the well-known theory of a bar on an elastic foundation, first developed by Winkler,[9] and subsequently elaborated on by Zimmerman,[10] Hayashi, Wieghardt, Timoshenko, and others. This theory was applied by Cook[11] to the calculation of a pipe reinforced by steel rings, a case which is essentially the same as that of a retaining ring shrunk on to the centering ring. In addition, the authors apply the well-known theory of deformations of thick rings, often referred to as the Lamé theory. Nevertheless, although the underlying theory is well known, they deserve much credit for bringing the whole solution together in a unified form and determining the integration constants in such a way as to facilitate practical application of the formulas.

The writer wonders whether or not the solutions found in the literature for the various components of the retaining-ring assembly could not be superimposed, using the principle of superposition, and in this manner a solution could be arrived at without the necessity of carrying out the various integrations as was done in the paper. Possibly this might be a fairly simple method of attack.

In order to determine the true factor of safety, it is necessary to combine the various stresses (radial, tangential, and axial) according to certain theories of strength. It should be noted that the authors use the maximum-strain theory for this purpose. The practice of the Westinghouse Company is to use the maximum-shear theory. The use of the latter will give a somewhat more conservative design: i.e., the equivalent stresses as figured will be greater than will be the case if the maximum-strain theory is used. Furthermore, recent tests by Nadai, Lode, Ros, and Eichinger, and others seem to indicate that for ductile materials the maximum "energy-of-distortion" theory is more nearly in agreement with the truth than the maximum-shear theory as far as yielding of the material is concerned. Nevertheless it was also found that the maximum-shear theory is not far from the truth and is on the safe side. For this and other reasons the writer believes that the maximum-shear theory is more consistent for practical use where ductile materials are involved than is the maximum-strain theory used by the authors. It should be noted that for the example considered by the authors where one principal stress is much greater than the others, the difference between the two theories will not be great. There are other cases, however, where a much more pronounced difference will exist.

M. STONE.[12] The authors have presented a clear and accurate solution of the mechanical action of an important element of the turbine-generator. They are probably correct in stating that this is the first time that such an analysis has been published.

However, for some five years now, such a solution as the authors published has been used at the Westinghouse Co., with variations that have to do with the slightly different mechanical construction employed. The problem there has usually been one in which the retaining ring was (1) either shrunk on on both ends—on the centering rings and on the rotor, or (2) the same as (1), with the centering ring free—i.e., not shrunk on the shaft.

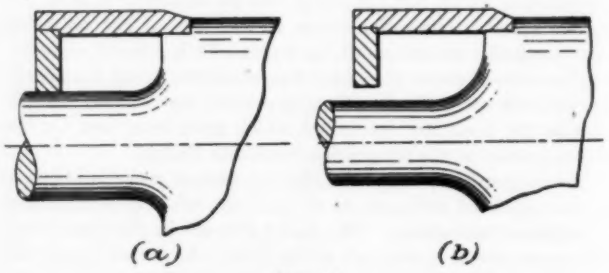


FIG. 10

Referring to Fig. 10, in case (*a*) we have made measurements on the form of the shrunk-on retaining ring—which takes on an "hour-glass" shape, in which the diameter in the middle was considerably less than the original machined diameter. This fact, it is understood, was predetermined analytically and agrees with the authors' statement on page 8.

The authors have gone to some length in developing the fundamental formulas [17] and [29]—which, however, may be taken directly from Timoshenko and Lessells' book,[13] when it is realized that the retaining ring is virtually *a beam on an elastic foundation*.

The general solution of this fourth-degree equation may be expressed through the trigonometrical functions, as Dr. Timoshenko has done, or by hyperbolic functions, as the authors have

[7] Professor of Mechanical Engineering, University of Michigan, Ann Arbor, Mich. Mem. A.S.M.E.

[8] Research Engineer, Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa. Jun. A.S.M.E.

[9] "Die Lehre v.d. Elastizität u. Festigkeit," Prague, 1867, p. 182.

[10] "Die Berechnung des Eisenbahn Oberbaues," Berlin, 1888.

[11] *Engineering*, vol. 116 (1923), p. 479.

[12] Mechanical Engineer, Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. At present on special investigations in Europe.

[13] Timoshenko and Lessells, "Applied Elasticity," Equations 313 and 99, respectively.

done. In our own work, we have prepared tables of evaluated functions—from which any arbitrary boundary conditions may be satisfied.

A remark must be made which has to do with the actual importance of such a solution as the authors have published. Since it is at once appreciated that the highest stresses are the tangential stresses existing in the retaining ring, where it is shrunk on the centering ring or the rotor, it is only necessary to calculate this simple stress due to the shrink fit. If further cognizance is taken of the proper remark of the authors, "that a change in the thickness of the centering ring does not materially alter the stress," then, within the same limits of approximation, these simple stresses are close enough to the maximum to render unnecessary the more involved calculations. (Compare 50,000 and 54,000 lb. per sq. in. in the paper.) The speed at which the retaining ring will lose contact with the centering ring or the rotor body is just as simply calculated, so that in actual designing, only these latter calculations need be made.

Authors' Closure

The authors wish to thank those taking part in the discussion for their just and constructive criticism. One of the points brought out was that the theory of a beam on an elastic foundation, as first developed by Winkler in 1867 and so ably presented by Dr. S. Timoshenko in his recent book on elasticity, could have been applied, once it was realized that the retaining ring is virtually a beam on an elastic foundation. This is obviously true—after one has reviewed the two methods, and the authors are gratified to find that the results obtained by their method, which to them was original, are corroborated by those obtained by using other methods.

Mr. Soderberg mentioned another aspect of the retaining-ring problem; that is, that of considering the free end of the retaining ring as shown in Fig. 1 shrunk upon the rotor body, thereby forming a rigid extension to the rotor body at each end. This type of design was not mentioned in the paper, since the company with which one of the authors is still associated discontinued the design mentioned about eight years ago. The design then sponsored was the one shown in Fig. 1, which performs satisfactorily and eliminates the possibility of any movement at the shrink fits, as pointed out by Mr. Soderberg.

The absence of consideration of working stresses in the paper was not due to any underestimation of their importance, but rather to the fact that the authors had no thought that this question would enter into the discussion. It is generally accepted that the choice of proper working stresses depends not only upon the practical designer and his particular design, along with the results of the theory of elasticity, but also upon the metallurgist and the materials used. Modern practice has accumulated much empirical data on the basis of previous experience, and it is the opinion of the authors that the employment of such information, in conjunction with the results of qualitative analysis, will lead to a sound policy in the selection of proper working stresses.

Mr. Soderberg's statement that the maximum stress of 54,000 lb. per sq. in. shown in Fig. 7, and the mean stress which he estimates at 36,000 lb. per sq. in., would be considerably greater at 20 per cent overspeed, might possibly be misinterpreted. While the lower stresses do increase considerably with overspeed, it was pointed out in the paper that the maximum stress in the retaining ring is due chiefly to the shrink fit, and rises from 50,000 lb. per sq. in. at zero speed, to 54,000 at 100 per cent speed. At 20 per cent overspeed this maximum stress would rise only something like 2 per cent, which would be barely considerable. The other stresses of course never exceed the maximum, and remain less until an overspeed is reached that will break the contact between the retaining and centering rings.

Dr. Timoshenko's statement that it is desirable to start with the separating speeds in designing retaining rings, is very well put. It is only in this way that proper shrink fits can be determined that will insure satisfactory operation. His suggestion of calculating independently, by known formuals, the expansion of both rings due to centrifugal force, and of adjusting their differences to obtain the reactive forces between the rings, was in practice while the knowledge of the actual distribution of stresses was yet incomplete.

Mr. Wahl prefers the maximum-shear theory for determining the true factor of safety rather than the maximum-strain theory, as used by the authors, although he points out that the difference between the results obtained by the two theories will not be very great in this particular problem. Several well-known theories have been advanced from time to time, all of which are of an empirical nature. The authors chose the maximum-strain theory for the reason that of the different theories, one is nearest to reality for brittle metals, and another for very ductile metals, while metals intermediate in ductile qualities may be looked upon as following an intermediate empirical formula which may be devised to suit any individual case. From the notable experiments carried out by Cook and Robertson[14] on thick cylinders, it is possible to conclude that the maximum-strain theory overestimates the true elastic strength for very ductile metals, while the maximum-shear theory underestimates it by about 25 per cent.

Mr. Stone's discussion is essentially covered by the authors' replies to Mr. Soderberg. Mr. Stone also points out, however, that it is only necessary to calculate the maximum stresses for every day practice, which is a simple matter, and with which the authors agree.

[14] *Engineering*, Dec. 15, 1911.

APM-53-7

Determination of Stresses in Rotating Disks of Conical Profile[1]

By F. C. RUSHING,[2] EAST PITTSBURGH, PA.

In this paper a determination is made of the accuracy of stress-distribution curves obtained by the application of Donath's "Sum and Difference Curves" to rotating disks of conical profiles.



ROTATING disks of conical profiles (Fig. 1) are used rather extensively on account of their economy in the use of material and the facility with which they are produced. A rotating disk in which the stress is equally distributed is relatively thick near the axis and thin at the rim; the variation in the thickness of such a disk is not constant, however, and it is difficult to machine the sides to the exact shape. Since a conical profile approximates the profile necessary for equal stress distribution, and since a disk with straight sides is easily machined, such a disk is commonly used.

In calculating the stresses in conical disks, use is made of approximate methods whose accuracies have never been determined. One of the most extensively used approximate methods is an application of Donath's "Sum and Difference Curves."[3]

Fortunately, an exact method of calculating stresses in conical-profiled disks has been published by Mr. H. M. Martin.[4] The chief purpose of the present paper is to determine, with the use of Mr. Martin's exact method, the accuracy of the results obtained by the approximate application of Donath's "Sum and Difference Curves" to disks of conical profiles. In accomplishing this purpose, it is deemed desirable to include here a brief description of the derivation of each method, together with an example of its application; followed by a comparison of results obtained by the application of the two methods to a wide variety of shapes and sizes of conical disks.

[1] The material presented in this paper was used in a thesis by the author for an M.S. degree from the University of Pittsburgh.

[2] Westinghouse Electric & Manufacturing Co. Mr. Rushing was born in 1906 in Runge, Texas, where he lived until he entered the University of Texas in 1924. He received a B.S. degree from this university in 1928, and became an employee of the Westinghouse Elec. & Mfg. Co. shortly thereafter. As an employee of this company, he was permitted to take the design training course, which enabled him to attend the University of Michigan during the Summer session of 1929 and to receive also some graduate credit from the University of Pittsburgh. He received an M.S. degree from the University of Pittsburgh in 1930. At the present time he is in the design section of the Industrial Motor Engineering Department of the Westinghouse Company.

[3] Developed by M. Donath in "Die Berechnung rotierender Scheiben und Ringe" (Berlin, G. Springer, 1912), and described by H. Hearle in "The Strength of Rotating Discs," *Engineering*, vol. cvi, Aug. 9, 1918, pp. 131–134.

[4] "Stress Distribution in Rotating Disks of Conical Profile," *Engineering*, vol. cxv, Jan., 1923, p. 1.

Contributed by the Applied Mechanics Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

NOTATION

- x = radial distance from the axis
- y = thickness of the disk at the distance x
- σ_r = radial stress in pounds per square inch
- σ_t = tangential stress in pounds per square inch
- μ = mass density of material of the disk
- ω = angular velocity of rotation
- $\frac{1}{\nu}$ = Poisson's ratio
- ξ = radial displacement at the end of radius x
- R = radius at which extended sides would meet
- D = $2R$
- Y = thickness at center if sides were extended to center
- N = revolutions per minute
- z = x/R
- P = $(1 - z)\sigma_r$
- Q = $(1 - z)\sigma_t$
- u = ωx
- S = $\sigma_t + \sigma_r$
- D = $\sigma_t - \sigma_r$
- T = stress that would be produced in ring at radius R due to the centrifugal force of the ring.

MARTIN'S EXACT METHOD

Mr. Martin, in his paper, derived a differential equation covering the stress distribution in a rotating disk of conical profile. This derivation can be accomplished by applying the principle of equilibrium and the theory of elasticity to a particle in the disk. Applying the principle of equilibrium to a particle in a disk gives

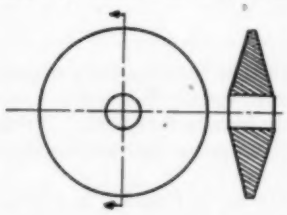


FIG. 1 DISK OF CONICAL PROFILE

$$\frac{d(xy\sigma_r)}{dx} - y\sigma_t + \mu\omega^2 x^2 y = 0. \quad [1]$$

Applying the theory of elasticity to a particle in a disk gives the following pair of relations:

$$\left.\begin{aligned} \sigma_r &= \frac{E}{1-\nu^2}\left(\frac{d\xi}{dx} + \nu\frac{\xi}{x}\right) \\ \sigma_t &= \frac{E}{1-\nu^2}\left(\nu\frac{d\xi}{dx} + \frac{\xi}{x}\right) \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots [2]$$

The following compatability equation can be obtained by eliminating ξ in Equation [2]:

$$\frac{d\sigma_t}{dx} - \nu\frac{d\sigma_r}{dx} = (1+\nu)\frac{\sigma_r - \sigma_t}{x} \quad \ldots\ldots\ldots\ldots [3]$$

By combining Equations [1] and [3], eliminating σ_t between them, and by making the following changes in coordinates, namely,

$$x = Rz$$
$$y = Y(1 - z)$$

$$P = \sigma_r(1-z), \text{ and}$$
$$Q = \sigma_t(1-z),$$

the following differential equation is obtained:[5]

$$z(1-z)\frac{d^2P}{dz^2} + (3-2z)\frac{dP}{dz} + (1-\nu)P + (3+\nu)\,Tz(1-z)^2 = 0 \ldots [4]$$

In deriving this differential equation, the following useful relations were found:

$$Q = P + z\frac{dP}{dz} + \mu\omega^2R^2(1-z)$$

$$T = 2 \times 10^{-6}D^2N^2$$

A complete solution of Equation [4] gives

$$P = P_1 + P_2 + P_3$$

where P_3 is the particular solution and

$$P_1 = A\left(1 - \frac{0.7}{3}z - \frac{0.7}{3}\frac{1.3}{8}z^2 - \frac{0.7}{3}\frac{1.3}{8}\frac{5.3}{15}z^3 - - - - - -\right)$$

$$P_2 = B(1-z)^2\left[1 + \frac{5.3}{3}(1-z) + \frac{5.3}{3}\frac{11.3}{8}(1-z)^2 + \frac{5.3}{3}\frac{11.3}{8}\frac{19.3}{15}(1-z)^3 + + + + + \right]$$

$$P_3 = T(0.13525 - 0.03156z - 0.34263z^2 + 0.23894z^3)$$

A and B being constants of integration.

Referring now to the change made in the coordinates, note that

$$\sigma_r = \frac{P_1 + P_2 + P_3}{(1-z)}$$

$$\sigma_t = \frac{Q_1 + Q_2 + Q_3}{(1-z)}$$

The use of this method is simplified by reducing the functions of z contained in P and Q to numerical form. This step is possible and practical since z varies only from zero to unity. Table 1 is the result of the introduction of numerical quantities in these functions.

Furthermore, since each of the terms in P and Q is divided by $(1-z)$, this division can be made in Table 1, and Table 2 is established.

In order to fully explain the use of this method, it is applied to a specific example (Fig. 2).

Calculating the radius at which the extended sides will meet gives

$$R = 32.5 \text{ in.}$$

$$z_{hub} = \frac{5}{32.5} = 0.1538$$

$$z_{rim} = \frac{21.5}{32.5} = 0.6616$$

By graphical interpolation the following values for Table 2 are obtained for the boundary values of z:

z	p_1	q_1	p_2	q_2	p_3	q_3
0.1538	0.174	0.1743	1.632	1.568	—27.150	35.000
0.6616	0.120	0.1525	3.490	2.608	— 0.747	3.407

[5] An error appeared in Mr. Martin's paper as published. In the last term of Equation [4], the factor $(3+\nu)$ was written and used as $(3-\nu)$.

TABLE 1[1]

z	$P_1 = Ax$	$Q_1 = Ax$	$P_2 = Bx$	$Q_2 = Bx$	$P_3 = Tx$	$Q_3 = Tx$
0.00	1.00000	1.00000	∞	∞	0.1655	0.1655
0.05	.....	.....	—259.74	274.18		
0.10	0.97627	0.95214	— 59.957	69.549	0.1577	0.1555
0.15	.....	.....	— 24.380	31.065		
0.20	0.95170	0.90163	— 12.431	17.525	0.1434	0.1410
0.25	.....	.....	— 7.1497	11.159		
0.30	0.92616	0.84800	— 4.4597	7.6257	0.1233	0.1239
0.35	.....	.....	— 2.8517	5.5453		
0.40	0.89953	0.79059	— 1.8949	4.1494	0.1017	0.1043
0.45	.....	.....	— 1.2804	3.1844		
0.50	0.87163	0.72853	— 0.87163	2.4718	0.07799	0.0838
0.60	0.84220	0.66047	— 0.39950	1.5264	0.05423	0.0631
0.70	0.81091	0.58429	— 0.17011	0.93046	0.03280	0.0433
0.80	0.77723	0.49589	— 0.05943	0.52287	0.01611	0.0257
0.90	0.74021	0.38711	— 0.01205	0.22632	0.00442	0.0110
1.00	0.69702	0.20911	— 0.00000	0.00000	0.00000	0.0000

[1] Values of P_1, Q_1, P_2, Q_2 from Martin's paper.

TABLE 2[1]

	Stresses due to centrifugal forces		Stresses due to a pull of 1 lb. per inch run applied to knife edge forming periphery of disk		Stress coefficients due to an infinite pressure applied to interior of infinitely small hole in the center	
z	Radial $p_1 = Tx$	Tangential $q_1 = Tx$	Radial $p_2 = Ax$	Tangential $q_2 = Ax$	Radial $p_3 = Bx$	Tangential $q_3 = Bx$
0.00	0.1655	0.1655	1.435	1.435	∞	∞
0.05	0.1709	0.1695	1.497	1.475	—273.400	288.600
0.10	0.1753	0.1725	1.559	1.518	— 66.620	77.280
0.15	0.1782	0.1749	1.627	1.565	— 28.680	36.550
0.20	0.1794	0.1763	1.707	1.617	— 15.540	21.910
0.25	0.1784	0.1773	1.796	1.674	— 9.553	14.880
0.30	0.1761	0.1767	1.898	1.738	— 6.371	10.890
0.35	0.1734	0.1757	2.015	1.809	— 4.387	8.531
0.40	0.1694	0.1739	2.151	1.890	— 3.158	6.915
0.45	0.1635	0.1712	2.311	1.983	— 2.328	5.788
0.50	0.1560	0.1675	2.501	2.090	— 1.743	4.944
0.55	0.1465	0.1633	2.733	2.217	— 1.309	4.301
0.60	0.1355	0.1579	3.021	2.369	— 0.9988	3.816
0.65	0.1229	0.1525	3.390	2.556	— 0.7523	3.419
0.70	0.1094	0.1445	3.860	2.794	— 0.5670	3.102
0.75	0.0956	0.1370	4.559	3.111	— 0.4161	2.835
0.80	0.0805	0.1286	5.563	3.557	— 0.2971	2.614
0.85	0.0634	0.1193	7.263	4.276	— 0.1995	2.421
0.90	0.0442	0.1100	10.620	5.554	— 0.1203	2.263
0.95	0.0231	0.0976	20.645	8.890	— 0.0555	2.140
1.00	0.0000	0.0840	∞	∞	— 0.0000	2.051

[1] Values of p_2, q_2, p_3, q_3 taken from Martin's paper.

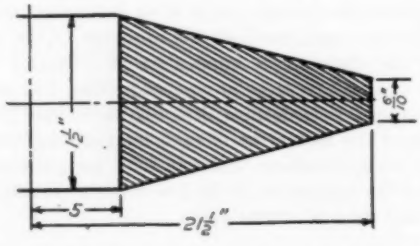


FIG. 2 EXAMPLE OF USE OF MARTIN'S METHOD
(Material of disk, steel; rim load, 8850 lb. per sq. in.; hub load, —710 lb. per sq. in.; speed, 3000 r.p.m.)

Substituting in the expression for T gives

$$T = 2 \times 10^{-6} \times 65^2 \times 3000^2 = 76{,}040$$

Since, according to the solution of the differential equation,

$$\sigma_r = Tp_1 + Ap_2 + Bp_3$$

and since the radial stresses on the rim and the hub are known, together with the above computed values of Table 2 for the rim and the hub,

$$8850 = 76{,}040 \times 0.12 + 3.490A + (-0.747)B$$
$$-710 = 76{,}040 \times 0.174 + 1.632A + (-27.15)B$$

Solving the two equations for A and B gives

$$A = 29.9 \qquad \text{and} \qquad B = 528$$

Having found A and B from known boundary conditions, the stresses can be found for any radius in the disk. Consider

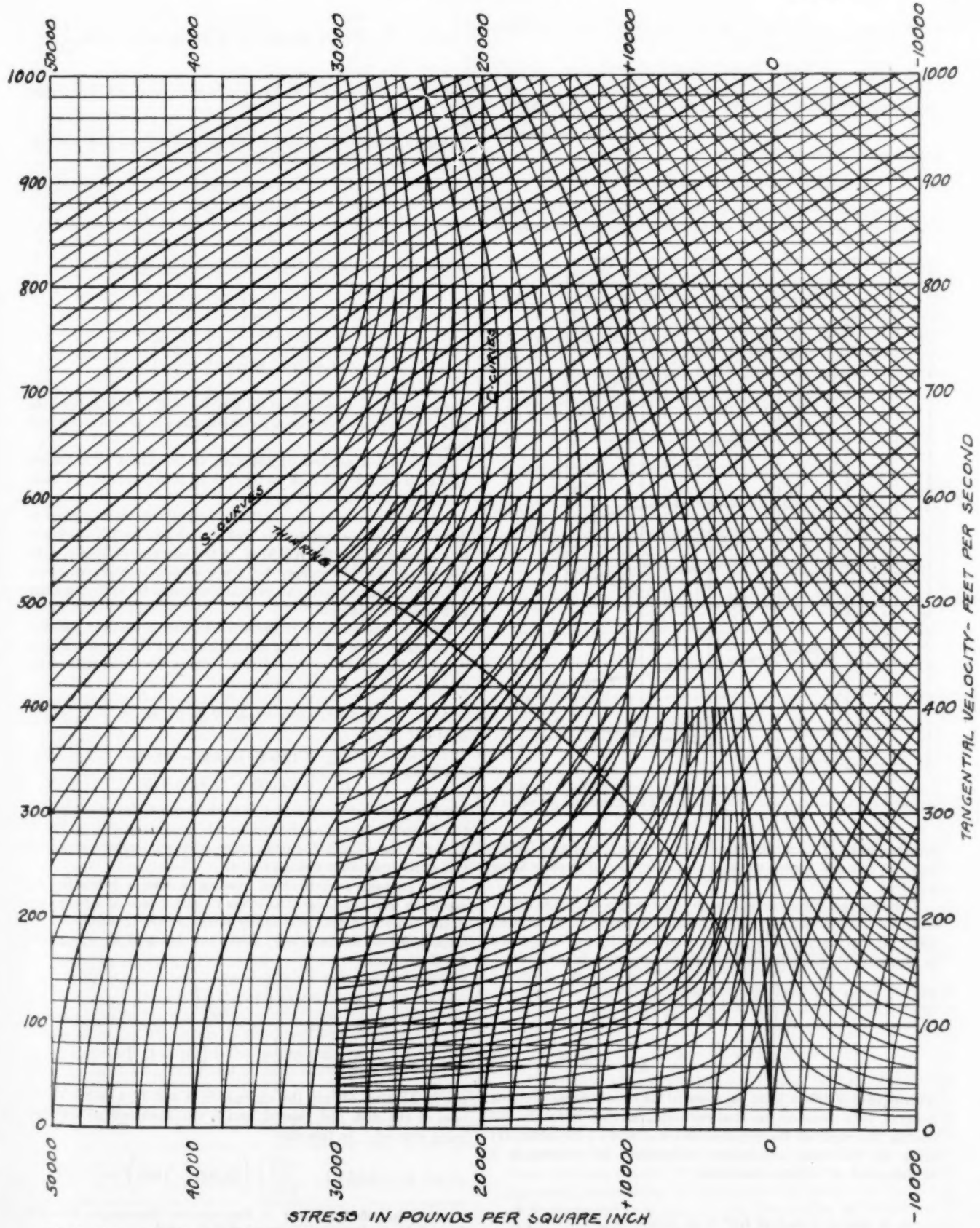


FIG. 3 DONATH'S "SUM AND DIFFERENCE" CURVES OF TANGENTIAL AND RADIAL STRESSES FOR PARALLEL-SIDED ELASTIC STEEL DISKS

a case when $z = 0.2$; then x the radius $= 32.5 \times 0.2 = 6^1/_2$ in., and at $6^1/_2$ in. the stresses are:

$$\sigma_r = 76{,}040 \times 0.1795 + 29.9 \times 1.707 - 528 \times 15.54 = 5{,}500$$

$$\sigma_t = 76{,}040 \times 0.1833 + 29.9 \times 1.617 + 528 \times 21.9 = 25{,}600$$

The complete stress-distribution curves found in this manner are the exact stress curves in Fig. 8.

Donath's "Sum and Difference Curves"

An approximate application of Donath's "Sum and Difference Curves" has been a popular method of determining stresses in

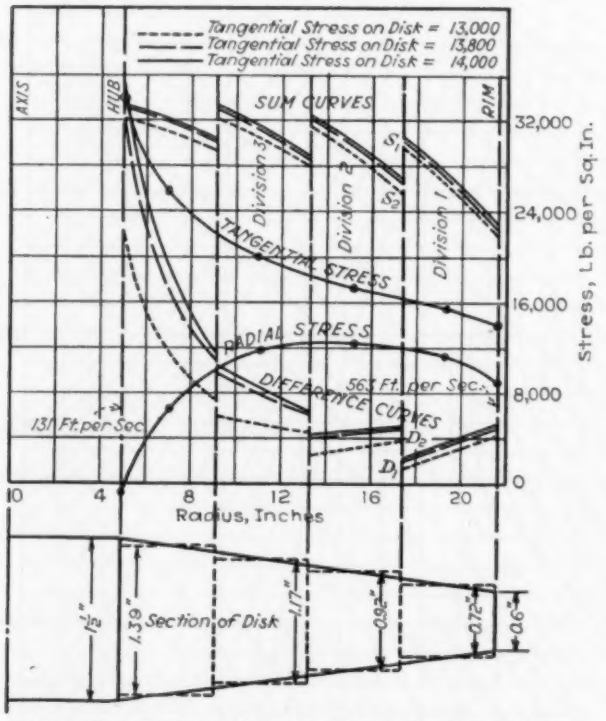


Fig. 4 Use of Donath Curves—"Equal-Division" Method
(Material of disk, steel; speed, 3000 r.p.m.; rim load, 8850 lb. per sq. in.; hub load, —710 lb. per sq. in.)

rotating disks of variable cross-section. The "Sum and Difference" in this case means the sum and difference of stresses along a radius.

The sum and difference curves in this case are of course linked with the differential equation covering stresses in a rotating disk. If the values of σ_t and σ_r, Equation [2], are introduced in Equation [1] the following differential equation is obtained:

$$\frac{d^2\xi}{dx^2} + \left[\frac{d(\log_e y)}{dx} + \frac{1}{x}\right]\frac{d\xi}{dx} + \left[\frac{\nu}{x}\frac{d(\log_e y)}{dx} - \frac{1}{x^2}\right]\xi + \frac{(1-\nu)}{E}\mu\omega^2x^2 = 0 \ldots . [5]$$

This equation differs from [4] only in its coordinates. Setting y equal to a constant in this equation, solving it for ξ, and placing the result in the expressions for σ_t and σ_r, Equation [2] gives the following well-known expressions for stresses in a rotating disk of uniform thickness:

$$\sigma_r = \frac{E}{1-\nu^2}\left[(3+\nu)\,Kx^2 + (1+\nu)b_1 - (1-\nu)b_2\frac{1}{x^2}\right]$$

$$\sigma_t = \frac{E}{1-\nu^2}\left[(1+3\nu)\,Kx^2 + (1+\nu)b_1 + (1-\nu)b_2\frac{1}{x^2}\right]$$

where $K = \dfrac{(1-\nu^2)\mu\omega^2}{8E}$, and b_1 and b_2 are constants of integration.

Donath discovered that by adding and subtracting the expressions for σ_t and σ_r and letting

$$S = \sigma_t + \sigma_r$$

$$D = \sigma_t - \sigma_r$$

$$u = \omega x$$

the following relatively simple functions were obtained:

$$S = (1+\nu)\frac{\mu}{2}\left[-u^2 + K_1\right]$$

$$D = (1-\nu)\frac{\mu}{4}\left[u^2 + \frac{K_2}{u^2}\right]$$

These functions were plotted for a series of K's, Fig. 3, covering a field of stresses and speeds in which disks in practical use will fall.

The common method of applying these curves to a conical disk is to assume the disk to be divided into a number of disks of uniform thicknesses (Fig. 4).

A simplification of the method of applying these curves to disks of variable cross-section was introduced by Mr. Driessen in his paper before the A.S.M.E. in the summer of 1928.[6] The use of the Donath curves by the common method with the Driessen simplifications can best be explained by applying them to a practical example.

Refer to the example represented in Fig. 4, assume the disk to be made up of four disks of uniform axial thicknesses and of equal radial thicknesses as shown. Since the tangential stress on the rim is not known, assume

$$\sigma_t \text{ for the rim} = 13{,}000 \text{ lb. per sq. in.}$$

then

$$S_{\text{rim}} = 13{,}000 + 8850 = 21{,}850$$

$$D_{\text{rim}} = 13{,}000 - 8850 = 4150$$

Having these values of S and D at the known rim speed locates the S- and D-curves applicable to this imaginary disk in Fig. 3. Transfer the S- and D-curves between the boundary speeds of this outer imaginary disk to Fig. 4.

Mr. Driessen's method of passing across a boundary by assuming that the tangential stress on either side of a boundary is the same and that the radial stress varies inversely as the thicknesses of the imaginary disks can be used at this point. He showed that

$$\Delta S = -0.65\left(1 - \frac{t_1}{t_2}\right)\left(S_1 - D_1\right)$$

$$\Delta D = 0.35\left(1 - \frac{t_1}{t_2}\right)\left(S_1 - D_1\right)$$

where ΔS and ΔD are the changes in S and D, t_1 is the thickness of the disk just passed, and t_2 the thickness of the disk being entered. In this case

$$\Delta S = -0.65\left(1 - \frac{0.72}{0.92}\right)\left(29{,}500 - 1500\right) = -3970$$

[6] "A Simplified Method of Determining Stresses in Rotating Disks," M. G. Driessen, Trans. A.S.M.E., 1928.

$$\Delta D = 0.35\left(1 - \frac{0.72}{0.92}\right)\left(29{,}500 - 1500\right) = 2140$$

Adding these increments to S_1 and D_1 gives S_2 and D_2, which are known values of S and D on the outer boundary of Division 2.

Following this procedure in crossing each of the imaginary disks, the S- and D-curves can be established across the entire radius of the disk. It happens in this case that the radial stress indicated at the hub by these curves is 5100, while the actual radial stress is known to be —710; this difference is an indication that the wrong tangential stress was assumed for the rim.

Assume another tangential stress of 13,800 for the rim, and establish the second set of S- and D-curves as are indicated in

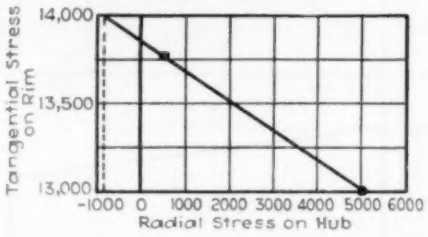


FIG. 5 RELATION BETWEEN RADIAL AND TANGENTIAL STRESSES ON HUB

Fig. 4. At the hub, this set of curves indicates a radial hub stress of 500, which is also incorrect.

At this point in the process, another one of Mr. Driessen's simplifications can be used. He showed that there was a direct relation between the radial stress on one boundary and the tangential stress on the opposite boundary. In this case the two assumed tangential rim stresses with their consequent radial hub stresses can be plotted as in Fig. 5. On this graph the tangential stress on the rim which will give a radial stress of —710 on the hub is indicated to be 14,000.

With this new value of tangential stress on the rim, the third set of S- and D-curves in Fig. 4 can be established.

Since the midpoint between the boundaries of each of the divisions is of the same thickness as the actual disk at that point, the values of S and D indicated at such points are assumed to be most accurate. Through these points the stress-distribution curves, Fig. 4, are established.

The stress-distribution curves obtained in this manner, as well as others obtained by using two and eight divisions of the disk, have been transferred to Fig. 8 where their relative accuracies can be obtained by comparing them to the exact stress-distribution curves.

ONE-EQUIVALENT-DISK METHOD

Some experimentation with the application of Donath's curves has shown that there are other satisfactory methods of applying them. One of the methods involves the assuming of one equivalent disk of uniform thickness equal to the hub thickness of the conical disk for the conical disk. Also, it is assumed that the radial rim stress on this imaginary disk is equal to the radial rim stress of the conical disk decreased in a ratio of rim thickness to hub thickness of the conical disk. To this new disk apply Donath's curves just as they would be applied to a disk of uniform thickness. The stresses indicated by these curves at the hub and at the midpoint between the rim and the hub and the known stresses on the rim are used to construct the stress-distribution curves.

For a better explanation of this equivalent-disk method, consider an example (Fig. 6). Assume

σ_t for the rim = 13,000

then

$$S_{\text{rim}} = \sigma_t + \sigma_r = 13{,}000 + 8850 = 21{,}850$$

$$D_{\text{rim}} = \sigma_t - \sigma_r = 13{,}000 - 8850 = 4150$$

Reducing the rim radial stress in a ratio of rim thickness to hub thickness gives the change in S and D by Mr. Driessen's method as

$$\Delta S = -0.65\left(21{,}850 - 4150\right)\left(1 - \frac{0.6}{1.5}\right) = -6900$$

$$\Delta D = 0.35\left(21{,}850 - 4150\right)\left(1 - \frac{0.6}{1.5}\right) = 3700$$

so $S_1 = 14{,}950$ and $D_1 = 7850$.

Locating these values of S and D on the Donath curve sheet at the rim speed establishes two curves which, when followed to the hub speed and transferred to Fig. 6, prove to be the correct ones since the indicated radial hub stress is the same as the actual hub stress. (σ_t for the rim was chosen correctly just by chance, otherwise other assumptions would have had

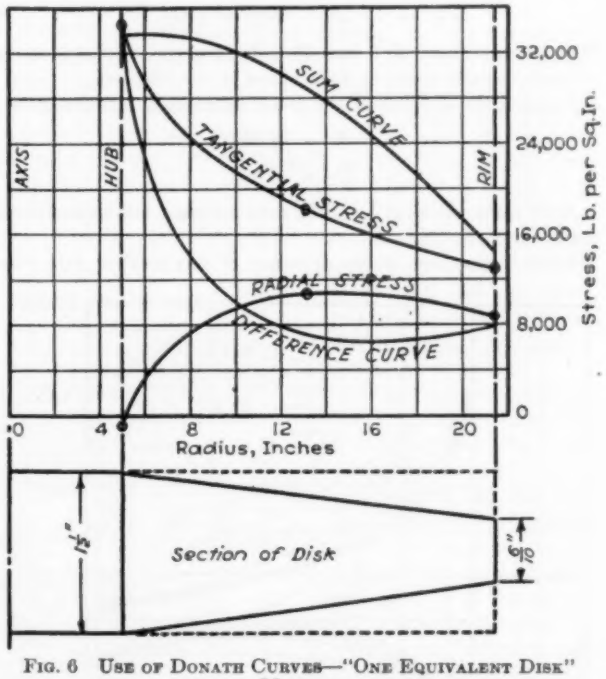


FIG. 6 USE OF DONATH CURVES—"ONE EQUIVALENT DISK" METHOD

(Material of disk, steel; speed, 3000 r.p.m.; rim load, 8850 lb. per sq. in.; hub load, —710 lb. per sq. in.)

to be made and Mr. Driessen's simplification would have been used if the second assumption had been incorrect.)

Refer to Fig. 8 for a comparison of the accuracy of this method with the exact stress curves and with the other approximate curves.

TWO-DIVISION METHOD

Experiment also shows that, by dividing the disk in the neighborhood of three-quarters of the distance from the hub to the rim, very satisfactory results can be obtained. With this method the S- and D-curves are established across the outer imaginary disk just as if the outer imaginary disk were of uniform thickness equal to the thickness of the rim. The boundary is crossed by assuming the indicated radial stress at the boundary to be decreased in a ratio of rim thickness to hub thickness of the conical disk. With the new values of S and D thus ob-

tained the curves are established across the inner imaginary disk. The values of stress indicated at the rim and hub and at midpoint between the division and hub are taken as points through which to construct the stress-distribution curves.

For a better explanation of the use of this method, apply it to an example (Fig. 7).

The radius of the division which has been found to give the best results can be calculated from the following formula:

$$X = \sqrt{\frac{\frac{b^4}{4} + \frac{a^4}{12} - \frac{ab^3}{3}}{\frac{b^2}{2} + \frac{a^2}{2} - ab}}$$

where b = rim radius, and
a = hub radius

Assume

$$\sigma_t \text{ for the rim} = 13{,}250$$

then

$$S_{\text{rim}} = 13{,}250 + 8850 = 22{,}100$$

$$D_{\text{rim}} = 13{,}250 - 8850 = 4400$$

With these values of S and D, establish the S- and D-curves between the rim speed and the speed at the division.

Crossing the division by the use of Driessen's formulas gives

$$S_2 = 17{,}500$$

$$D_2 = 6{,}000$$

With these values of S and D, the curves are established across the inner part of the disk.

For a comparison of the accuracy of this method with other methods, refer to Fig. 8. Note that the three points obtained for the tangential-stress curve fall upon the exact tangential-stress-distribution curve.

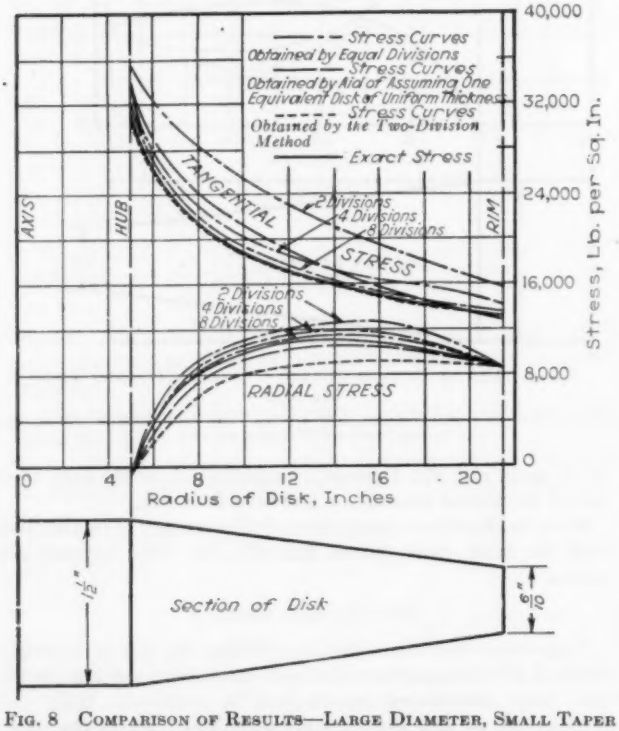


FIG. 8 COMPARISON OF RESULTS—LARGE DIAMETER, SMALL TAPER
(Material of disk, steel; speed, 3000 r.p.m.; rim load, 8850 lb. per sq. in.; hub load, —710 lb. per sq. in.)

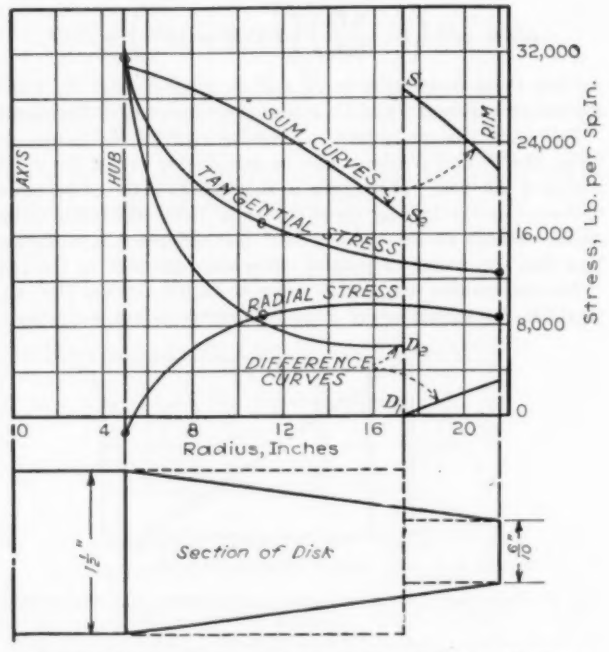


FIG. 7 USE OF DONATH CURVES—TWO-DIVISION METHOD
(Material of disk, steel; speed, 3000 r.p.m.; rim load, 8850 lb. per sq. in.; hub load, —710 lb. per sq. in.)

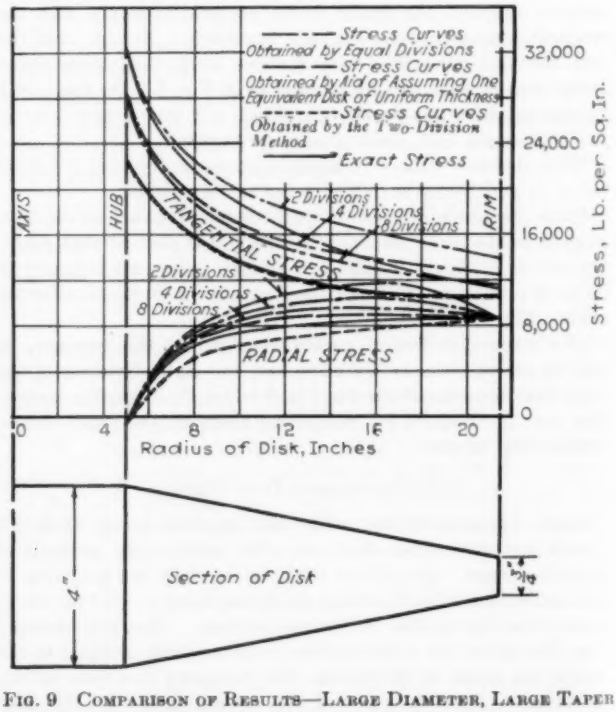


FIG. 9 COMPARISON OF RESULTS—LARGE DIAMETER, LARGE TAPER
(Material of disk, steel; speed, 3000 r.p.m.; rim load, 8850 lb. per sq. in.; hub load, —710 lb. per sq. in.)

APPLICATION TO A VARIETY OF DISKS

Figs. 8, 9, 10, 11, and 12 show the comparisons of the results obtained by the various methods on a variety of disks. The variations in the examples included are the diameters and tapers. From study of these examples, the most accurate approximate method can be selected for general use.

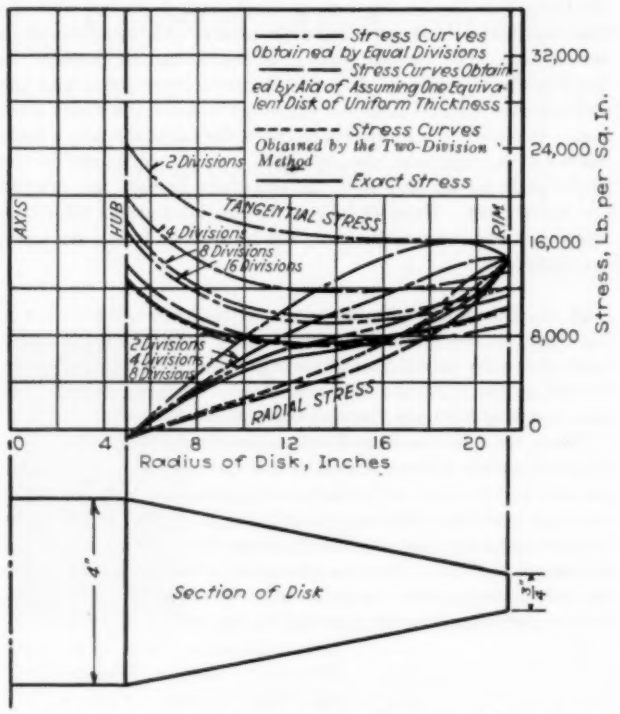


FIG. 10 COMPARISON OF RESULTS—LARGE DIAMETER, LARGE TAPER
(Material of disk, steel; speed, 1500 r.p.m.; rim load, 15,000 lb. per sq. in.; hub load, —500 lb. per sq. in.)

RESULTS

One feature about the stress curves obtained by the equal-division method is that they are in all cases higher than the exact stress curves. The reason for this higher stress indication is apparent since, in any one of the imaginary disks used, Fig. 4, there is assumed to be some material added and some material dropped along the radius, the added material being at a larger radius than the corresponding dropped piece of material. Since the effect of material varies as the square of its radius, this added material more than compensates for the dropped material.

In the comparisons of results, the relative accuracies obtained by two, four, and eight divisions of the disk can be seen. One important feature of the stress distributions obtained in this manner is that the accuracy increases at a rapidly decreasing rate as the number of equal divisions increases. For this reason, four equal divisions give as good results as eight. In one case where there was a high rim load and large taper, eight equal divisions gave an appreciable increase in accuracy over four divisions; however, sixteen divisions were very little better than eight in that case.

The stress-distribution curves obtained by the equivalent-disk method are satisfactory except in a case where the disk has a large taper and is running at a high speed. This method, Fig. 6, assumes a disk with more material at the rim, but the midpoint between the two boundaries is the point at which the additional material will compensate for the lower radial stress at the rim. As can be seen in the comparison of results, this method gives good results in most cases; and the error is fortunately on the safe side. The reason it does not give satisfactory results for a disk of large taper and high speed is that the added material becomes more effective as the speed increases.

The "two-division" method is a successful method of assuming a decreased radial load at some point in the disk and adding material to compensate for it at some certain point. Having this method work satisfactorily on the variety of examples included in this paper is an indication that it will give good results in all cases.

The practical value of Mr. Driessen's "Simplified Method of Determining Stresses in Rotating Disks" has been determined in solving the examples included in this paper. It would seem on first sight that his method of determining a correct value of tangential stress to assume on the rim, after having assumed two values unsuccessfully, would be of great value in the use of Donath's curves. However, after having used these curves

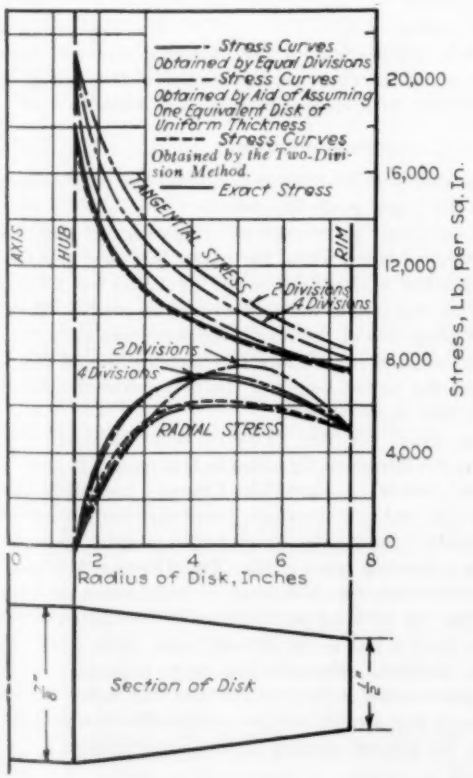


FIG. 11 COMPARISON OF RESULTS—SMALL DIAMETER, SMALL TAPER
(Material of disk, steel; speed, 5000 r.p.m.; rim load, 5000 lb. per sq. in.; hub load, —500 lb. per sq. in.)

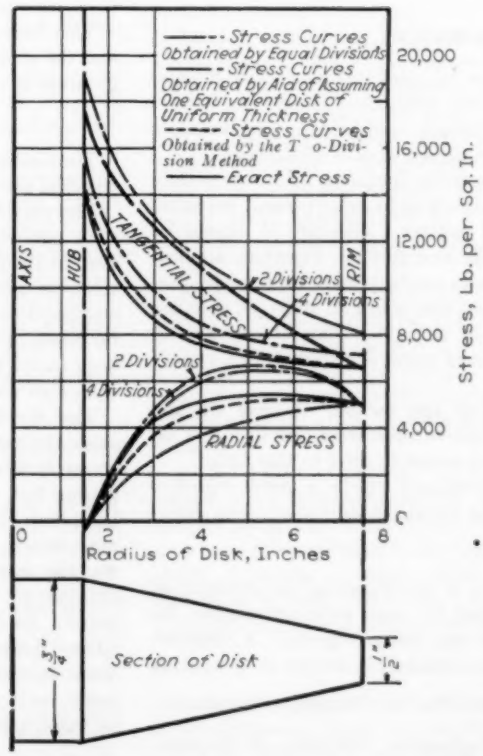


FIG. 12 COMPARISON OF RESULTS—SMALL DIAMETER, LARGE TAPER
(Material of disk, steel; speed, 6000 r.p.m.; rim load, 5000 lb. per sq. in.; hub load, —500 lb. per sq. in.)

for a number of disks, the author has found that a person with some experience in their use can determine mentally the correct stress to assume. Due to the limited field which the published S- and D-curves cover, a person will usually have to assume several stresses before he can establish curves which will not run off the curve sheet before they reach the hub speed; and by the time he has found a tangential rim stress which will not cause the D-curve to run off the sheet before reaching the hub speed, he is able to determine mentally the correct stress value to assume.

This condition is caused by the nature of the difference curves. The value of D on the hub is affected greatly by a small change in the tangential stress on the rim, while the value of S on the hub under similar conditions is affected but a comparatively small amount. The value of Mr. Driessen's method would be increased by an increase in the field which the published S- and D-curves cover.

Mr. Driessen's formulas for use in crossing a boundary between two imaginary disks of uniform thicknesses are of great value, as these simple expressions eliminate the necessity of computing an extensive table for that purpose.

Conclusions and Recommendations

1 In applying the Donath curves by equal divisions along the radius of a disk, four equal divisions of the disk will be found to give satisfactory results.

2 By the method of assuming one equivalent disk of uniform thickness for a conical disk, results of satisfactory accuracy can be obtained for all disks except in cases where the speed is high and the taper large.

3 By the two-division method proposed in this paper, accurate results can always be obtained.

Discussion

A. L. Kimball.[7] The writer has read this paper with interest as one with some familiarity with methods of computing stresses in revolving turbine disks, though not having that intimate knowledge that comes from continued work in disk design.

The value of this paper consists primarily in an up-to-date review of the subject such as cannot be found in present literature without dipping into several sources. Although it cannot be said to contain a fundamentally new method, ingenious application of the Donath and Driessen methods have been made, particularly in separating a conical disk into two steps, as specified by the formula at the top of page 96. This should greatly shorten the labor of calculation of stress distribution in a variety of shapes of conical disks.

Apparently the equations at the bottom of page 94 and the top of page 95 refer to a different disk from that of Fig. 4, although the discussion following seems to refer to this figure.

The paper is clearly presented, and will be a useful reference to have at hand when brushing up on the subject of stress determination in revolving disks.

S. Timoshenko.[8] The idea of investigating the accuracy of the well-known Donath method by using an exact solution for disks of conical profile seems a very interesting one. A considerable amount of work has been done by the author in comparing the Donath method, using various numbers of divisions, with the exact method. It is very desirable to give this comparison in percentages. For certain cases it appears that an increase in the number of divisions does not improve the results, and the approximate method does not converge toward the exact solution. It would be a good idea to check the calculations for these cases. It is important, also, to check over the equations on the fourth page for ΔS and ΔD; the equations as they are printed are not correct. Possibly in some of the calculations the incorrect equations were used, so it will be well to look over these calculations.

M. G. Driessen.[9] So far as the writer knows, the author is the first to compute the widely used approximate Donath method for the calculation of these stresses in rotating disks of conical profile. His studies show clearly the magnitude of the error made in applying the sum and difference curves.

The author also introduced a simplification by which the calculation of conical disks can be shortened considerably. He divides the disk in two parts as the radius of gyration, so that the upper part is a cylindrical disk with a width equal to the width at the rim and the lower part also a cylindrical disk with a width equal to that of the hub. He also showed that this latter method is the best approximation for different kinds of conical disks. The author uses the following formula for the radius of gyration X:

$$X = \sqrt{\frac{\frac{b^4}{4} + \frac{a^4}{12} - \frac{ab^2}{3}}{\frac{b^2}{2} + \frac{a^2}{2} - ab}}$$

where b = rim radius

a = hub radius.

This formula is independent of the widths at rim and hub, and the writer would therefore ask the author if the radius of gyration is the same for a large and for a small taper.

Author's Closure

The author wishes to express his gratitude for the constructive criticism given this paper in the discussion.

The general equations to be used in evaluating ΔS and ΔD were not written correctly when the paper was first printed; also, in the numerical example where the equations for ΔS and ΔD were used, the expressions were not written correctly. These changes do not affect any of the results obtained because a careful check of the original calculations has been made and it has been found that the correct forms of the two above-mentioned equations were used at all times.

The dividing point for the "Two-Division Method" was originally called the Radius of Gyration in this paper; it was incorrectly named, however. Since Mr. Driessen has called this glaring fact to the author's attention, stress-distribution curves have been obtained by using the actual radius of gyration of the cross-section as a dividing point in the "Two-Division Method." But the results obtained were not nearly as satisfactory as those obtained by using the dividing point originally introduced in the paper. As the facts stand at the present time, when using the "Two-Division Method," the dividing point originally introduced in the paper gives better results than any other known point; so it would best remain as it is until another can possibly be found when the subject receives further investigation.

[7] Research Engineer, General Electric Company, Schenectady, N. Y. Assoc. A.S.M.E.

[8] Professor of Mechanical Engineering, University of Michigan, Ann Arbor, Mich. Mem. A.S.M.E.

[9] Reymersbeek, Nuth (L).

APM-53-8

Elastic and Inelastic Behavior in Spring Materials[1]

Progress Report No. 7 of the A.S.M.E. Special Research Committee on Mechanical Springs (Technological)

By M. F. SAYRE,[2] SCHENECTADY, N. Y.



AS SUGGESTED at meetings of the committee held a year ago, work during the past year under the auspices of the committee at Union College has been divided between a continuation of the fundamental studies previously started and some items of work of a more immediately practical nature. Progress Report No. 6[3] belongs in the second class. The present report describes the results obtained to date in connection with the more fundamental studies of the elastic behavior of spring materials.

These results may be summarized as follows:

1 A new method of work has been adopted, by which a greatly increased order of accuracy has been obtained in studies of behavior of spring materials in tension. Using a 52-ft. gage length, with dead-weight loading, under carefully controlled temperature conditions, readings of elongation can be taken to one part in fifteen million on the total length of the wire sample, with a probable mean accuracy under favorable conditions of one part in one million.

In most cases the controlling factor in accuracy tends to be the error resulting from the irregular plastic permanent yield, and from the gradual elastic or pseudo-elastic creep and recovery in the wire itself when loading and unloading.

2 Measurements obtained by this method have confirmed in general the tentative conclusions suggested in Progress Report No. 5. In particular it is now possible to give numerical values for the rate of change of Young's modulus of elasticity with stress for several materials. There is reasonably strong evidence that these same equations hold both in tension and in compression, although it has not yet been possible to obtain direct verification in compression.

3 The practical effect of this change of modulus as applied to spring design and manufacture is as follows:

a True stresses in compression are somewhat higher, and in tension somewhat lower, than the values given by the usual formulas. A sufficiently close approximation to numerical results may be obtained by first computing the maximum stress by the usual formula, and determining the per cent difference between the corresponding modulus of elasticity and the modulus as of zero stress. The per cent error in stress will be 70 to 75 per cent as great as the per cent difference in modulus

b The increasing stiffness in compression with increasing stress more or less completely counterbalances the decreasing stiffness in tension, so that apparent stress-deflection curves in bending or torsion will be almost unaffected by the variation in modulus

c In high-strength, highly elastic materials, such as are used in springs, it may be desirable, when selecting values of elastic limit from tensile stress-strain results, to make allowances for the curvature.

4 Tests have confirmed the general principle of a decrease in modulus of elasticity in tension as a result of previous overstrain in tension. This is an important reason for more or less erratic variations in stiffness of springs resulting from overstrain, as in surging. It also appears to be an important reason for a hidden safety factor in design, in that it causes the true stresses in highly overstressed areas to be appreciably smaller than the computed values. The amount of this decrease in modulus with overstressing is evidently different for different materials, and naturally depends upon the previous history of the material. It is therefore difficult to give numerical values, except in general terms.

An article in Japanese published last year by Mr. Tadashi Kawai (*Kinzoku no Kenkyu*, 1930, no. 7, pp. 177–201; abstract in the *Journal of the Institute of Metals*, vol. 43, p. 430) describes an extended study of this effect.

5 Work has been started on studies of the elastic after-effect. This has not yet gone far enough to justify drawing any conclusions.

6 Mechanical hysteresis values for unidirectional stress in tension have been obtained for several additional samples of various materials. Enough data have been obtained to serve as a basis of comparison in further studies of the relative suitability of various materials and of the effect of various types of cold work or heat treatments. A possible correlation between hysteresis values and fatigue strength deserves investigation.

Details of Work

In Progress Report No. 5, the results were given of a series of tests in tension made using an Amsler universal testing machine and a Riehlé-Sayre mirror extensometer. This method gave a relatively high order of precision, but it was evident that still higher accuracy was needed to obtain the best results. It was also evident that, in obtaining this higher accuracy, it was de-

---

[1] Continuation of the fundamental studies begun in Progress Report No. 5. Trans. A.S.M.E., vol. 52, no. 15, May-August, 1930, paper APM-52-9.

[2] Supervisory Member in Charge of Research at Union College, and Associate Professor of Applied Mechanics, Union College. Mem. A.S.M.E. Professor Sayre was graduated from Columbia University with the degrees of E.M. and A.M. Seven years of engineering experience were followed by fifteen years at Union College, the latter portion diversified by research and consulting work carried on for the General Electric Co. and for other concerns.

[3] "Stresses and Deflections in Helical and Spiral Torsional Springs as Affected by Curvature." Presented at the Spring Meeting of the A.S.M.E., Detroit, Mich., June 11, 1930.

Contributed by the Special Research Committee on Mechanical Springs and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

sirable to use a new method sufficiently different in character from the old method to aid in detecting any systematic errors which may have been present. This suggested immediately the use of dead-weight loading rather than a testing machine, and the use of long gage lengths—as long as possible. It also suggested the need for accurate control of temperature.

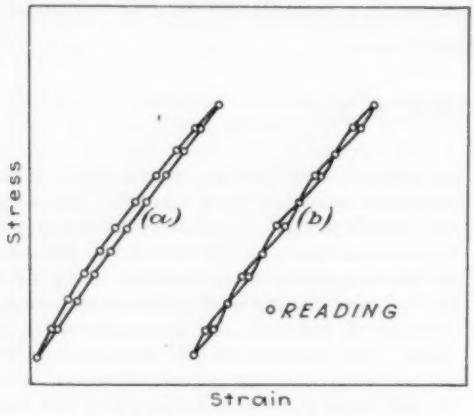


FIG. 1 ROUTINE OF TESTING
(a) Hysteresis measurements.
(b) Modulus-of-elasticity measurements.

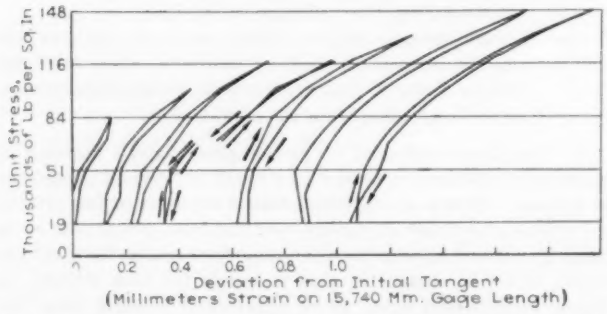


FIG. 2 TYPICAL HYSTERESIS LOOPS—0.67 PER CENT CARBON HEAT-TREATED STEEL WIRE

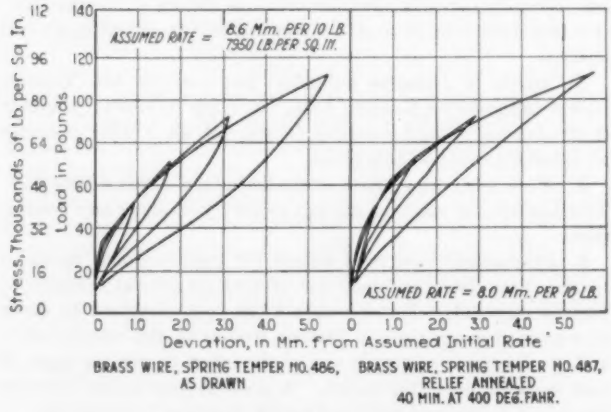


FIG. 3 TYPICAL HYSTERESIS LOOPS

There was available in the college laboratory a standpipe, five feet in diameter by somewhat over fifty feet high, with conical bottom and open top, which was used occasionally for hydraulic experiments. A hole was cut in the conical bottom, 4 in. from the circumference, and a 2-in. vertical steel pipe was placed in the standpipe above this hole, extending to slightly above the top of the tank. In this pipe were suspended two wires of the same material, a test wire and a comparison wire. At the upper end both wires could be sighted by a micrometer-microscope through a hole in the 2-in. pipe, and any vertical movement of either wire could be measured. At the lower end the comparison wire was connected to a lever, to which a small constant load was attached, and the test wire was attached to a basket, which could be loaded with standard weights. Movement of a gage point on the comparison wire, located just below the point where it emerged from the standpipe, was measured by a micrometer-microscope, and movement of the gage point on the test wire was measured by a Gaertner cathetometer, with 100-mm. lead screw. The overall gage length was approximately 15,740 mm., and each scale division on the cathetometer drum 0.005 mm.

The standpipe was filled with water to within a few inches of the top, so holding the temperature of the 2-in. pipe essentially constant during runs. Drafts of air up the pipe were prevented by lightly tucking a little cotton wool around the wires at the lower end of the pipe. The true elongation of the test wire under a given load was taken as equal to the change in the cathetometer reading, less the movement of the test wire at the top as given by the micrometer-microscope, corrected for any change in readings on the comparison wire which simultaneously occurred.

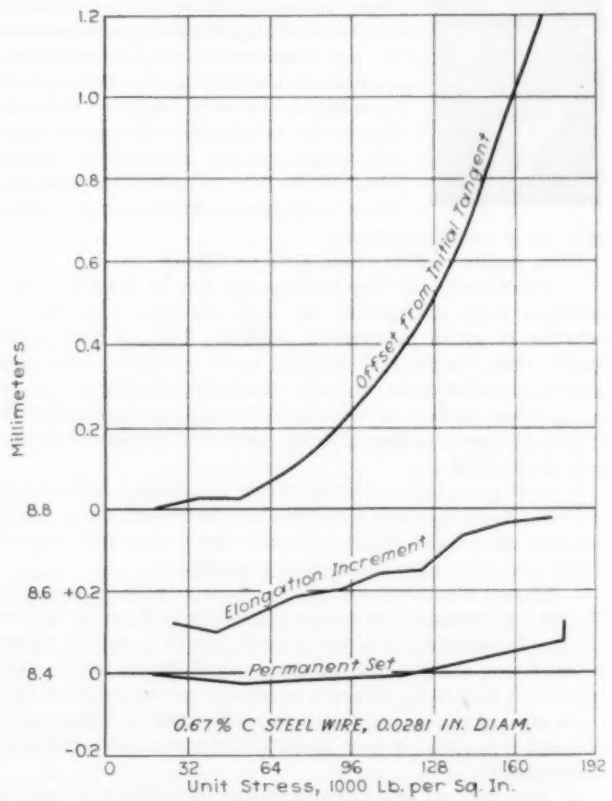


FIG. 4 ELONGATION RESULTING FROM STRESS INCREASE OF 16,000 LB. PER SQ. IN.

After a certain amount of preliminary experience, the following specifications were made up to cover the type of sample desired:

For convenience in loading, the diameter of the wire should be such that its breaking point will run between, say, 150 and 400 lb. Otherwise, the exact diameter does not matter. The method of preparation and heat treatment will, of course, depend upon the particular material, the general principle being to so treat as to develop its best properties for use as a spring.

Of each material there will be needed half a dozen lengths, each 55 ft. long, or, say, 400 ft. length in a single coil, free from sharp kinks or bends, and coiled on a sufficiently large-diameter reel so that it will straighten out under an initial load of 10 to 20 lb. when hung vertically. At first it was thought that the wire would have to receive its last reduction on some form of draw bench, so that it could be pulled out perfectly straight, but recent experience is that much more trouble is caused by small, sharp bends due to careless handling or by attempts to straighten the wire than by gradual curvature due to coiling on a drum. These sharp bends gradually straighten out under loading and ruin the numerical results. However, the larger this drum radius can be made, the better.

Please send with the material information as to the analysis, method of heat treatment, and final steps in the drawing, sufficiently detailed to thoroughly identify the material to others, and to make possible later comparison with other methods of handling.

Samples of 0.67 per cent carbon heat-treated steel, 17 SRT aluminum alloy, phosphor-bronze, and 70-30 brass in a drawn condition and after relief annealing have so far been tested. These samples are later described in detail.

It was of course necessary to overcome various practical difficulties. It was originally planned to use dead-weight loading for the comparison wire and a lever with a three-to-one ratio for the test wire, the lever having curved surfaces against which both the test wire and the load-basket support were carried so that the lever arms did not change as the wire elongated and

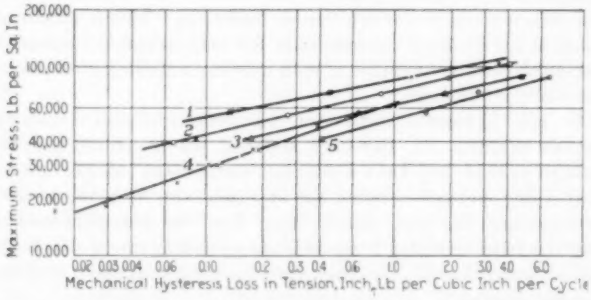


FIG. 5 VARIATION OF MECHANICAL HYSTERESIS LOSS IN TENSION WITH MAXIMUM STRESS

(Wire samples 52 ft. gage length.)

1—Phosphor bronze.
2—Phosphor bronze after permanent elongation of 0.12 per cent.
3—70-30 brass, relief annealed 40 min. at 400 deg. fahr.
4—17 SRT aluminum alloy.
5—70-30 brass, as drawn.
Minimum stress in cycle Nos. 1, 3, and 5: 9540 lb. per sq. in.
No. 2: 7950 lb. per sq. in.
No. 4: 6040 lb. per sq. in.

the lever tilted. Erratic readings were obtained, which were eventually traced to minute frictional resistance in the lever system. To avoid this, the lever was transferred to the comparison wire, and dead-weight loading was used for the test wire. This in turn left the test wire entirely free to swing, so that very careful technique was necessary in applying and removing loads, in order to avoid putting it in vibration. Another source of difficulty was combined backlash and friction in the micrometer used. By adjustments in the micrometer and by following a careful routine in the work, this has been reduced, but it is still a source of trouble.

Certain other difficulties also inhere in the nature of the materials themselves. For the purposes of this work it was desirable to distinguish as far as possible between the elastic elongation, which occurs instantaneously or nearly so; the plastic elongation or "permanent set," which is a function of time, but apparently an irregular function, sometimes stopping then starting again with but little evident cause; and the elastic or pseudo-elastic after-effect, a gradual elongation on loading, followed by as gradual a recovery to original length on unloading.[4]

[4] An investigation of the characteristics of this elastic after-effect has been started and the results will presently be reported.

It is accompanied by mechanical hysteresis or "back error," but the two seem to be more or less independent in character. At the sensitivity present in this work all these factors are large

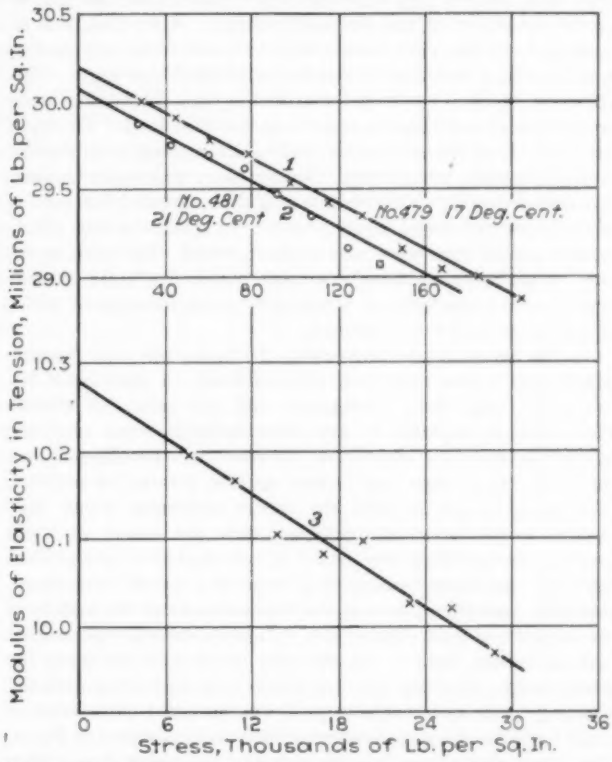


FIG. 6 VARIATION OF MODULUS OF ELASTICITY WITH TENSILE STRESS

Nos. 1 and 2: 0.67 per cent carbon heat-treated steel wire.
No. 3: 17 SRT aluminum-alloy wire.

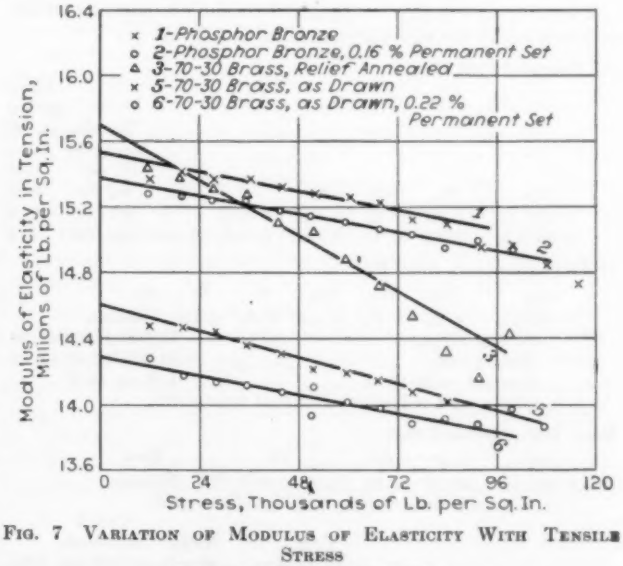


FIG. 7 VARIATION OF MODULUS OF ELASTICITY WITH TENSILE STRESS

(Wire samples 52 ft. gage length.)

enough to be important, and careful planning of work and adherence to a time schedule of observation are needed to avoid widely erratic readings.

In studying hysteresis, the specimen was first loaded to the desired maximum load, and the load left on until plastic yield

had ceased and the length of the specimen had become substantially constant. The specimen was then put through a series of cycles between a nominal initial load and the desired maximum load until it had approached a cyclical state, as judged by the constancy of the zero-load reading. After this, sets of readings were taken for three complete cycles from zero load to maximum load, and back to zero again by small load steps. The difference between readings for ascending and descending loads gave the width of the hysteresis loop for that load. To apply the load, adjust the micrometer, and take a reading, took slightly over one minute per reading. Every effort was made to keep this time constant. This routine was then repeated for successively larger and larger stress ranges. To study the time effect, certain special runs were also made in which after each increment of load a first reading was taken within five to six seconds after the load was applied, followed by other readings at sixty-second intervals for five minutes.

As the stress ranges increased, the hysteresis loop became wider, and it was more and more difficult to distinguish between the true elastic elongation and the collateral effects. This made it desirable to use other methods when studying modulus-of-elasticity variations. In runs made for this purpose, the total stress range was broken up into a series of sections, each short enough so that the elastic hysteresis within that portion would be small. Starting with the lowest of these sections, the specimen was placed in a cyclical or nearly cyclical state by successive loading and unloading within this range. Readings were then taken at the two ends and at the middle of the range for at least three cycles. This routine was repeated for each successive section. In this way, instead of obtaining the stress-strain curve for the specimen with ascending and descending loads as a single loop, it was obtained as a series of small loops touching each other at their tips, as shown in Fig. 1. The slope of a center line through the successive loops, after correcting for plastic yield between the loops, was taken as the true elastic line.

Materials Used

The analyses and physical properties of the wire samples tested, as furnished by the manufacturers, are as follows:

Oil-Tempered Steel Wire

Size	C	Mn	S	P	Si	Tensile strength, lb. per sq. in.	Elongation in 10 in., per cent
0.028 in.	0.67	0.35	0.047	0.054	...	277,000	3.8
0.0465 in.	0.74	0.41	0.045	0.036	0.160	292,000	4.4

17 SRT Aluminum-Alloy Wire

Size	Cu	Mn	Mg	Si	Fe	Al
0.064 in.	4.25	0.62	0.49	0.34	0.63	Balance

Tensile strength ..... 71,500 lb. per sq. in.
Yield point ..... 61,000 lb. per sq. in.
Elongation in 10 in. ..... 1.0 per cent
30 per cent reduction after heat treatment

Brass Wire, Spring Temper

Size	Cu	Fe	Pb	Zinc
0.040 in.	70.45	0.02	Less than 0.075	Balance

Wire was drawn from 0.100 in. diam. cold.

	Coil 1 as drawn	Coil 2 Relief annealed, 40 min. at 400 deg. fahr.
Tensile strength, lb. per sq. in.	119,500	118,500

Phosphor-Bronze Wire

Size	Cu	Sn	P	Pb	Fe
0.040 in.	95.12	4.65	0.19	0.07	0.02

Finished with a reduction of 8 B and S Nos. (84.5 per cent reduction of area)

Summary of Results

Typical sets of stress-strain readings for complete closed loops in approximately cyclical condition are given in Tables 6A to 6H.[5] The correaponding stress-strain curves are shown in Figs. 2 and 3. In these figures, stress is used as an ordinate. As an abscissa, is used not the true strain but rather the deviation from an assumed initial rate of strain, the difference between the actual strain and that which would have occurred had the assumed initial rate of strain been uniformly maintained. In this way it has been possible to tremendously exaggerate the variations which it was desired to observe. These curves are plotted point to point from actual observations.

The area of these loops represents the energy lost in internal friction in the corresponding cycle. Converted into terms of inch-pounds of energy per cubic inch of material, these values for the different wires tested are given in Fig. 5. Dividing by the total elastic energy involved per cycle, the per cent loss may also be obtained.

The following general characteristics may be noted:

1 The energy loss increases sharply with stress, at a higher power than the square.

2 It was shown in Report No. 5 that for the heat-treated steel samples there tested, a sharp rise in hysteresis occurred on approaching the yield point of the metal. No such sharp rise has been noticed in the non-ferrous materials. This is a natural result of the nature of the materials, but may be partly connected also to the relative amount of cold drawing following heat treatment in the two cases.

3 The hysteresis loops are unsymmetrical and typically convex upward. In the steel samples, the loops are narrow enough so that they have a definite "string bean" shape, and so that a line midway between the two sides can be taken as approximating the true elastic line. For the phosphor-bronze and the relief-annealed brass wire the ascending line is definitely convex upward, while the descending lines are straight or slightly convex upward.

In successive loops, starting from the same nominal initial load, and rising to successively greater peak loads, the ascending lines essentially coincide, while the descending lines differ for the different loops.

Modulus of Elasticity

Measurements made specifically for the purpose of obtaining modulus of elasticity were, as already described and as indicated in Fig. 1, taken as a series of successive short loops. Measurements were in each case taken only after the specimen had been loaded and unloaded over the load range of that loop a sufficient number of times to place the sample in approximately cyclic condition. Some of the resulting values are given in Tables 3 and 4,[5] first in terms of the actual increments of load, and then in terms of the equivalent moduli of elasticity. These results are shown in graphical form in Figs. 6 and 7. In each case the rate of elastic elongation increases as the stress in tension increases.

Fig. 4 shows these same results for the steel wire in a somewhat different fashion. It also shows the magnitude of the permanent set resulting from the application of various loads to the wire. This did not reach 0.05 mm., or one part in 300,000 until the stress reached 160,000 lb. per sq. in. It will be noticed that both the width of the hysteresis loop (as given in Fig. 2) and the amount of the permanent set are small as compared to the variations in the elongation increments and to the offsets from initial tangent, so that these variations apparently represent the true elastic behavior of the materials rather than deviations from it.

[5] Not published in the present report.

TABLE 1 MODULI OF ELASTICITY OF SPECIMENS TESTED

MEASUREMENTS MADE USING AMSLER MACHINE

Specimen	Material	Temp., deg. fahr.	Modulus
111	0.68% C heat-treated steel wire, 1/8 in. × 1/2 in.	76	29,700,000 — 7.7 × S
112	Do.	76	30,100,000 — 6.8 × S
113	Do.	76	29,500,000 — 8.3 × S
530 } 538 }	3 1/2% nickel steel 1/8 in. diam.	78	29,140,000 — 4.3 × S
476	Do.	73	28,930,000 — 10.9 × S
306	Spring-tempered phosphor-bronze, 1/8 in. × 1/2 in.	72	14,680,000 — 10.8 × S

MEASUREMENTS MADE USING DEAD-WEIGHT LOADING[1]

481	0.67% C heat-treated steel wire, 0.0281 in. diam.	62.6	30,040,000 — 6.6 × S
479	Do., 0.0465 in. diam.	69.8	30,160,000 — 6.5 × S
483	17 SRT aluminum-alloy wire, 0.065 in. diam.	72.5	10,280,000 — 10.9 × S
493	Spring-temper brass wire as drawn, 0.040 in. diam.	72	14,610,000 — 7.1 × S
	Do., after 0.22 per cent permanent set	70	14,300,000 — 4.9 × S
485	Do., relief-annealed, 0.040 in. diam.	69	15,680,000 — 13.8 × S
494	Phosphor-bronze wire, 0.040 in. diam.	70	15,540,000 — 5.0 × S
	Do., after 0.16 per cent permanent set	70	15,380,000 — 4.5 × S

S = unit stress in tension, lb. per sq. in.

[1] The figures obtained using dead-weight loading and long gage length are somewhat more reliable than the other figures.

Due to the amount of elastic creep and recovery present, it is somewhat more difficult to specify true values for modulus of elasticity for the other materials. By varying the technique of operation, varying amounts of this elastic after-effect can be included in the readings, with resulting effect on the apparent modulus of elasticity. After some experimenting, it was found that reasonably consistent results could be obtained by taking the reading in each case at one minute after application of the load increment.

As obtained from the data so far obtained, the values of modulus of elasticity given in Table 1 are tentatively suggested. It should be emphasized that these values represent only the particular materials tested. A large amount of time and attention has been required for each series of tests, and therefore only a limited variety of samples have been tried. The investigation has not yet gone far enough to justify conclusions as to whether the rate of change of modulus of elasticity is a definite characteristic of the material or whether it varies with different types of heat treatment or of mechanical work. Further experiments are in progress for the purpose of covering some of these points.

A detailed study of the original data shows for each material a decrease of modulus of elasticity following overstresses great enough to produce a permanent elongation of the specimen. This is evident even for elongations no greater than one-tenth of 1 per cent, and even for material already severely cold-drawn. The per cent change in modulus is several times as great as the corresponding permanent set, even after all corrections have been made for the change in length and change in area of cross-section. There is a certain amount of evidence suggesting that in some cases with more severe overstressing there may be a reversal, the modulus increasing somewhat. A further knowledge of this factor is needed in order that we may understand the true stress conditions in cold-wound springs or in springs after surging and the investigation is being continued.

Discussion

R. L. TEMPLIN.[6] The author has given some very accurate and very interesting test results obtained from precise measurements of strain using very long gage lengths or specimens. The difficulties encountered in making such tests are quite numerous because factors which can frequently be ignored in the short-time tests may become of considerable significance in tests of the type represented.

It is generally accepted that at stresses near the elastic limit the modulus of elasticity of metals decreases quite rapidly, with the result that many investigators attempt to show that the stress-strain curve is curved from the origin. Dr. R. H. Canfield of the Naval Research Laboratories, Washington, D. C., pointed out before the Third International Congress, held at Stockholm last summer, that there appears to be a threshold stress below which the internal friction or slip was less than any measurable quantity. Interpreted in terms of Figs. 6 and 7 of the paper, this would mean that the curves would tend to become horizontal near the zero stress. Close scrutiny of the data presented by the author would appear to justify this conclusion. Fig. 7 shows data which exhibit this tendency to a marked degree, while there is less of a tendency in that direction shown in the data for Fig. 6.

R. W. CARSON.[7] The author is working on a problem which is interesting many manufacturers of precision apparatus. For rough-and-ready measurements of loads under the elastic limit, Hooke's law is usually satisfactory. However, when an accuracy greater than 1 or 2 per cent is required, these set and creep effects begin to manifest themselves. As the author indicates, the greater the sensitivity, the more involved becomes this elastic after-effect.

Is not the effect of moderate overstrain in cold-drawn phosphor-bronze wire to cause plastic flow in zones having high residual stresses, thus leaving the wire when returned to the unloaded condition with uniformly smaller residual stresses in the direction of the loading? If so, upon again moderately loading the wire in the same direction as that in which the over-stress was applied, the total of all the stresses across the section would then be smaller by an amount equal to the residual stresses relieved. Thus, the difference in elastic limit as noted in Fig. 7 would indicate the intensity of the residual stresses relieved rather than a change in the elastic properties of the material.

The same apparent change in modulus of elasticity might result from a mild heat treatment that would be just sufficient to allow a redistribution of high local stresses which would result in lower average stresses.

If the rate of creep or recovery action decreases with time and increases with stress, at some time very soon after the application or removal of a given load the creep or recovery action might equal the rate of purely elastic strain. Thus, the purely elastic and the creep or recovery action might be continuous and might be one and the same action over the entire time scale from the instant of applying load to the instant of further changing the load.

If such be the case, the width of the hysteresis loops would indicate only a relative amount of creep and recovery dependent on the time at which readings are taken. The recovery action might be of a slightly different order than the creep action, and in that event the reading time after applying load would vary for increasing or decreasing loads. Further, the hysteresis loop would not represent internal friction or work, since with sufficient time all of the work done in creep would be utilized in recovery.

W. G. BROMBACHER.[8] The fundamental nature of the author's work on the elastic behavior of spring materials should be emphasized. While the research does not now appear to have much application in many cases, undoubtedly, as with most fundamental research work, the results will be found to be use-

[6] Aluminum Research Laboratories, Aluminum Company of America, New Kensington, Pa.

[7] Meter Engineer, Westinghouse Elec. & Mfg. Co., Newark, N. J.

[8] Department of Commerce, Bureau of Standards, Washington, D. C.

ful. There is a tendency at the present time in metallurgical literature to avoid giving exact data on the moduli of elasticity. It is the practice to give either approximations or the two limiting values of the moduli. This is a clear recognition of the lack of precise knowledge as to the moduli of a given material under various conditions. The author's experiments are bringing light on this matter. His results show clearly that "elastic after-working" and "elastic hysteresis" must be considered in obtaining accurate and complete experimental data. These elastic effects have long been known and recognized by workers, especially those in the field of precision instruments. The first clear analysis of these phenomena was presented by Boltzmann, and it may be added that but very little improvement has been made up to the present in his theoretical presentation.

Special attention should be called to the author's method of experimentation since it is peculiarly well adapted for obtaining satisfactory data. It has two distinct advantages over most methods which should be clearly recognized. First, the specimen is uniformly stressed during the experiments. Second, the large length of the specimen, about 50 feet, permits results to be obtained to the essential high order of sensitivity with a minimum of expense and complication. The first advantage, of course, pertains only to the data obtained when the specimen is stressed in tension. The method will also enable data to be obtained for the torsional stresses, in which case the stressing will be non-uniform.

The lack of a satisfactory and universally accepted nomenclature should be stressed. Thus the author uses the term "elastic after-effect," which is referred to in the literature on the subject by such terms as "after-working," "drift" or "creep," and "recovery." These terms usually refer to the effect which is dependent on the time of application of the load as well as on the fiber stress and the material. The term "mechanical hysteresis" used in the report has been designated by other workers as "statical hysteresis," "elastic hysteresis," and "elastic lag due to elastic hysteresis." These terms usually refer to the effect which is independent of the time of the application of the load, but of course may depend on the fiber stress and the material. It should not be assumed that the preceding list of terms is complete, since many other expressions have been coined. It would appear desirable that workers in this field give consideration to obtaining the universal acceptance of a consistent nomenclature.

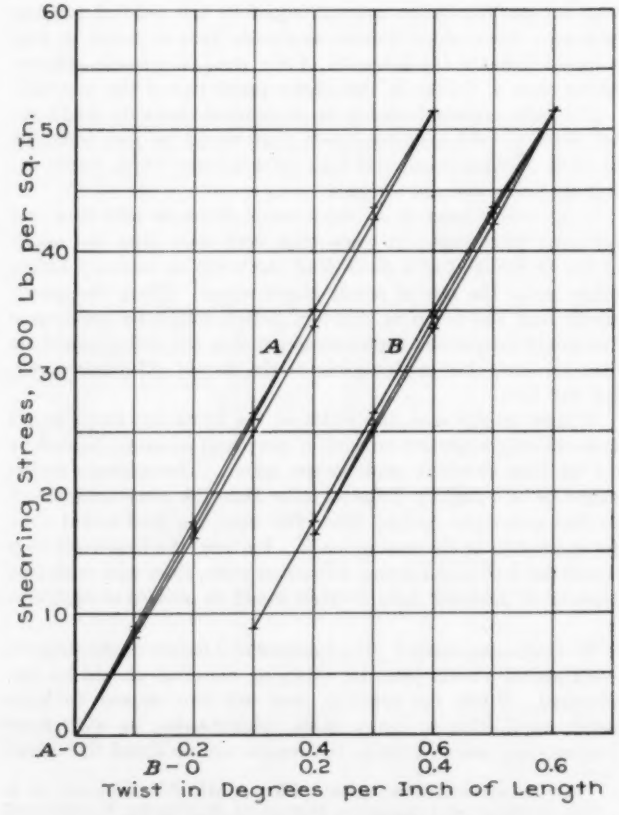


FIG. 8 HYSTERESIS CURVES—MILD STEEL

W. J. SWEETSER.[9] The author mentions the noticeable vibration set up in the test specimen when a truck passed at some distance from the building in which the work was being done. It seems probable that some of the apparently erratic behavior of the test specimen under constant load may be due to vibrations which are not detectable by ordinary means. The set-up of the test with suspended load is ideal for producing impact by vibration, and even unobserved vibrations of the foundation under the tank might give observable changes in elongation, especially when the order of magnitude of the measurements is so small.

Fig. 1 shows the curves for hysteresis and modulus-of-elasticity measurements, but in work which we have been doing at the University of Maine on steel in torsion, we have been unable to get such a series of readings as indicated at (*b*). Fig. 8 shows the hysteresis curve of an overstressed piece of mild steel in torsion. Three months before these curves were made the steel had been stressed to over 60,000 lb. per sq. in. in torsion and had been allowed to rest in the meantime. In curve *A*, the stress was increased gradually up to the maximum shown and then reduced to 0, giving a complete hysteresis loop. It was then restressed up to 35,000 lb. per sq. in. and returned to 0, giving the narrow hysteresis loop. The up curves in the two cases coincide. Curve *B*, shown at the right, shows a second hysteresis loop made by reducing the load from the maximum down to a stress of about 16,400 lb. per sq. in. and then returning to the maximum stress. In this case the curve coming down coincides with the original down side of the loop.

It is quite evident that the down side of the small loop in *A* is steeper than the down side of the larger loop, and similarly the up side of the small loop in *B* is steeper than the up side of the large loop, which would indicate that if a series of these small loops were put together as shown in (*b*), Fig. 1, the general slope of the series of small loops would not be the same as the mean slope of the curves shown at (*a*).

Taking the up curve of one of the small loops in (*b*), if it were continued it would tend to cross over and to the right of the coming down side of the loop above, and the writer is unable to see how this condition is possible.

D. E. ACKERMAN.[10] In his presentation of the paper the author mentioned that the curvature of the wire due to coiling on a drum was an important source of experimental error, and stated that several lots of wire had to be discarded because they had been coiled on too small a radius. It would be of interest to know the order of magnitude of the error, introduced in this way, as a function of the wire diameter and radius of curvature.

In Figs. 6 and 7 the author has indicated that the modulus-of-elasticity–tensile-stress relationship is linear. The writer would ask him whether the scattered data of curves 1 and 3,

[9] Professor of Mechanical Engineering, University of Maine, Orono, Me. Mem. A.S.M.E.

[10] Research Laboratory, International Nickel Co., Inc., Bayonne, N. J.

Fig. 7, are indicative of a high experimental error in these two cases, or whether the relationship for the corresponding materials simply follows a law of higher order; smooth curves can be drawn through the points plotted for the two curves in question. This point is of practical interest, for if the true relationship is of a higher order the plotted data suggest that at high tensile stresses the more severely cold-worked material may have the higher modulus of elasticity.

Author's Closure

One of our greatest difficulties in securing satisfactory results in these long-wire tests has been that of obtaining wire which was sufficiently straight for the purpose. The effect of local kinks or of curves in wire is, of course, greatest under light loads, the wire straightening out under heavier loads. As a result, there has been frequently found a spurious low modulus of elasticity under light loads. This has been very marked, indeed, in several cases, and unmistakable in its origin. It has therefore been necessary to discard altogether the results obtained from certain samples of wire and to use a grain of salt in interpreting other results where the author did not feel justified in discarding them entirely. Of the wires listed in Figs. 6 and 7, the steel ones were almost perfectly straight as received. The sample of aluminum wire which we were able to use was sufficiently straight so that it gave us very little trouble. The phosphor-bronze and brass wires were coiled on smaller-sized spools and so had a more pronounced curvature. This leads the author to regard with some suspicion the readings for these wires taken under light load.

On all the bona-fide straight samples which we have tested so far, our stress-strain line has been curved from the beginning. The one point of doubt remaining in the author's mind is as to the effect of elastic creep or elastic after-working upon the results. When a metal is loaded, most of the elongation occurs practically instantaneously, but it continues to elongate at a rapidly decreasing rate for five minutes or longer. This is true even at very low stresses. On removing the load an identical phenomenon occurs, but in the reverse direction. It is just possible that if our readings could be made coincidentally with the application of the load, and this after-effect entirely removed, our stress-strain line would be perfectly straight.

There are, however, very strong theoretical reasons for believing that this would not be so. For example, it has been definitely known for some time that the modulus of elasticity of a material under hydrostatic compression increases with increasing loads. The modulus in hydrostatic tension will obey the same equation as for hydrostatic compression, but with reversed sign, decreasing with increasing loads. Unidirectional tension is essentially a combination of hydrostatic tension and shear, and a direct relation should exist between the variation with stress of the moduli for hydrostatic tension and for direct tension. This is simply one of a series of theoretical reasons which make the author feel that it is highly unlikely that a stress-strain line for such relatively homogeneous materials as those which are here under investigation will be anything else but curved. Inhomogeneous materials such as wood, concrete, and cast iron also show a curved stress-strain line, but this curvature occurs for different reasons and is of an entirely different character, the modulus decreasing with increasing stress both in tension and in compression.

Reference may well be made to various comments on the erratic behavior shown by the plotted points in Fig. 7 for the 70–30 brass relief-annealed wire. A second wire of the same material, tested since December, has given the same curved line arrangement of points, so that it is apparently characteristic of the material. In certain of the other materials the decrease in modulus of elasticity at high stresses can be attributed to the permanent lowering of modulus which is likely to follow overstressing, but this does not appear to be the case in the wire just mentioned. No other material so far tested has duplicated this phenomenon, and the author does not as yet know how to explain it. It might, however, be well to notice that at low stresses the curved line which best represents the readings for wire No. 3 becomes asymptotic to the inclined line for specimen No. 1 rather than to a horizontal line.

The suggestion made by Mr. Carson that the creep or recovery action might be essentially continuous with the purely elastic action, is identical with the theories advanced by von Bennewitz in a paper in the *Physikalische Zeitschrift* for 1924. Von Bennewitz gave mathematical equations based on experiment to justify his theory, but these equations involved so much extrapolation beyond the limits of his experimental work that they have not been regarded with favor.

Figure 1(*b*) is misleading in that it shows the series of small loops as directly in line at their tips. Experimentally, when passing from cyclic condition for one load range to cyclic condition for a slightly higher load range, there is inevitably a certain amount of permanent, or semi-permanent, deformation so that two loops do not exactly match at their tips. This deformation will, the author believes, account for the discrepancies mentioned by Professor Sweetser as compared to his curves *A* and *B* in Fig. 8. When computing the modulus of elasticity, correction was made for this deformation so that figures were based on the effective slope of the individual small loops only.

APM-53-9

Forced Vibrations With Combined Coulomb and Viscous Friction

By J. P. DEN HARTOG,[1] EAST PITTSBURGH, PA.

Steady-state vibrations of mechanical systems damped by a combination of dry and viscous friction occur quite frequently, but until very recently no method for their calculation existed. A very satisfactory approximation was presented by L. S. Jacobsen before the A.S.M.E. in June, 1930. In the present paper the author gives the exact solution of the problem, together with a number of test results.



THE damping forces occurring in practical engineering vibration problems are of a great variety. The four types met most frequently are:

1 "Viscous" damping, where the force is proportional to the velocity, which fairly accurately describes the condition in viscous fluids

2 "Coulomb" damping, or dry friction, where the force is independent of the velocity, and depends only on the normal pressure

3 "Air resistance," which is assumed to be proportional to the square or some higher power of the velocity, and

4 "Internal hysteresis," where the force is independent of the velocity and depends only on the amplitude of motion.

The calculation of the steady-state vibrations of a damped system is possible with comparatively simple means only in the case of pure viscous damping, while for the other types of damping no solutions are known.

Whenever such problems were attacked in the past, the actual damping was usually replaced by some equivalent "viscous" damping, the amount of which was either assumed or calculated from an experiment.[2]

Recently a more rational method has been proposed for finding this "equivalent" viscous damping constant, consisting of calculating the amount of energy dissipated during a cycle of the vibration and adjusting the viscous damping constant to such a value that this dissipation in the calculated viscous case agrees with the actual case. This method was used by Southwell and Carter[3] for the case of Coulomb or dry friction at resonance; by Lewis[4] and Kimball[5] on internal hysteresis in metals, while Jacobsen[6] generalized the method for any damping proportional to the nth power of the velocity.

The accuracy of this general procedure, in the author's opinion, cannot be estimated unless its results are compared with those of an exact solution, which for most of the cases enumerated above has not been found as yet. Eckolt[7] has published a solution for a special case (motion with one stop) of pure Coulomb damping. His results are erroneous in several places. The author[8] has given a solution for another special case (non-stop motion) for combined Coulomb and viscous damping. The present paper, however, contains a complete solution of the problem.

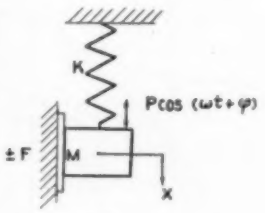


FIG. 1

FORCED VIBRATIONS WITH DRY FRICTION ONLY

Consider the simple system shown in Fig. 1, where a mass m is suspended from a spring with a stiffness of k lb. per inch and is acted upon by a periodic disturbing force $P \cos(\omega t + \varphi)$. In moving, the mass rubs against a solid wall which exerts a frictional force F on it opposing its motion but is otherwise constant and entirely independent of the velocity. After the system has been in motion for some time a "steady state" will have been reached which must satisfy the following conditions: (1) The frequency of the motion must be the same as that of the disturbing force; and (2) the downward half-cycle of the motion must follow the same law as the upward half-cycle. Figs. 2 and 3 show two possible types of such a motion.

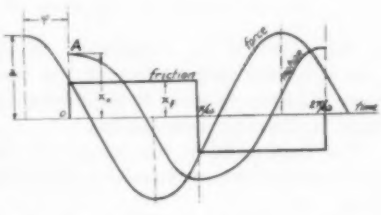


FIG. 2 DIAGRAM OF MOTION WITHOUT STOPS

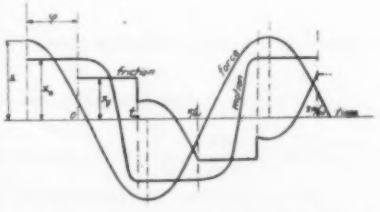


FIG. 3 DIAGRAM OF MOTION WITH ONE STOP

In Fig. 2 it is assumed that the mass never comes to a dead stop. Since the frictional force is always opposed to the motion it must be represented by the rectangular top curve shown in

---

[1] Engineer, Research Department, Westinghouse Electric & Manufacturing Company. Assoc-Mem. A.S.M.E. Mr. Den Hartog received the degree of E.E. from the University of Delft, Holland, in 1924. He entered the employ of the Westinghouse Electric and Manufacturing Company the following year and since then has been working on general mechanical problems, specializing in dynamics and mechanical vibrations.

[2] H. Holzer, "Die Berechnung der Drehschwingungen," Berlin, 1921. H. Wydler, "Drehschwingungen in Kolbenmaschinenanlagen," Berlin, 1922.

[3] Carter, "Torsional Vibrations in Engines, etc.," British Aeronautical Research Committee Report No. 1053 (E 22), 1926.

For presentation at the National Applied Mechanics Meeting, Purdue University, Lafayette, Ind., June 15 and 16, 1931, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. All papers are subject to revision.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[4] F. M. Lewis, "Torsional Vibration in the Diesel Engine," Trans. Soc. Naval Arch. & Marine Eng., 1925, p. 109.

[5] A. L. Kimball, "Vibration Damping, Including the Case of Solid Friction," Trans. A.S.M.E., 1929, APM-51-21.

[6] L. S. Jacobsen, "Steady Forced Vibration as Influenced by Damping," Trans. A.S.M.E., 1930, APM-52-15.

[7] W. Eckolt, "Ueber erzwungene Reibungsschwingungen," *Zeit. f. techn. Physik*, vol. 7, 1920, p. 226.

[8] *Phil. Mag.*, May, 1930, p. 801.

the figure. The spring force acting on the mass is k times the deflection or motion, so that the motion curve can be regarded as representing the spring force. It will be noticed that at the upper and lower peaks the motion curve suddenly changes its curvature. The explanation of this is that the curvature is proportional to the second derivative. The second derivative

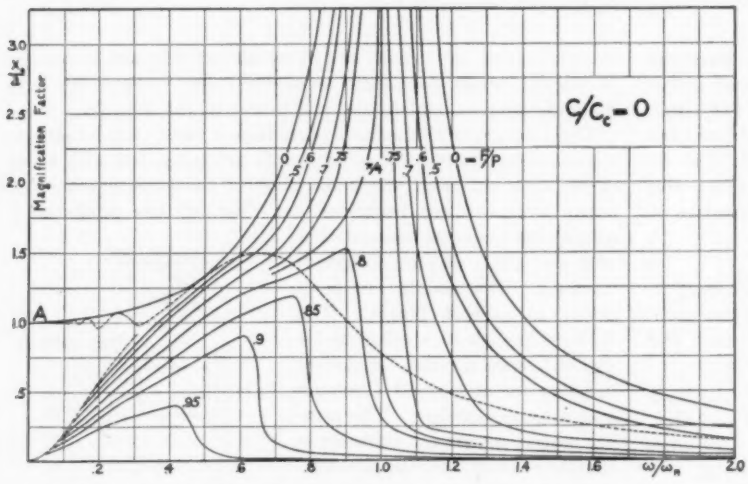


FIG. 4 AMPLITUDE DIAGRAM WITH COULOMB DAMPING ONLY

of the motion is the acceleration, which is proportional to the sum of all the forces acting on the mass. Since the frictional force suddenly reverses at the peaks, the curvature there must show a discontinuous change.

The amplitudes of the motion of the type shown in Fig. 1 have been solved analytically in Appendix No. 1, with the result that

$$\frac{x_0}{a} = \sqrt{V^2 - \left(\frac{F}{P}\right)^2 U^2} \quad \ldots\ldots\ldots\ldots [1]$$

where $\frac{x_0}{a}$ is the "magnification factor,"

$V = \frac{\beta^2}{\beta^2 - 1}$ the "response function,"

$U = \frac{\beta \sin \beta\pi}{1 + \cos \beta\pi}$ the "Coulomb damping function," and

$\beta = \omega_n/\omega$ the "frequency ratio."

These expressions will become clear from an inspection of Fig. 4, where the numerical results of [1] are plotted in a family of curves. Only the part of the diagram above the broken line corresponds to Formula [1] and Fig. 2.

The "frequency ratio" is the ratio between the natural frequency of the undamped system and the frequency of the disturbing force.

The "magnification factor" is the ratio between the actual amplitude of the motion and the amplitude which would occur without friction and with an infinitely slow motion.[9] It can be easily seen from Fig. 1 that this latter "statical" amplitude must be equal to P/k. In Fig. 4 it is represented by the point A.

The "response function" is the magnification factor as a function of the frequency when there is no friction. In Fig. 4 it is the upper curve of the diagram. It is a well-known and fundamental curve in all vibration investigations.[10]

In Formula [1] there are two terms under the radical sign. Both are squares and therefore necessarily positive. It is seen from the formula that without damping ($F/P = 0$) the magnification factor is given by the response function V, and that for any value of the friction, F/P the magnification factor is smaller than without damping, as it should be. A remarkable property of the solution [1] is that for any friction ratio F/P smaller than $\pi/4$ or 0.785 the amplitude becomes infinitely large. In other words, even with a considerable amount of dry friction, the amplitudes at resonance still extend to infinity. This apparently paradoxical fact can be understood by considering the energy involved as follows:

Referring to Fig. 1, before the system, excited at resonance ($\omega = \omega_n$), reaches the "steady state," the energy per cycle put into the system by the force $P \cos(\omega t + \varphi)$ will be proportional to the product of the force and the amplitude,[11] and since the force does not depend on the amplitude, the energy input per cycle will be proportional to the first power of the amplitude. The work absorbed by the constant damping force F will be also proportional to the amplitude for the same reason. Both will be represented by straight lines in Fig. 5(a). The slope of the input curve depends on P and the slope of the dissipation curve on F. As long as F is small enough to make the slope of the dissipation curve smaller than that of the input curve (which happens for $F/P < \pi/4$) there will be always positive work input into the system, which will make the amplitude larger and larger.

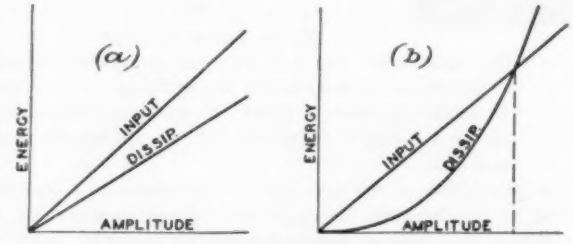


FIG. 5 ENERGY RELATIONS AT RESONANCE

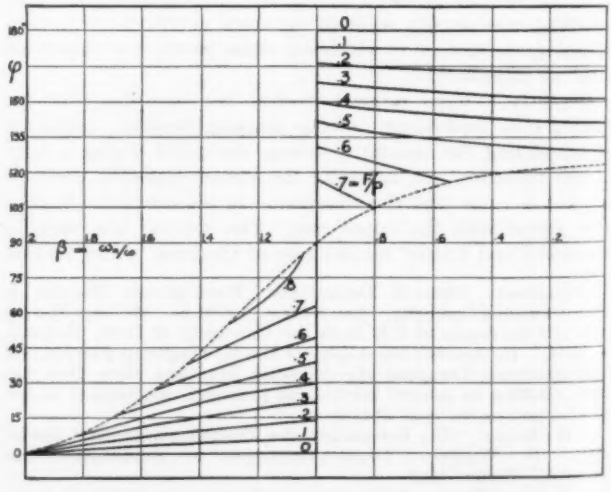


FIG. 6 PHASE-ANGLE DIAGRAM WITH COULOMB DAMPING ONLY

[9] S. Timoshenko, "Vibration Problems in Engineering," p. 9.

[10] Timoshenko, loc. cit., Fig. 19 on p. 29.

[11] "Torsional Vibration Dampers," Trans. A.S.M.E., 1930, APM-52-13, formulas [4] and [5].

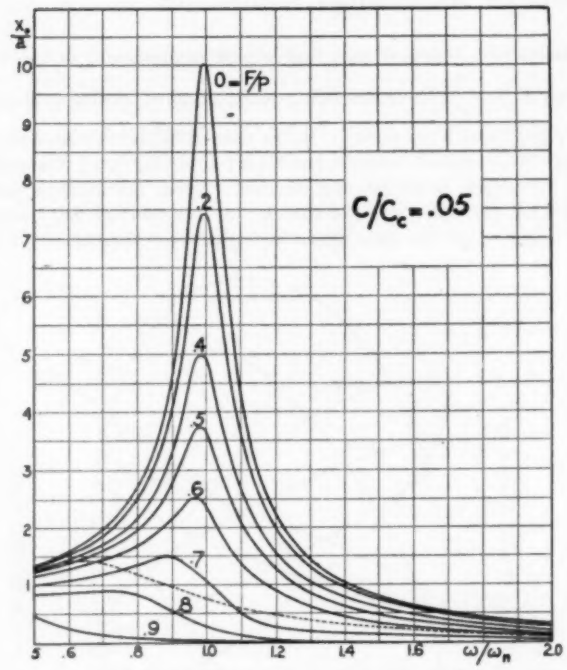


FIG. 7 AMPLITUDE DIAGRAM WITH MIXED DAMPING

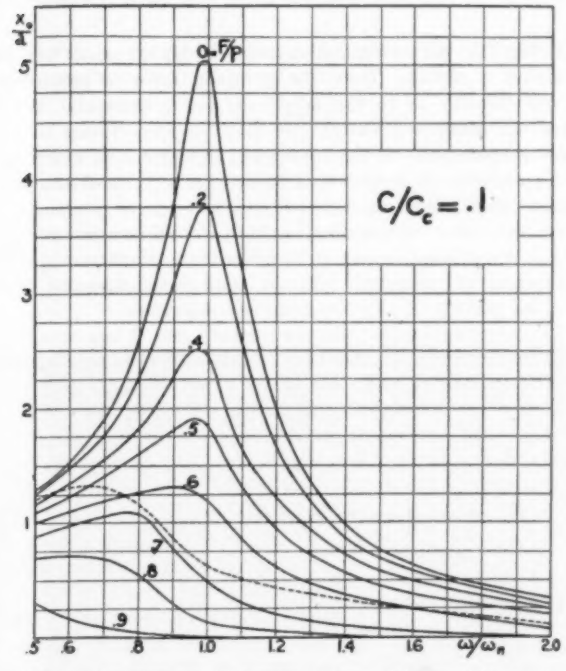


FIG. 8 AMPLITUDE DIAGRAM WITH MIXED DAMPING

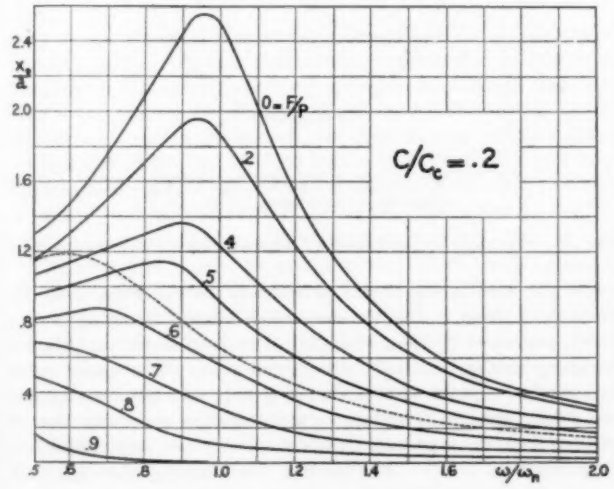


FIG. 9 AMPLITUDE DIAGRAM WITH MIXED DAMPING

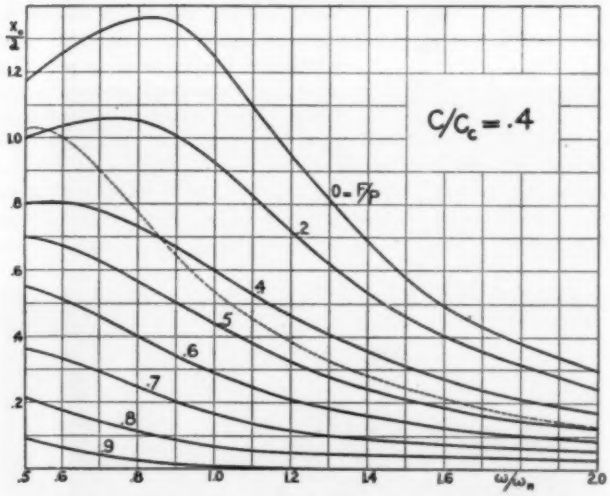


FIG. 11 AMPLITUDE DIAGRAM WITH MIXED DAMPING

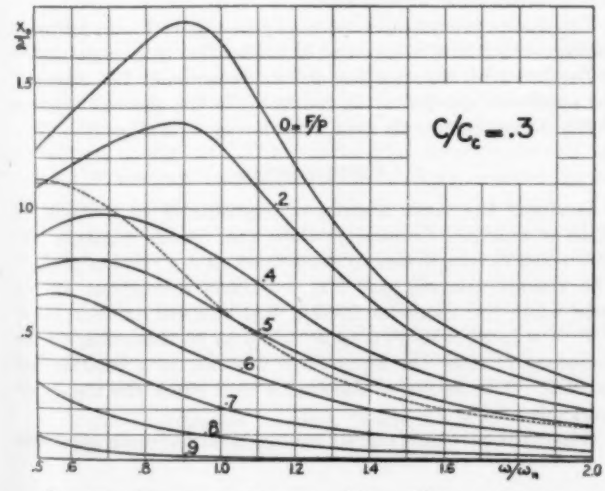


FIG. 10 AMPLITUDE DIAGRAM WITH MIXED DAMPING

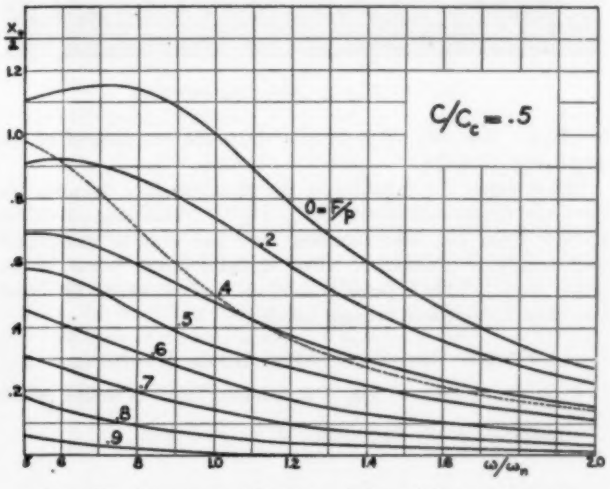


FIG. 12 AMPLITUDE DIAGRAM WITH MIXED DAMPING

In Fig. 5(*b*) the corresponding condition for the usual "viscous" damping is shown. Here the damping force is proportional to the velocity or to the amplitude for a sinusoidal motion. Thus the dissipated energy per cycle is proportional to force times amplitude or to the square of the amplitude, which plots as a parabola. It is seen that even for a very small amount of viscous friction the input and dissipation curves intersect. At the amplitude corresponding to this point of intersection there will be energy equilibrium, so that this is the amplitude at which the system will ultimately vibrate. For viscous damping, therefore, no infinite amplitudes occur at resonance.

At other frequencies than the resonant one, it has been found that an increase in the damping force F will decrease the amplitude of motion (Fig. 2) and will also reduce the curvature at A in Fig. 2. When F reaches a certain value, this curvature becomes zero, and for larger values of F than this, the motion of Fig. 2 cannot exist and will be replaced by that of Fig. 3. During each half-cycle the mass m will stop moving for a while, and while stopping the value of the friction force may be anything between $+F$ and $-F$. Since the mass m during this interval has no acceleration, the sum of the disturbing force $P\cos(\omega t + \varphi)$, the spring force $-kx$, and the friction force must be zero. This determines the friction force as shown in Fig. 3. Motions of this type are also solved analytically in Appendix No. 1, though the result cannot be put in an explicit form like [1]. The calculation has to be done point by point, with the result shown in the curves below the broken line of Fig. 4.

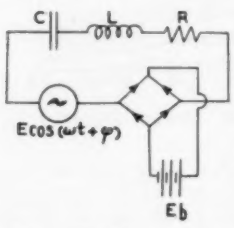


Fig. 13 Equivalent Electrical System

It will be noticed that there is a somewhat triangular area at the left of Fig. 4 which is not filled in. In this area no motions of the two types discussed are possible, but motions with more than one stop per half-cycle must occur. This could not be demonstrated analytically because of the great complications involved. However, experiments show the correctness of the conclusion (Fig. 18).

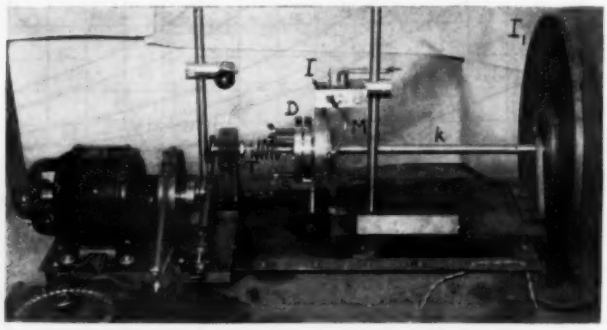


Fig. 14 Model for Experiments

Fig. 6 shows the behavior of the phase angle φ between the force and the motion as defined by Fig. 2. This diagram differs greatly from the corresponding one for viscous damping,[12] and mainly at resonance, where definite jumps occur in the phase angle. This is possible only on account of the fact that the amplitudes there become infinite.

Forced Vibrations With Mixed Damping

If on the system of Fig. 1 a damping force is acting which consists of a constant part and a part proportional to the velocity $F + c\frac{dx}{dt}$, the same two general types of motion as shown in Figs. 2 and 3 can occur. In the case of motion without stops (Fig. 2) an explicit solution has been found (Eq. [31], Appendix No. 2) which is rather complicated. Again for motions with one stop per half-cycle the calculation can be done only point by point. All the derivations are given in Appendix No. 2,

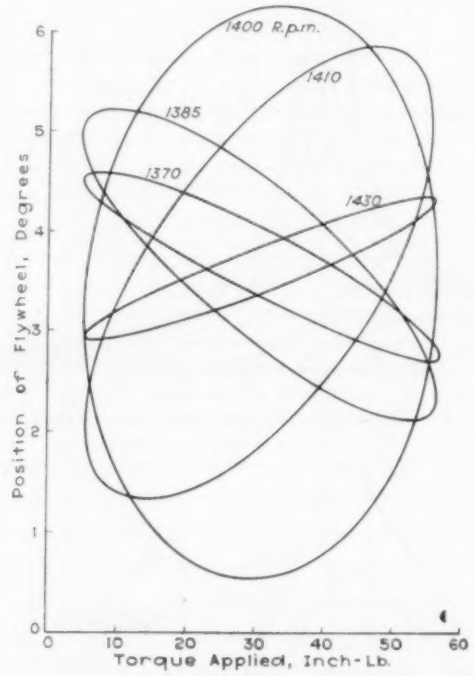


Fig. 15 Records With Viscous Damping Only Near Resonance

while the results are shown graphically in the Figs. 7 to 12. In each of these figures the amount of viscous damping is constant, expressed by c/c_c, which is the ratio of the actual viscous damping to the "critical" viscous damping. The upper curve in each figure is for viscous damping only and has been known for a long time.[10] The lower curves are for a combination of viscous friction (expressed by c/c_c) and dry or Coulomb friction (expressed by F/P). The broken line running through each figure denotes the boundary between non-stop and one-stop motion; the type of motion of Fig. 2 occurring above and that of Fig. 3 below the broken line.

The results given in the various figures apply to any physical problem which can be regarded as a vibration of a single degree of freedom with the damping under discussion.[8] For instance, the maximum charge on the condenser in the electrical system of Fig. 13 can be directly read from the curves of Figs. 6 to 12.

Experimental

Jacobsen[8] has made a series of experiments on a vibrating table at Stanford University where the viscous damping was very small in comparison to the amount of dry friction present. The viscous damping was $c/c_c = 0.0058$ throughout all the tests while the Coulomb friction was changed in four steps. $F/P = 0.21\ \beta^2$, $0.30\ \beta^2$, $0.48\ \beta^2$, and $0.65\ \beta^2$, depending on the frequency β, since the input force was due to a rotating unbalance. The results of Jacobsen's tests check the theory remarkably well.

Some tests have been carried out by the author on an apparatus

[12] Timoshenko, loc. cit., p. 30.

which was primarily designed for the investigation of torsional-vibration dampers.[13] Fig. 14 is a photograph of the apparatus, and it is seen that the system, instead of being a linear one as in Fig. 1, is a torsional one. The flywheel I takes the place of the mass m; the shaft k represents the spring and the flywheel I_1, in this case clamped to the frame, represents the solid wall at the top of Fig. 1. The two disks D are pressed against a third one between them, which is rigidly attached to I and lined

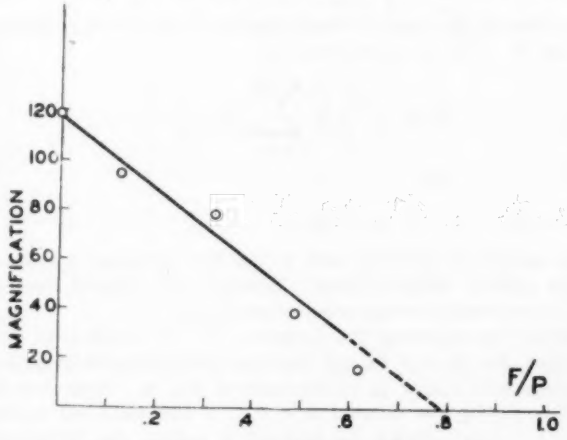


FIG. 16 RESULTS OF TESTS AT RESONANCE

with brake lining. The pressure is adjustable by means of the springs S. For this experiment, the moment of inertia of the disks D is made practically infinite by attaching to them two very long arms with weights at the ends, not shown in the photograph. This transforms the disks D into a solid rubbing wall as shown in Fig. 1. The harmonic force P in Fig. 1 here is a harmonic torque supplied by the spring T, one end of which is moved back and forth through an angle of about 30 deg. by means of an eccentric on the motor shaft.

The recording is done by means of a small mirror M mounted on universal pivots. This mirror is oscillated about a vertical axis by a wire driven by the moving end of the spring T. When the machine is running and there is no vibration in I, a ray of light reflected from the mirror describes a horizontal path on a screen or photographic plate. The vibration is transmitted by another wire to the mirror in such a way that when I is oscillating and the motor is standing still, a vertical path is described on the screen by the light ray. In any actual case the combination will be a Lissajous figure.

Fig. 15 shows several of such figures taken at various speeds near resonance with a small amount of viscous damping, and no Coulomb damping, giving pure ellipses. The shift in the phase angle is roughly indicated by the inclination of the ellipse. Records like this have been taken for several values of F/P, namely, for $c/c_c = 0.0042$ and $F/P = 0$, 0.12, 0.31, 0.48, and 0.62. The theoretical curve and the five experimental points are shown in Fig. 16. The experimental determination of the friction F was not very accurate, which may account for the discrepancy between theory and test shown in the figure. Fig. 17 is a record taken at a much larger magnification in the vertical direction than in Fig. 15, at approximately half critical speed $\omega/\omega_n = 0.5$. It illustrates a motion with one stop per half-cycle of the type shown in Fig. 3. Fig. 18 shows a motion having two stops per half-cycle with rather small friction and very slow speed. This figure would be represented by a point in the triangular area at the left of Fig. 4, and it is an experimental verification of the conclusion that in this area only motions with more than one stop are possible.

[13] "Torsional-Vibration Dampers," Trans. A.S.M.E., 1930, APM-52-13, closure to the discussion.

CONCLUSIONS

1 A solution has been found for the problem of the steady-state vibration of a system with a single degree of freedom, damped by a combination of viscous and Coulomb friction.

2 The theoretical results are presented in the diagrams of the Figs. 4 and 6 to 12, inclusive.

3 The results of tests on a model illustrated by Figs. 14 to 18 are in fair agreement with the theory.

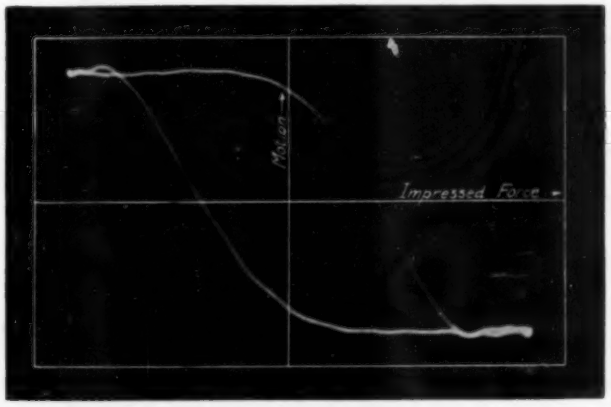


FIG. 17 RECORD OF MOTION WITH ONE STOP PER HALF-CYCLE

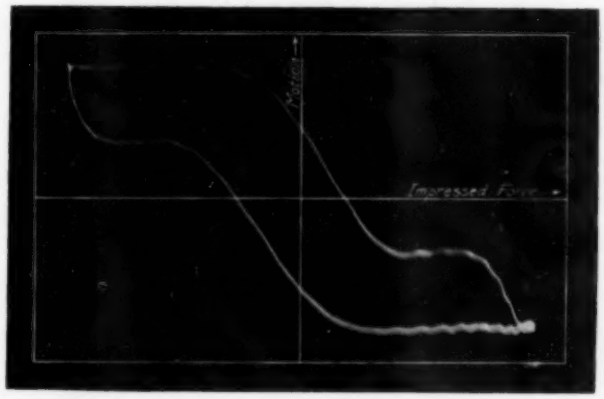


FIG. 18 RECORD OF MOTION WITH TWO STOPS PER HALF-CYCLE

Appendix No. 1

FORCED AMPLITUDES WITH COULOMB DAMPING ONLY

The differential equation of motion of the mass of Fig. 1 are:

$$m\ddot{x} + kx \pm F = P \cos(\omega t + \varphi) \quad \ldots\ldots\ldots\ldots [2]$$

where the + sign before the damping force F holds when the mass moves in the + x-direction, and vice versa. The "phase angle" φ in the disturbing force has no meaning in a differential equation and is only included for the purpose of subsequently writing the boundary conditions [4] in a simple form. During the half-cycle of the motion $0 < t < \pi/\omega$ of Fig. 2 the velocity is always negative so that [2] holds with the — sign before F. Using the abbreviations:

$$P/k = a, \quad F/k = x_f, \quad k/m = \omega_n^2$$

the differential equation for that interval of time can be written as:

$$\ddot{x} + \omega_n^2 (x - x_f) = a\omega_n^2 \cos(\omega t + \varphi) \quad [3]$$

A "steady-state solution" as pictured in Fig. 2 has to fulfil the following boundary conditions:

$$\left.\begin{array}{llll} t = 0 & x = x_0 & \dot{x} = 0 \\ t = \pi/\omega & x = -x_0 & \dot{x} = 0 \end{array}\right\} \quad [4]$$

These are four conditions, and the general solution of [3] contains only two integration constants. The two superfluous equations will be used for determining the unknown quantities x_0 and φ, that is, the amplitude and the phase angle.

The general solution of [3] is:

$$x = C_1 \sin \omega_n t + C_2 \cos \omega_n t + aV \cos(\omega t + \varphi) + x_f \quad [5]$$

where V, the "response function," is defined for [1]. By means of the two conditions at $t = 0$ in [4] the constants C_1 and C_2 can be solved, giving the result

$$x = x_0 \cos \omega_n t + x_f(1 - \cos \omega_n t) + aV\left[\cos\varphi(\cos\omega t - \cos\omega_n t) + \sin\varphi\left(\frac{1}{\beta}\sin\omega_n t - \sin\omega t\right)\right] \quad [6]$$

where $\beta = \omega_n/\omega$ is the frequency ratio. Now the two conditions [4] at $t = \pi/\omega$ can be substituted in [6], giving two equations which can be written in the form:

$$\left.\begin{array}{l} A\cos\varphi + B\sin\varphi + C = 0 \\ P\cos\varphi + Q\sin\varphi + R = 0 \end{array}\right\} \quad [7]$$

where

$$\begin{aligned} A &= -aV(1 + \cos\beta\pi) \\ B &= +aV\frac{\sin\beta\pi}{\beta} \\ C &= x_0(1 + \cos\beta\pi) + x_f(1 - \cos\beta\pi) \\ P &= aV\omega_n \sin\beta\pi \\ Q &= aV\omega\,(1 + \cos\beta\pi) \\ R &= (x_f - x_0)\omega_n \sin\beta\pi \end{aligned}$$

From [7] we deduce that

$$\cos\varphi = \frac{BR - CQ}{AQ - PB} \quad \text{and} \quad \sin\varphi = -\frac{CP - AR}{AQ - PB}$$

Substituting the values A, B, etc., in these formulas and reducing they can be written:

$$\cos\varphi = \frac{x_0}{a}\frac{1}{V} \quad \text{and} \quad \sin\varphi = -\frac{x_f}{a}\frac{U}{V} \quad [8]$$

where the "damping function" U has been defined for [1]. Elimination of φ from the two Equations [8] gives

$$\frac{x_0}{a} = \sqrt{V^2 - \left(\frac{x_f}{a}\right)^2 U^2} \quad [9]$$

which is the amplitude equation (same as [1]) for non-stop motions of the type of Fig. 2.

Substantially the same result can be obtained in another way. It is possible to write the differential equation [2] without the $\pm$ ambiguity by expressing the square-top damping curve of Fig. 2 analytically as a Fourier series:[14]

$$m\ddot{x} + kx - \frac{4F}{\pi}\sum_{n=1,3,5\ldots}\frac{1}{n}\sin n\omega t = P\cos(\omega t + \varphi) \quad [10]$$

The steady-state solution of this equation is:

$$\frac{x}{a} = V\cos(\omega t + \varphi) + \frac{4}{\pi}\frac{F}{P}\beta^2\sum_{1,3,5\ldots}\frac{\sin n\omega t}{n(\beta^2 - n^2)} \quad [11]$$

The phase angle φ must be such that at $t = 0$, $\dot{x} = 0$, as shown in Fig. 2. This condition leads to

$$\left.\begin{array}{l} \sin\varphi = \dfrac{4}{\pi}\dfrac{F}{P}\dfrac{\beta^2}{V}\displaystyle\sum_{1,3,5\ldots}\frac{1}{\beta^2 - n^2} \\ \text{and} \\ x_0 = aV\cos\varphi \end{array}\right\} \quad [12]$$

This solution is identical with [1] (or [9]), although it appears in an entirely different form. However, the original formula [1] is preferable for numerical computations.

Before investigating the boundary of the validity of this solution, let us pass on and find the steady-state solution for motions with one stop, of the type of Fig. 3. Here there is motion during the interval $0 < t < t_0$ and standstill during $t_0 < t < \pi/\omega$; during the interval of motion the differential equation [3] is valid, while the boundary conditions are:

$$\left.\begin{array}{llll} t = 0 & x = x_0 & \dot{x} = 0 \\ t = t_0 & x = -x_0 & \dot{x} = 0 \end{array}\right\} \quad [13]$$

The problem is the same as before, with the exception that there is a fifth unknown quantity, t_0. The additional relation necessary for its determination can be found from a consideration of what happens just before $t = 0$ in Fig. 3. The mass is in rest and consequently has no inertia force. The sum of the disturbing force, spring force, and damping force, therefore, has to be zero. At the instant $t = 0$ this still must be true, and moreover the amount of damping force is then known to be F. This gives the desired relation: $-kx + F + P\cos(\omega t + \varphi)$ must vanish at $t = 0$, or

$$-kx_0 + F + P\cos\varphi = 0 \quad [14]$$

The conditions [13] and [14] are sufficient to determine the five quantities C_1, C_2, x_0, φ, and t_0. First C_1 and C_2 are found from the first two conditions [13], giving the result [6]. With these the last two conditions [13] can be written as:

$$0 = x_0(1 + \cos\omega_n t_0) + x_f(1 - \cos\omega_n t_0) + aV\left[\cos\varphi(\cos\omega t_0 - \cos\omega_n t_0) + \sin\varphi\left(\frac{1}{\beta}\sin\omega_n t_0 - \sin\omega t_0\right)\right]$$

and

$$0 = (x_f - x_0)\beta\sin\omega_n t_0 + aV[\cos\varphi\{\beta\sin\omega_n t_0 - \sin\omega t_0\} + \sin\varphi\{\cos\omega_n t_0 - \cos\omega t_0\}]$$

Now [14] is used to eliminate x_0 from the last two equations by direct substitution, giving:

$$0 = 2\frac{x_f}{a} + \cos\varphi\left[1 + \cos\omega_n t_0 + V(\cos\omega t_0 - \cos\omega_n t_0)\right] + \sin\varphi V\left[\frac{1}{\beta}\sin\omega_n t_0 - \sin\omega t_0\right] \quad [15]$$

[14] This was done independently by Prof. E. O. Waters and the author; see discussion to Jacobsen's paper, loc. cit.

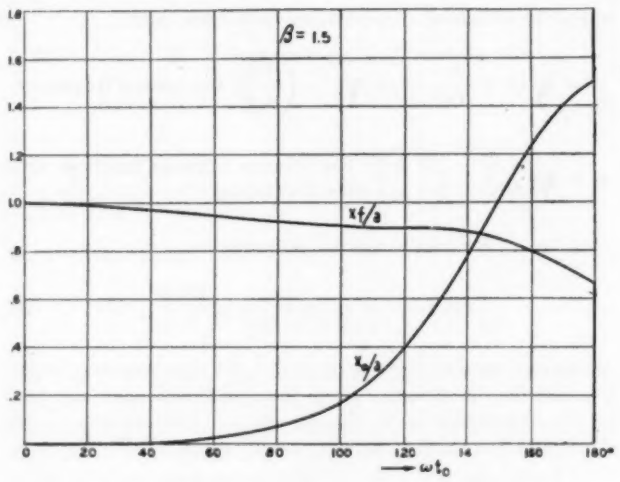


FIG. 19 DIAGRAM FOR STOP MOTION ($\beta = 1.5$)

and

$$0 = \cos\varphi\left[\frac{1}{\beta}\sin\omega_n t_0 - \sin\omega t_0\right] + \sin\varphi\left[\cos\omega_n t_0 - \cos\omega t_0\right] \quad \ldots\ldots [16]$$

From [16]

$$\cos\varphi = \frac{\cos\omega_n t_0 - \cos\omega t_0}{\sqrt{\left(\cos\omega_n t_0 - \cos\omega t_0\right)^2 + \left(\frac{1}{\beta}\sin\omega_n t_0 - \sin\omega t_0\right)^2}} \quad [17]$$

and substituting this in [15] we obtain an equation in which t_0 is the only unknown:

$$\frac{2}{V}\frac{x_f}{a} = \frac{\left(\cos\omega t_0 - \cos\omega_n t_0\right)\left(\frac{1}{V} + \cos\omega t_0 - \frac{1}{\beta^2}\cos\omega_n t_0\right) + \left(\sin\omega t_0 - \frac{1}{\beta}\sin\omega_n t_0\right)^2}{\sqrt{\left(\cos\omega t_0 - \cos\omega_n t_0\right)^2 + \left(\sin\omega t_0 - \frac{1}{\beta}\sin\omega_n t_0\right)^2}} \quad \ldots\ldots [18]$$

It would be logical next to solve t_0 from [18] and substitute it in [17] and then in [14] to find the expression for x_0. However, [18] is transcendental and an explicit solution is not possible. Instead [18] is used for a numerical calculation of x_f/a for a number of values of t_0 between 0 and π. By means of [17] and [14] the corresponding values of x_0/a are calculated. Fig. 19, for example, shows the result of such calculations for $\beta = 1.5$. From this diagram and from similar ones for other β-values the diagram of Fig. 4 has been plotted.

There is one slight complication in this process. It is seen that [18] takes the form 0/0 for $\beta = 1$ or $\omega = \omega_n$. In this case it is necessary to substitute in [18] $\beta = 1 + \delta$, where δ is a small quantity, and evaluate the limit when δ approaches zero. This gives

$$4\frac{x_f}{a} = \frac{(\omega t_0 + \sin\omega t_0)^2}{\sqrt{(\omega t_0\sin\omega t_0)^2 + (\sin\omega t_0 - \omega t_0\cos\omega t_0)^2}} \quad \ldots\ldots [19]$$

from which the diagram for $\beta = 1$ has been constructed.

Now, since the amplitude formulas for the two types of motion have been determined, it becomes necessary to discuss their validity. First let us return to the non-stop formulas [1], [6], and [8]. These have been derived under the assumption that $x < 0$ for $0 < t < \pi/\omega$; consequently they are valid only under this condition. By differentiation of [6] this condition becomes:

$$\frac{x_0}{x_f} \geqslant \beta^2\left[\frac{\beta\sin\omega_n t + U(\cos\omega t - \cos\omega_n t)}{\beta^2\sin\omega t}\right] \quad \ldots\ldots [20]$$

for $0 < t < \pi/\omega$

It can be verified that the bracketed portion of this expression becomes unity for $t = 0$. It has been found by numerical calculation that for all values of $\beta < 2$ this is also the absolute maximum of the bracketed quantity. For $\beta > 2$, the absolute maximum during $0 < t < \pi/\omega$ is sometimes unity, and sometimes greater. It has been found convenient to denote this value by S, so that [20] can be written:

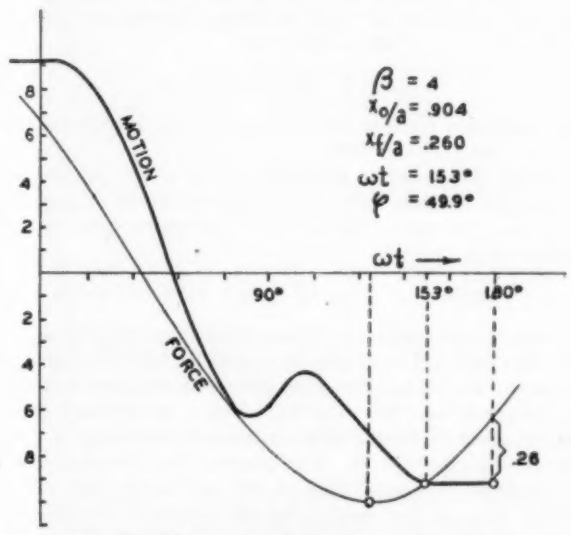


FIG. 20 MOTION VIOLATING CONDITION [19]

$$\frac{x_0}{x_f} \geqslant \beta^2 S \quad \ldots\ldots [20a]$$

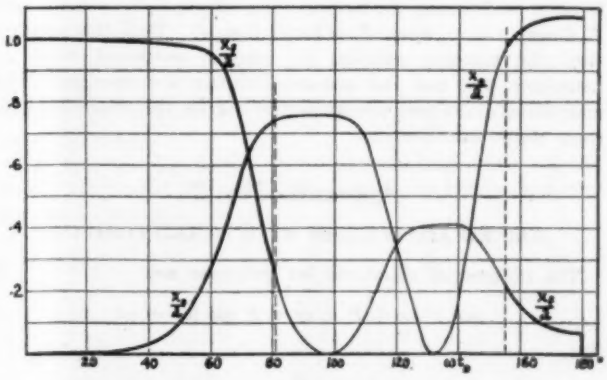


FIG. 21 DIAGRAM FOR STOP-MOTION ($\beta = 4$)

The combination of Equation [20a] with Equation [1] leads moreover to the following expressions for the condition of validity of [1]:

$$\frac{x_0}{a} \geqslant \sqrt{\frac{V^2}{1 + (U/S\beta^2)^2}} \quad \ldots\ldots [21]$$

$$\frac{x_f}{a} \leqslant \sqrt{\frac{V^2}{(S\beta^2)^2 + U^2}} \quad \ldots\ldots\ldots\ldots [21a]$$

With the aid of these formulas, the boundary of the region of validity of the non-stop motion has been plotted in Fig. 4.

With $S = 1$ the formulas [20a], [21], and [21a] also give the limit of the motion of Fig. 3. If in [17] and [18] we substitute $\omega t_0 = \pi$, the stop shrinks to zero and by means of [14] the results [20], [21], and [21a] are obtained. This completely determines the lower boundary of the non-stop motion in Fig. 4.

Let us next consider the validity of the solution [17], [18]. Since each half-cycle is made up of two distinct parts, a condition will be found for each of them. During the period of motion $0 < t < t_0$ the condition is:

$$x < 0 \quad \text{for} \quad 0 < t < t_0 \ldots\ldots\ldots\ldots [22]$$

The expression for $\dot{x}$ is found from [6] after substitution of φ and x_0 from [17] and [18].

During the period of standstill $t_0 < t < \pi/\omega$ the condition is that the absolute value of all external forces acting on the mass—i.e., the disturbing force and the spring force—shall be smaller than F:

$$|\cos(\omega t + \varphi) + x_f/a| \leqslant x_f/a \text{ for } t_0 < t < \pi/\omega \ldots [23]$$

In order for the solution to have a physical meaning, it is necessary that both [22] and [23] be satisfied. Fig. 20 gives an example of a motion satisfying the differential equation but violating the condition [20]. Fig. 21, which is of the same type as Fig. 19, gives the results of the numerical calculation of [16] to [18] for the case $\beta = 4$. The solution thus obtained is valid for values of ωt_0 smaller than 81 deg. and larger than 155 deg. For values of ωt_0 just over 81 deg. the condition [23] during the stop is violated, while for angles just below 153 deg. the condition [22] during the motion period is violated as Fig. 20 illustrates. It is seen that for $\beta = \omega_n/\omega = 4$ no solution can be found for $x_f/a = F/P$ between 0.22 and 0.26. In this case, no motion of either the types of Fig. 2 or 3 is possible, and it seems that the only possibility is a motion with more than one stop per half-cycle. This has been verified by experiment as shown in Fig. 18.

In the previous analysis the friction has been assumed to be entirely independent of the velocity. Sometimes the physical conditions can be better described by assuming the friction force to be F during the motion independent of the velocity and F_s during a stop, where F_s is larger than F. With this assumption, the non-stop solution [1] remains unaltered, while its boundary [21] and the one-stop solution are changed. The analysis as given here can be carried out for this case also, with some slight modifications.

Appendix No. 2

FORCED AMPLITUDES WITH MIXED DAMPING

The differential equations for this case are:

$$m\ddot{x} + kx \pm F + c\dot{x} = P\cos(\omega t + \varphi)$$

or

$$\ddot{x} + \omega_n^2 (x - x_f) + \frac{c}{m}\dot{x} = a\omega^2 \cos(\omega t + \varphi) \ldots\ldots [24]$$

when only motions with $x < 0$ are considered. The analysis of this equation runs parallel to that of the one in Appendix No. 1. The general solution of [24] is known to be:

$$x = e^{-\frac{c}{c_c}\omega_n t}\left[C_1 \cos pt + C_2 \sin pt\right] + \frac{a}{q}\sin(\omega t + \epsilon) + x_f \ldots [25]$$

where the following abbreviations have been used:

$p = \sqrt{\omega_n^2 - \frac{c^2}{4m^2}} = \omega_n \sqrt{1 - \left(\frac{c}{c_c}\right)^2}$, the natural frequency

$q = \sqrt{\left(\frac{1}{V}\right)^2 + \left(\frac{2}{\beta}\frac{c}{c_c}\right)^2}$, the inverse response function with viscous damping

$c_c = 2m\omega_n$, the critical viscous damping.

$$\tan(\varphi - \epsilon) = \frac{c\omega}{k - m\omega^2} = \frac{2V}{\beta}\frac{c}{c_c} \ldots\ldots [26]$$

Assuming a motion like that in Fig. 2, the four boundary conditions [4] hold as before. These four conditions again are used for the calculation of C_1, C_2, x_0, and φ. First C_1 and C_2 are eliminated by means of the two conditions at $t = 0$:

$$x = e^{-\frac{c}{c_c}\omega_n t}\left[(x_0 - x_f)\cos pt - \frac{a}{q}\cos\epsilon\cos pt + \frac{c}{2pm}(x_0 - x_f)\right.$$
$$\left.\sin pt - \frac{ac}{2pqm}\cos\epsilon\sin pt + \frac{a\omega}{pq}\sin\epsilon\sin pt\right]$$
$$+ \frac{a}{q}\cos(\omega t + \epsilon) + x_f \ldots [27]$$

Then the two conditions at $t = \pi/\omega$ can be written in the form

$$\left.\begin{aligned} A\cos\epsilon + B\sin\epsilon + C &= 0 \\ P\cos\epsilon + Q\sin\epsilon + R &= 0 \end{aligned}\right\} \ldots\ldots [28]$$

where

$$A = -\frac{a}{q}e^{-\frac{\pi c}{2m\omega}}\cos\frac{p\pi}{\omega} - \frac{ac}{2pmq}e^{-\frac{\pi c}{2m\omega}}\sin\frac{p\pi}{\omega} - \frac{a}{q}$$

$$B = \frac{a\omega}{pq}e^{-\frac{\pi c}{2m\omega}}\sin\frac{p\pi}{\omega}$$

$$C = (x_0 - x_f)e^{-\frac{\pi c}{2m\omega}}\cos\frac{p\pi}{\omega} + \frac{c}{2pm}(x_f - x_0)e^{-\frac{\pi c}{2m\omega}}\sin\frac{p\pi}{\omega} + x_f + x_0$$

$$P = \frac{a}{q}e^{-\frac{\pi c}{2m\omega}}\sin\frac{p\pi}{\omega}\left[\frac{c^2}{4m^2p} + p\right]$$

$$Q = \frac{a\omega}{q}\left[1 + e^{-\frac{\pi c}{2m\omega}}\cos\frac{p\pi}{\omega} - \frac{c}{2mp}e^{-\frac{\pi c}{2m\omega}}\sin\frac{p\pi}{\omega}\right]$$

$$R = -(x_0 - x_f)e^{-\frac{\pi c}{2m\omega}}\sin\frac{p\pi}{\omega}\left[\frac{c^2}{4m^2p} + p\right]$$

From [28] we deduce:

$$\sin\epsilon = \frac{CP - AR}{AQ - BP} \quad \text{and} \quad \cos\epsilon = \frac{BR - CQ}{AQ - BP}$$

Substituting the values just given in these equations,

$$\sin\epsilon = -qH\frac{x_f}{a}\cos\epsilon = q\left[\frac{x_0}{a} + G\frac{x_f}{a}\right] \ldots\ldots [29]$$

where G and H are functions independent of the Coulomb damping:

$$G = \frac{\sinh(\beta\pi c/c_c) - \sqrt{\frac{c/c_c}{1-(c/c_c)^2}} \cdot \sin\beta\pi \sqrt{1-(c/c_c)^2}}{\cosh(\beta\pi c/c_c) + \cos\beta\pi \sqrt{1-(c/c_c)^2}}$$

$$H = \frac{\beta}{\sqrt{1-(c/c_c)^2}} \cdot \frac{\sin\beta\pi \sqrt{1-(c/c_c)^2}}{\cosh(\beta\pi c/c_c) + \cos\beta\pi \sqrt{1-(c/c_c)^2}} \quad [30]$$

Eliminating ϵ from the equations of [29]:

$$\frac{x_0}{a} = -G\left(\frac{x_f}{a}\right) + \sqrt{\frac{1}{q^2} - H^2\left(\frac{x_f}{a}\right)^2} \quad [31]$$

which is an expression for the amplitude and will reduce to the response function for viscous damping $1/q$ when $x_f/a = F/P = 0$. The phase angle φ can be found in each case from [29] and [26]. This gives a complete solution for the non-stop motion, the results of which are plotted above the broken lines in Figs. 7 to 12.

Assuming a motion with one stop per half-cycle [Fig. 3] the solution [25] holds for the interval of motion $0 < t < t_0$. For the determination of C_1, C_2, x_0, φ, and t_0, the five conditions of [13] and [14] are available. Elimination of C_1 and C_2 gives the result [27].

By means of [14] and the last two conditions of [13] we solve x_0, φ, and t_0 from [27] exactly as was done in Appendix No. 1. The rather complicated result can be written as:

$$\begin{aligned} 2\frac{x_f}{a} = {} & e^{-\frac{c}{c_c}\beta\omega t_0} \cos pt_0 \left[-\cos\varphi + \frac{\cos\varphi}{Vq^2} + \frac{2}{\beta}\frac{c}{c_c}\cdot\frac{1}{q^2}\sin\varphi \right] \\ & + e^{-\frac{c}{c_c}\beta\omega t_0} \sin pt_0 \left[-\frac{c}{c_c}\frac{\omega_n}{p}\cos\varphi + \frac{c}{c_c}\frac{\omega_n}{p}\cdot\frac{1}{Vq^2}\cos\varphi \right. \\ & + \frac{2}{\beta}\left(\frac{c}{c_c}\right)^2 \frac{\omega_n}{p}\cdot\frac{1}{q^2}\cdot\sin\varphi - \frac{\omega_n}{p}\cdot\frac{1}{\beta}\frac{1}{q^2V}\cdot\sin\varphi \\ & \left. + \frac{\omega_n}{p}\frac{2}{\beta^2}\cdot\frac{c}{c_c}\cdot\frac{1}{q^2}\cos\varphi \right] + \left[-\cos\varphi - \left(\frac{\cos\varphi}{Vq^2} \right.\right. \\ & \left.\left. + \frac{2}{\beta}\frac{c}{c_c}\frac{1}{q^2}\sin\varphi \right) \cos\omega t_0 + \left(\frac{\sin\varphi}{q^2V} - \frac{2}{\beta}\frac{c}{c_c}\cdot\frac{\cos\varphi}{q^2} \right) \sin\omega t_0 \right] \end{aligned} \quad [32]$$

in which the angle φ is determined by:

$$\sin\varphi = \frac{A}{\sqrt{A^2+B^2}}; \quad \cos\varphi = \frac{B}{\sqrt{A^2+B^2}} \quad [33]$$

where

$$\begin{aligned} A = {} & \frac{1}{Vq^2} e^{\frac{c}{c_c}\beta\omega t_0} \sin\omega t_0 - \frac{1}{q_2}\cdot\frac{2}{\beta}\cdot\frac{c}{c_c} e^{\frac{c}{c_c}\beta\omega t_0} \cos\omega t_0 + \beta\frac{\omega_n}{p}\sin pt_0 \\ & - \beta\frac{\omega_n}{p}\frac{1}{Vq^2}\sin pt_0 - \frac{2}{\beta}\left(\frac{c}{c_c}\right)^2\frac{\omega_n}{p}\cdot\frac{1}{q^2}\sin pt_0 + \frac{2}{\beta}\frac{c}{c_c}\cdot\frac{1}{q^2}\cos pt_0 \end{aligned}$$

$$\begin{aligned} B = {} & -\frac{2}{\beta}\frac{c}{c_c}\frac{1}{q^2} e^{\frac{c}{c_c}\beta\omega t_0} \sin\omega t_0 - \frac{1}{q^2V} e^{\frac{c}{c_c}\beta\omega t_0} \cos\omega t_0 \\ & + \frac{\omega_n}{p}\frac{2}{q^2}\frac{c}{c_c}\sin pt_0 - \frac{c}{c_c}\frac{\omega_n}{p}\frac{1}{q^2V}\sin pt_0 + \frac{1}{q^2V}\cos pt_0 \end{aligned}$$

For any chosen value of ωt_0 the corresponding values of x_f/a and x_0/a are calculated from [32] and [14], and diagrams like those of the Figs. 19 and 21 are drawn. From these the curves in Figs. 7 to 12 below the broken line are constructed.

The broken lines in these several diagrams indicate the boundary between the motions without stop and with one stop per half-cycle. They can be calculated either as the lower limit of the non-stop motion or as the upper limit of the one-stop motion.

For the non-stop motion, the condition of validity is again: $\dot{x} < 0$ for $0 < t < \pi/\omega$, which can be written as:

$$\frac{x_0}{x_f} \geqslant \frac{e^{-\frac{ct}{2m}}}{\sin\omega t} \cdot \left\{ \left(\frac{p}{\omega} + \frac{c^2}{4m^2p\omega} \right)(1+G)\sin pt + H\left(\frac{c}{2pm}\sin pt - \cos pt \right) \right\} + H\cos\omega t - G \quad \text{for} \quad 0 \leqslant t \leqslant \pi/\omega. \quad [34]$$

It is found that for $\beta < 2$ the maximum value of this expression occurs at $t = 0$, so that it simplifies to:

$$\frac{x_0}{x_f} \geqslant -G + 2H\beta c/c_c + \beta^2(1+G) \quad [35]$$

For values of $\beta > 2$ the maximum of [34] may occur at a different time than $t = 0$. In that case let S_1 be the ratio of this maximum to the value of [34] at $t = 0$. In most cases, including all those of the Figs. 7 to 12, $S_1 = 1$. Then the condition [34] in combination with [29] can be written as:

$$\frac{x_0}{a} \geqslant \frac{S_1(I-G)}{q\sqrt{H^2 + \{S_1I + (1-S_1)G\}^2}} \quad [36]$$

$$\frac{x_f}{a} \geqslant \frac{1}{q\sqrt{H^2 + \{S_1I - (1-S_1)G\}^2}} \quad [37]$$

where

$$I = 2H\beta c/c_c + \beta^2(1+G)$$

With [36], in which $S_1 = 1$, the broken lines in Figs. 7 to 12 were calculated.

For the one-stop motion, the two conditions [22] and [23] hold. The results [35], [36], and [37] can also be obtained from [32] by substitution of $\omega t_0 = \pi$.

APM-53-10

Strength of Materials Subjected to Shear at High Temperatures[1]

By FRANKLIN L. EVERETT,[2] ANN ARBOR, MICH.

This paper treats of an experimental program which is basically fundamental in the study of creep. Uniformly pure shear is produced in a thin tubular specimen submitted to twist, whereas the usual method employs a solid bar in tension. This torsion form of test permits of greater accuracy in the measurement of creep and provides the simplest stress condition, besides offering other decided advantages.

Since this work in twist represents virtually a pioneer attack, it has been necessary to develop practically from the beginning each phase of the investigation. The type of specimen, apparatus for the creep tests, including the measuring instruments, and experimental technique are described.

The experiments have been of three kinds: (1) Room-temperature tests were first made to determine the usual physical properties in torsion and in tension; (2) at various temperatures short-duration tests were made to learn the effect of different rates of loading, the modulus of elasticity in shear, and the ultimate strength in shear; (3) at 400 deg. cent. and 500 deg. cent. several creep tests in shear of relatively long duration were conducted. An accelerated method of making creep tests has been devised. In addition several special problems were investigated such as a study of the increase of the yield point by creep and the restoration of the original yield point by annealing, the effect of vibrations which included a continuous record of creep, and the effect of temperature fluctuations.

INTRODUCTION

1 Statement of the Problem



THE outward manifestations of creep are now generally recognized. In cases of commercial importance, metal which is subjected to the simultaneous action of stress and high temperature undergoes gradual distortion. A large number of investigators of creep and others interested in the problem, particularly in the United States and Europe, have contributed considerable information on the practical significance of creep and data applicable for purposes of design of equipment which operates under conditions where creep obtains. So satisfactorily has this part of the subject of creep been covered that it appears neither necessary nor possible to devote more than an occasional sentence to a review.

It is felt that the broad subject of creep deserves more fundamental study. The manner in which creep occurs is too interesting a question to remain unexplained. Very low rates of creep are required to furnish a more complete picture of the problem. The immediate investigation is believed to approach more closely than is possible by tension methods a situation which permits this fundamental research.

A large amount of work has been done to determine creep characteristics of metals in tension. In the present study the attack is different—creep in uniformly pure shear is produced by using a tubular specimen in twist.

In this study there have been developed the machines, type of specimen, and the measuring instruments which give more refined readings than are known to have been obtained in any other laboratory. The experimental technique has also been developed.

Some results have been obtained for giving the characteristics of medium-carbon steel at three temperatures, namely, 400, 450, and 500 deg. cent., and in addition some work has been accomplished at ordinary temperatures. Only one material has been used in the tests, since it was desired to prepare as complete information for comparative purposes as was possible.

The experimental work has been of three kinds: (1) A number of tests have been performed at room temperature to learn the usual physical properties of the material; (2) a series of short-duration tests at high temperature were made in order to establish the more important physical properties under such conditions and to provide some preliminary tests which might be useful in predicting the behavior of the material in creep; and (3) a few long-duration creep tests were run to study the material when it was exposed to service conditions of high temperature and stress over some time.

Also several special problems have received attention which deal with practical considerations relating to the matter of creep.

2 Torsion Vs. Tension

Torsion methods of studying the problem of creep present certain advantages over those in tension. It is due to these and the basic value of creep results in general that the present project has been undertaken and developed.

As a result of temperature fluctuations of the specimen during a test, the effect upon strain measurements is much more serious in the case of tension than in torsion. Due to a thermal change of 1 deg. cent. on the length alone, the effect upon strain readings in tension tests is 1.2×10^{-5} inches per inch of length, whereas

[1] From a dissertation presented for the degree of Doctor of Philosophy in the University of Michigan. The experimental program was conducted in the Applied Mechanics Laboratory under the supervision of Prof. S. Timoshenko.

Presented at the National Applied Mechanics Meeting, Purdue University, Lafayette, Ind., June 15 and 16, 1931, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] Instructor in Engineering Mechanics and Research Associate, Department of Engineering Research, University of Michigan. Jun. A.S.M.E. Dr. Everett was graduated in mechanical engineering from the University in 1925, with the degree of B.S.E. in M.E. During the following three and one-half years he was connected with the Standard Oil Co. (New Jersey), first as a member of the student-engineering course and later as an inspector of refinery high-pressure equipment in the general engineering department. He returned in February, 1929, to the University of Michigan to continue graduate work, and received the degree of M.S.E. in M.E. in June of the same year. Since that time he has been connected with the department of engineering research at the University. He obtained the degree of Ph.D. in June, 1931.

in torsion the geometric changes due to temperature are of only very slight consequence. Consideration of all effects upon measurements in torsion is given in a discussion in Section V, Article 3. This situation immediately presents a distinct advantage of torsion methods over tension.

Means of obtaining very fine measurements of strain in torsion are simpler than in tension. It is possible to combine mechanical and optical levers favorably so as to detect very refined values of strains due to torsion.

In torsion a moment of constant magnitude is easily designed to act automatically on the specimen throughout the experiment. The usual tension creep apparatus requires that the load beam be readjusted to maintain a uniform force on the specimen during elongation.

The maximum-shear theory is usually selected as the basis of design of machine parts, so that information concerning the shearing stress of ductile material such as steel is of practical importance. It is generally accepted that deformation of ductile material, such as steel, takes place due to gliding resulting from shearing stresses, the maximum gliding occurring on the plane of maximum shear.

Experiments in uniformly pure shear provide the simplest stress condition and therefore allow the most favorable situation for making a study of such a problem as is offered by creep. Tension methods produce a more complicated stress condition on the plane of maximum shear, since normal tensile forces are also present.

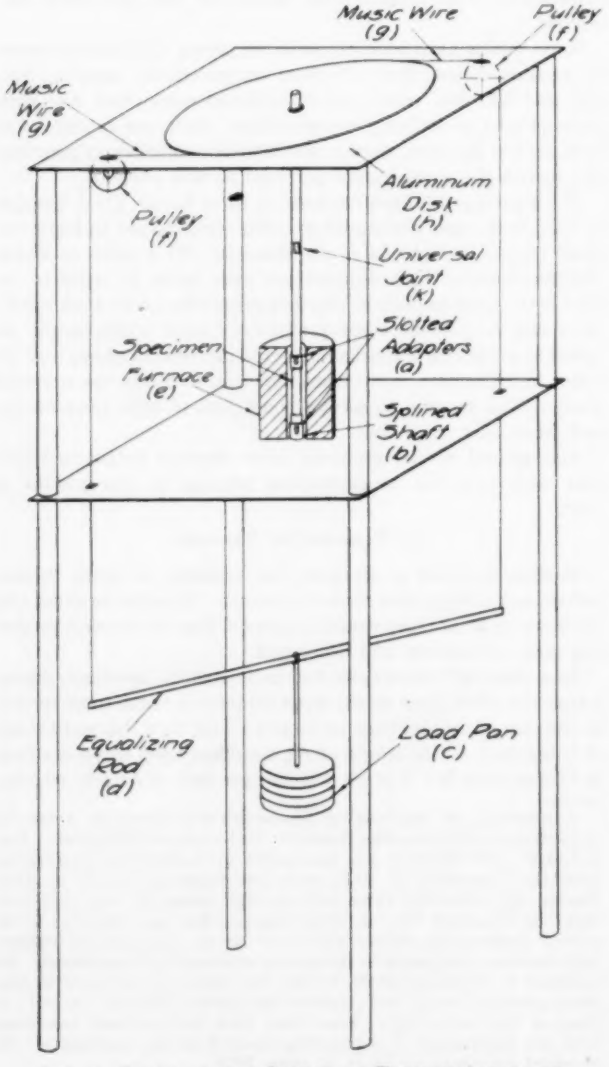


FIG. 1 DIAGRAMMATIC SKETCH OF TORSION APPARATUS

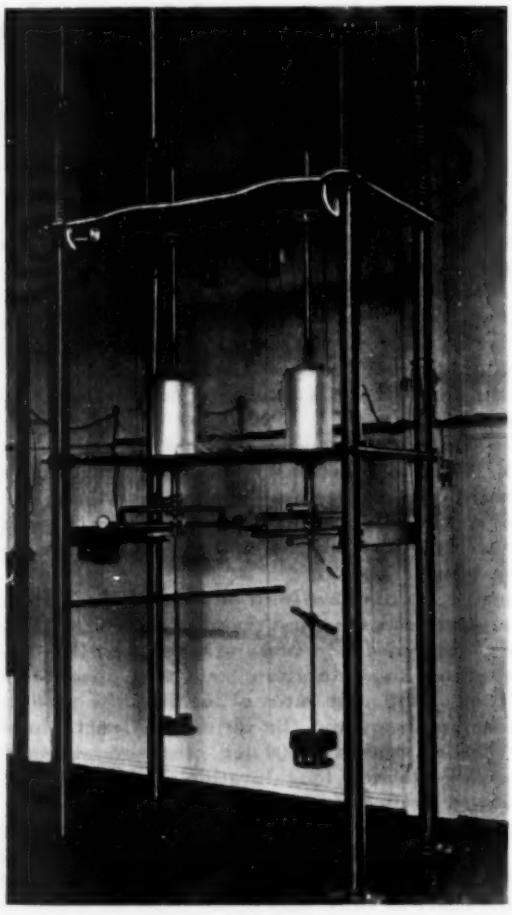


FIG. 2 NOS. 2 AND 3 TORSION MACHINES

I—APPARATUS AND SPECIMENS

1 TORSION MACHINES

In the design of the apparatus for the torsion investigation, a number of requirements were imposed. The specimen was to be tubular, held rigidly at one end, and twisted free from bending or longitudinal pressure at the other end with a known moment which remained constant during angular deformation of the specimen. Friction was to be held to a minimum. The specimen was to be contained in a furnace and maintained at constant temperature. Some provision had to be made for accurately measuring shear strain.

A diagrammatic sketch of the essential features of the torsion machine is given in Fig. 1. Two of the new machines built integral in the same frame are shown in Fig. 2. Discussion will be directed toward the design of the improved apparatus of the last two units. The fact that the torsion specimen was tubular established the form of the slotted adapters (*a*), Fig. 1. The end which resisted rotation was made a splined shaft (*b*) to allow for axial adjustment in inserting and removing the specimen. The other end was connected to a large, lightweight aluminum disk (*h*) on the circumference of which two fine music wires (*g*) produced a couple. These wires passed over pulleys (*f*) of relatively large diameters and were fastened

to both ends of an equalizing rod (*d*), to the center of which a platform (*c*) was attached for applying loads. The large disk and pulleys were mounted on ball bearings to reduce the friction. Between the rigid end and the aluminum disk which furnished the moment, a universal joint (*k*) was supplied in the shaft to eliminate the possibility of bending action upon the specimen.

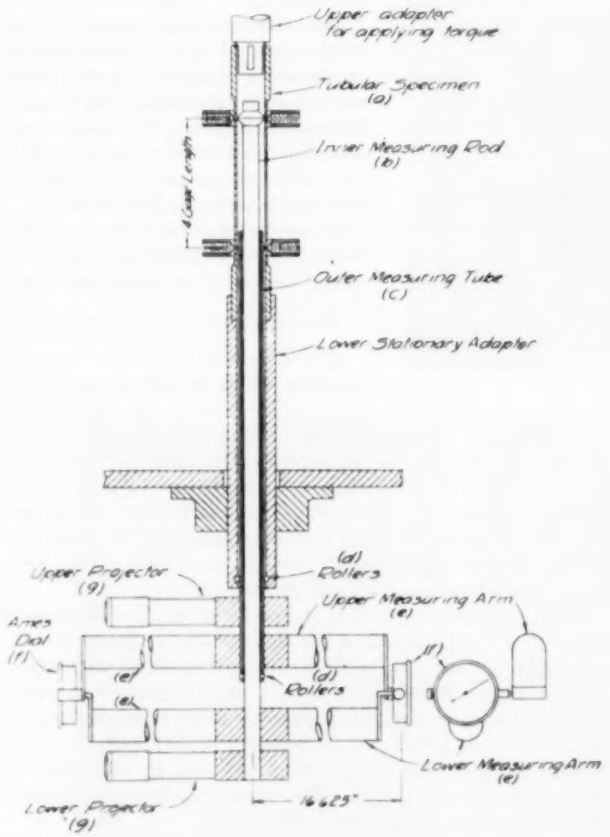


FIG. 3 SHEAR-STRAIN MEASURING DEVICES

Some play between the specimen and the adapters allowed axial thermal expansion without causing longitudinal pressures.

Since the furnace (*e*) surrounded the specimen, part of the shear-strain-measuring apparatus had necessarily to be within the heated zone. It was decided that it was best to select a vertical design of machine in order to permit these measuring devices to hang downward and thereby avoid bending and general distortion due to the heat. Detailed descriptions of the instruments and furnaces are given in the following articles.

A salt or lead bath to protect the specimen from excessive oxidation could be used very easily with this apparatus. The design of the machine is favorable to this arrangement.

2 STRAIN-MEASURING INSTRUMENTS

Exceeding simplicity and accuracy have been embodied in the principle and design of the devices for measuring angular deformations of the specimen. It was necessary to devise means of measuring the relative angular motion of two cross-sections of the specimen. Several methods have been developed, and during various stages of the same test, each, depending upon the rate of movement and the degree of sensitivity demanded, may be employed. The systems are mechanical, optical, or combinations of the two. The different methods that have been developed number four and will be discussed in the order of the accuracy which they are capable of obtaining.

The gage length which has been arbitrarily selected as 4 in. is constant in all experiments where strain is measured. A concentric rod and tube, accordingly, are clamped by suitable means inside the specimen. A sectional drawing which illustrates the method of securing the rod and tube at the gage points and the attachment of the strain-measuring instruments is given in Fig. 3. The relative angular movement between this rod and tube describes the shear strain of the 4-in. portion of the specimen (*a*) which is under investigation. To the rod (*b*) and tube (*c*) one or more of the devices (*f*, *g*) for measuring the strain are attached. Rollers (*d*) are provided to steady the rod and tube at the lower ends.

The simplest method of measuring shear strain which is employed throughout the entire test consists of a pair of projectors (*g*), one each attached to the rod and tube, equipped to cast a sharp vertical shadow on a curved graduated scale. The projectors may be readily seen in Figs. 4 and 5. This optical lever is used in connection with either of two scales of 96-in. and 174-in. radius, respectively. The smallest reading that can be observed on the scales is 0.01 in., which corresponds to angles of 1.04×10^{-4} and 5.75×10^{-5} radians, respectively. The shear strain which can be measured is then 1.29×10^{-5} and 7.09×10^{-6} in order. Although this method constitutes the least refined means of measuring the strain, the minimum rate of creep, for example, over a 12-hour period, may be read to 1.07×10^{-6} for the 96-in. and 5.91×10^{-7} for the 174-in.-radius scale, strain

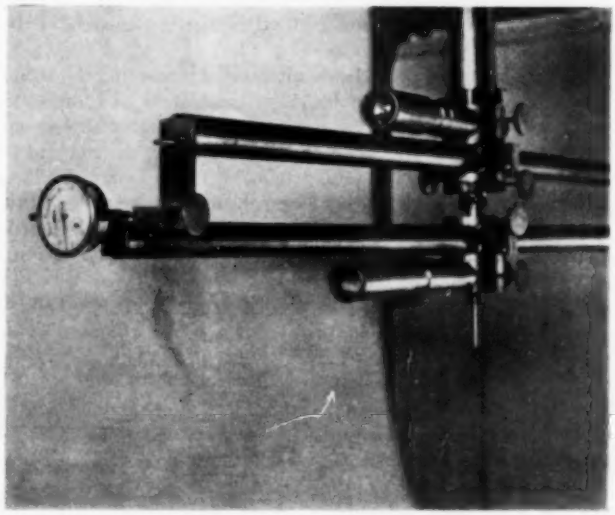


FIG. 4 AMES DIALS AND PROJECTOR ATTACHMENTS

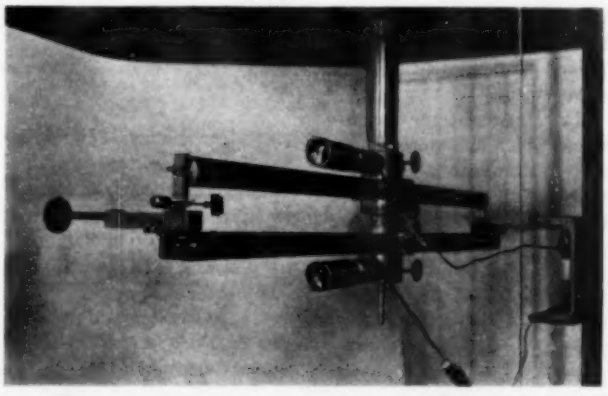
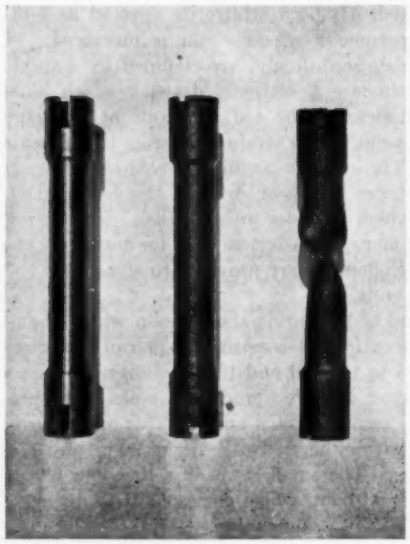


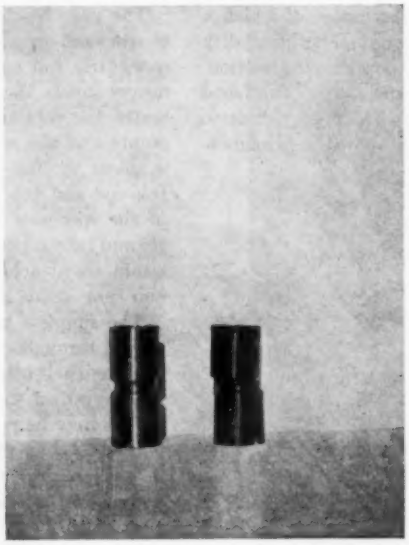
FIG. 5 MARTENS MIRRORS AND PROJECTOR ATTACHMENTS



(a) (b) (c)

FIG. 6 TUBULAR SPECIMENS FOR ELASTIC AND CREEP TESTS IN TORSION

[(a) Unstressed tube. (b) After creep test. (c) After buckling failure.]



(a) (b)

FIG. 7 SHORT-LENGTH SPECIMENS FOR ULTIMATE SHEARING STRENGTH

[(a) Unstressed tube. (b) After shear failure.]

per hour. When rates are so low that more refined devices must be resorted to, this projector method furnishes a check and the amount of the total deflection.

The other three more delicate methods of measuring the strain take advantage of a mechanical-lever mechanism in addition to the sensitivity provided by the various strain instruments themselves. Arms (*e*) in Fig. 3 are fastened rigidly to the rod and to the tube. Various types of instruments (*f*) are mounted which measure the relative linear displacement of the outer ends. It is assumed, at the comparatively large radius at which the instruments are located, that for small angular movements the arc and chord are equal. Linear-strain-measuring devices serve to indicate these movements.

In the first of these cases Ames dials (*f*), which will permit of a minimum reading of $^1/_{10,000}$ of an inch, are employed. Rates of creep with readings at 12-hour intervals are accurately read to some 6.19×10^{-8} strain per hour. This method gives very satisfactory results for rates of creep within this range. Fig. 4 is a photograph which shows the Ames dials being employed for measuring the shear strain.

The Huggenberger tensometers, which are more widely used in European laboratories than in this country, permit the measurement of linear displacements of the order of $^1/_{100,000}$ of an inch. These instruments consist essentially of a compound set of mechanical levers. The Huggenberger instruments are attached to the end of the arms (*e*, Fig. 3) in the same manner as the Ames dials. It is seen that the accuracy obtained is about ten times as great as with the Ames dials. The minimum rate of creep is of the order of 5.15×10^{-9} strain per hour when the readings are made at 12-hour intervals.

The Martens-type mirror instruments, seen in Fig. 5, provide a method of obtaining a slightly greater sensitivity than is furnished by the Huggenberger tensometers. The rate of creep which can be detected is about 5×10^{-9} shear strain per hour, based on the same interval previously used. This method furnishes the most sensitive means which has been developed at the laboratory for measuring creep.

3 DEVELOPMENT OF SPECIMEN DESIGN

It was desired to find a shape of specimen which would approach a condition of uniformly pure shear. Obviously a solid shaft in twist presents an unfavorable situation. The phenomenon is complicated due to the fact that the stresses are not uniform. A thin tubular section was therefore selected, for which the assumption can be made that stresses are evenly distributed over the cross-section, and their magnitude is based on the mean radius.

Numerous tests were made in torsion of such tubes at room temperature. The length of the thin section for determinations of angular deformation necessarily had to be of appreciable amount. Experience showed that it was not possible to obtain the ultimate shearing strength, since sudden buckling occurred and failure took place due to tension. Photographs of an unstressed tube, one following a creep test and one after buckling failure, are given in Fig. 6. This design was certainly adapted only to a stress somewhat below ultimate.

Since buckling was not observable in the least until the final

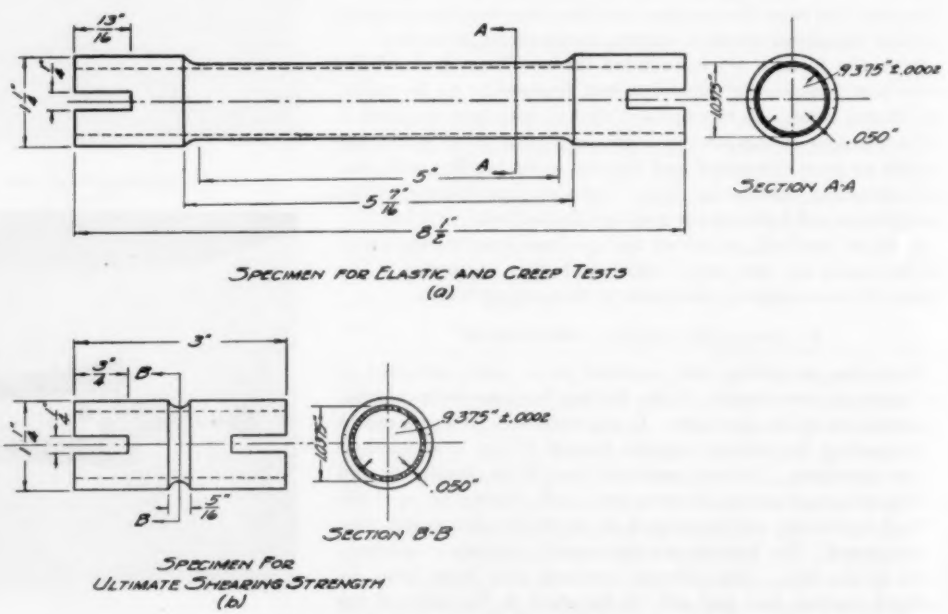


FIG. 8 TUBULAR SPECIMENS

stage of the test, observations of strain during the first part of the test, which was the important part for creep measurements, were not affected.

In order to get data for ultimate strength, the final design adopted was merely a rounded groove with a long radius at the bottom, a photograph of which is shown in Fig. 7 both before and after failure.

The two standard type specimens, then, which were employed throughout the torsion tests reported here, were of the design described in the above-mentioned development. The specimen which was used for the investigation of all elastic and creep-test determinations is shown in Fig. 8(*a*), while the design adopted for the few tests where ultimate strength was required is given in Fig. 8(*b*).

Later tests have indicated conclusively that no measurable buckling took place when the specimen chosen for creep tests was subjected in the usual experiments to shear stress over long periods of time.

II—MATERIAL

1 Rough Stock

As much care as possible was exercised in the selection of material for the program of experiments. It was desired to have a steel which exhibited usual physical properties and which at the same time was homogeneous. Since the finished specimens were to assume the form of a thin hollow cylinder, tubular stock was preferred. A medium-carbon seamless steel mechanical tubing was chosen.

This steel has properties which are well known at ordinary temperatures. The material was obtained in one lot sufficient for the entire investigation. The manufacturer's chemical analysis furnished the following:

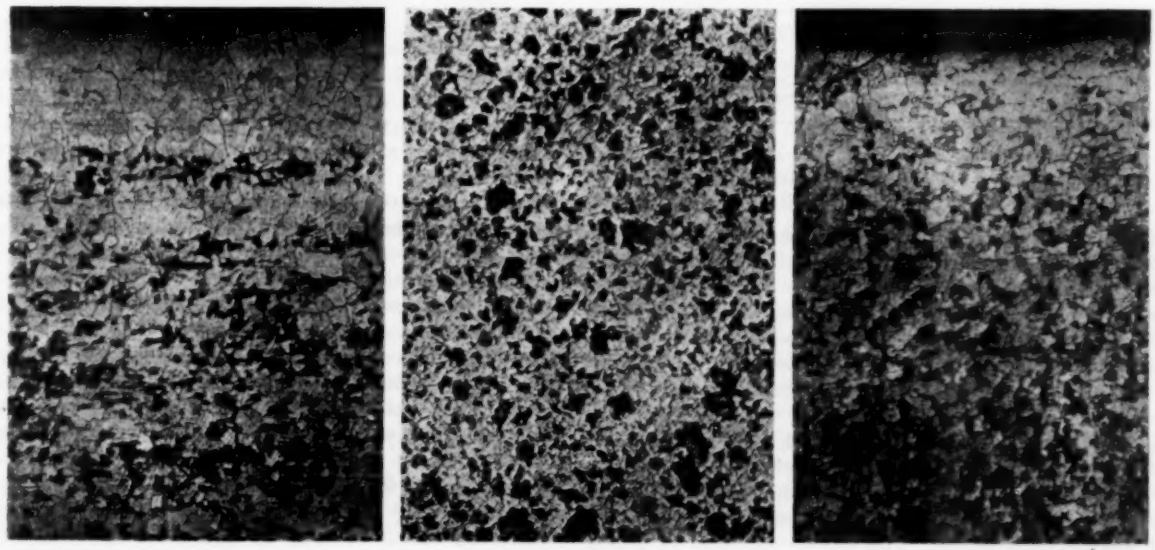
	Per cent
Carbon	0.34
Manganese	0.80
Phosphorus	0.02
Sulphur	0.03
Silicon	0.10

An analysis in the University laboratory showed the carbon content of the tubes to be 0.347 per cent, with a maximum variation of 0.001 between samples.

Upon receipt in the laboratory the tubes were cut to the length desired for the specimens and annealed in an electric furnace in an atmosphere of air. The annealing temperature in all cases was 900 deg. cent., which was maintained for a period of from fifteen to thirty minutes. The tubes were permitted to cool slowly in the furnace.

2 Preliminary Tests at Room Temperature

It was desired to investigate the physical properties of the material in tension and in torsion at ordinary temperatures by the usual short-time methods. For the tension tests two types of specimens were used. Small rectangular strips were cut from the walls of both unannealed and annealed tubular stock and subjected to tension. From these tests it was decided that only annealed material would be used in subsequent experiments, since uniformity in results and a sharp yield point could be obtained only when the material was in the annealed condition. Tubular specimens, as the second type, were tested in tension also. The tube stock was turned down into the same essential form which was adopted for the torsion experiments, as seen in Fig. 8(*a*).



(*a*) (*b*) (*c*)

Fig. 9 Photomicrographs of Material

[(*a*) Highly decarburized inner surface. (*b*) Homogeneous interior structure. (*c*) Slightly decarburized outer surface.]

TABLE 1 TENSION TESTS AT ROOM TEMPERATURE

Type of specimen	Condition	Mark No.	Prop. limit, lb. per sq. in.	Yield point, lb. per sq. in.	Tensile strength, lb. per sq. in.	Elongation, per cent	Red. of area, per cent	Modulus $E \times 10^6$, lb. per sq. in.
Strip	Unannealed	1	60,000	...	91,200	13	45	30.5
Strip	Annealed	3	40,700	40,700	72,300	36	48	28.9
Tube	Annealed	4	34,350	34,350	67,600	35.5	..	29.3

The magnitudes of proportional limit, yield point, ultimate tensile strength, elongation, reduction of area, and modulus were obtained for each specimen. In the case of the tubular form, all of these values were consistently lower than for the strip type. This was later attributed to the fact that there was a thin layer of decarburized material at the inner surface of the original tube stock which formed a larger proportion of the cross-section in the thin tubular specimen than in the thicker strip. Representative tension tests are reported in Table 1.

It was further decided to subject the material to metallographic examination. Sections at right angles to the axis of both

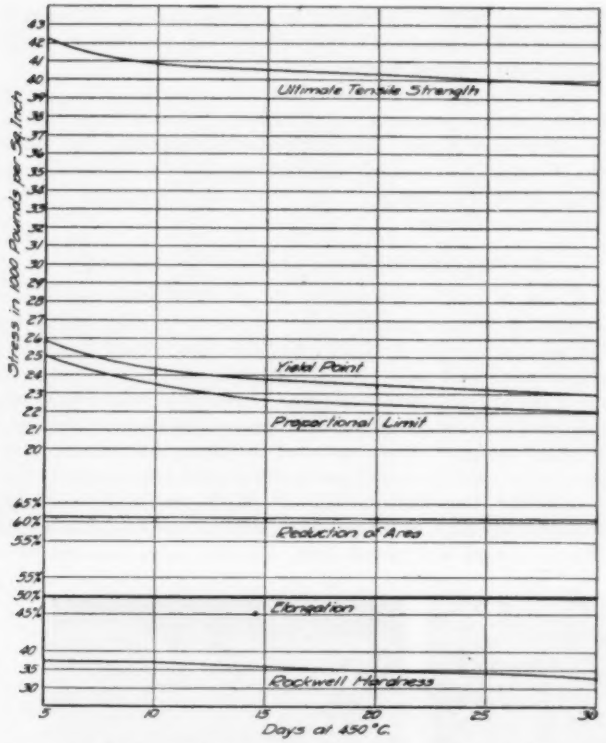


FIG. 10 MECHANICAL TESTS OF THIN STRIPS AT ROOM TEMPERATURE AFTER VARIOUS LENGTHS OF HEATING AT 450 DEG. CENT.

the strips and tubes were polished and photomicrographs prepared. As can be seen by referring to Fig. 9, the material throughout the cross-section is not homogeneous. Fig. 9(*a*) shows the structure of more than half of the thickness of the tube specimen. A large proportion is decarburized and consequently weaker than the interior material. A photomicrograph of a section of the strip showing the center is given in Fig. 9(*b*). This is the normal structure. Another sample was taken from the thick end at the grip of a tubular specimen. The outer surface, as may be seen in Fig. 9(*c*), is slightly decarburized.

From this study it was perceived that the tubular specimens must be redesigned so as to include only homogeneous material which was not decarburized. This was accomplished by merely increasing the internal diameter as well as the external diameter in order to maintain the same wall thickness of 0.050 in. The material of the final form of specimen, then, consisted wholly of the homogeneous structure shown in Fig. 9(*b*).

For the torsion tests at room temperature, annealed tubular specimens of two forms were used. To determine the elastic properties the original design described above was employed. In order to obtain the value of ultimate shearing strength a special type was devised of considerably shorter length and of reduced section. Details of this form of specimen and its development have been discussed in Section I, and reference in that connection may be made to Figs. 7 and 8(*b*).

Table 2 gives the characteristic data derived from the torsion tests at room temperature.

The yield point in torsion for this annealed material was well defined. In the case of the specimen with only $^1/_{32}$ in. length of reduced section it is probable that the dimensions of the wall were not accurately known. The value of ultimate shearing strength obtained from No. 71 specimen is considered reliable.

Bearing in mind that the reduced section of the tubes with the exception of No. 71 consisted to an appreciable extent of weaker decarburized material, we can relate the information of the tension and torsion tests. Comparing the values of yield point, the ratio obtained in uniformly pure shear to that in tension at ordinary temperatures assumes the value of approximately 0.6, which is in good agreement with the maximum-energy theory.

3 EFFECT OF LONG-TIME HEATING ON PHYSICAL PROPERTIES AT ROOM TEMPERATURE

It has been conjectured that steel subjected to the continued effect of high temperatures may undergo some structural change. This alone may produce differences in the physical properties, so that in creep tests it may be possible that the specimen becomes continually weaker—or perhaps stronger—under the single action of prolonged heating. Furthermore there is a possibility that long-time exposure to some temperatures at which creep tests are conducted results in decarbonization of the surfaces of the metal

These questions were approached by investigating strips of steel $^1/_{16}$ in. thick and 1 in. in width, of 0.068 ± 0.004C content. The strips were given an initial anneal at 900 deg. cent. and allowed to cool in the furnace. They were then heated for various lengths of time at 450 deg. cent. This temperature was arbitrarily selected. Two strips were heated for 5 days, two for 10 days, and so on through 30 days, so that twelve specimens were prepared. At the conclusion of the heating periods the strips were again allowed to cool slowly in the furnace.

After the heating, mechanical tests were made on the strips at room temperature to determine their hardness and tensile properties.

Although it is appreciated that the hardness of strips as thin as $^1/_{16}$ in. undoubtedly cannot be accurately determined by means of the method of the Rockwell machine,[3] the $^1/_{16}$-in.-diameter ball and 100-kg. weight were used to study the relative hardness of the strips. It was felt, however, that the consistency of the observed values for a given specimen and the reasonable qualitative agreement with the subsequent tension tests presented interesting information. It was found that prolonged heating resulted in a continual softening of the material. The curve for hardness is given in Fig. 10.

Each strip was tested in tension at ordinary temperatures. The general physical properties—proportional limit, yield point, ultimate tensile strength, breaking strength, reduction of area, and elongation—were determined. Again it is observed by further reference to Fig. 10 that the effect of heating was to weaken the steel, and that there is a material falling off in the magnitude of tensile strength, yield point, and proportional limit. The reduction of area and elongation appear to remain unchanged.

The results of this particular set of tests are offered as a qualitative indication only of what may obtain in situations where metal is subjected for long periods to high temperatures. More experimental evidence is needed to establish convincing arguments concerning the problem.

TABLE 2 TORSION TESTS AT ROOM TEMPERATURE

Type of specimen	Mark	Prop. limit, lb. per sq. in.	Yield point, lb. per sq. in.	Shearing strength, lb. per sq. in.	Modulus $G \times 10^6$, lb. per sq. in.
Long tube [Fig. 8(*a*)]	1	20,300	20,600	....	11.5
$^1/_{32}$-in. reduced section	31	....	....	57,300	....
$^1/_{4}$-in. reduced section	34	....	....	59,200	....
$^1/_{4}$-in. reduced section with fillets	35	....	....	63,600	....
Long tube with steel disks	29	....	....	57,800	....
Round groove [Fig. 8(*b*)]	71	....	....	65,870	11.5

[3] G. A. Hankins, "Hardness Tests of Steel Strip," *Engineering* (London), vol. 130, 1930, p. 324.

III—SHORT-DURATION TESTS AT HIGH TEMPERATURES

1 Problems Investigated

All of the experiments at high temperature in twist where the tubular specimen was subjected for the major part of the test to a varying load and uniform temperature are for convenience discussed here under "Short-Duration Tests." Some of these single tests required as much time as two or three weeks, while others took only a few minutes.

Three types of problems were investigated: (1) the effect of various rates of loading; (2) the determination of modulus of elasticity in shear; and (3) the value of ultimate strength in shear. These tests were made at temperatures ranging up to 500 deg. cent.

2 Effect of Various Rates of Loading

Prior to making the regular creep tests, a series of experiments at each of several selected temperatures was run to study the effect of the rate of application of the load and the behavior of the creep during the intervals between loading. The form of specimen is shown in Fig. 8(*a*). These tests were directly used to predict roughly what creep rates could be expected subsequently in the long-duration creep tests. It was felt that later a more accurate method of prediction could be discovered, based on these short-time tests.

The same general procedure of applying the load was followed in each of these experiments. A specimen was loaded with equal increments (always 10-lb. weights, which corresponded to about 1600 lb. per sq. in. shear stress, depending upon the exact measurements of the particular specimen) at definite intervals of time. These loading periods in each series were 1 minute, 1 hour, and 24 hours, and the tests repeated for 400, 450, and 500 deg. cent.

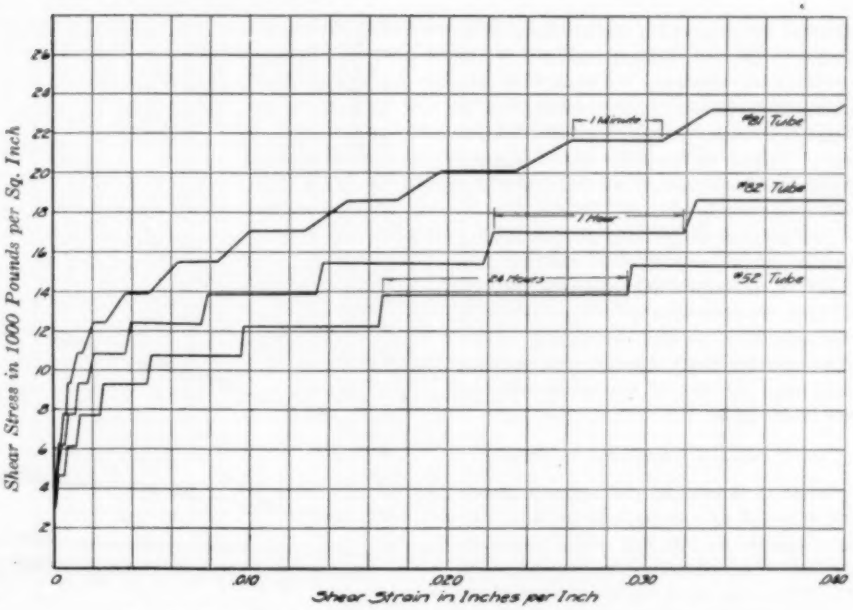


Fig. 12 Effect of Various Rates of Loading at 450 Deg. Cent.

The results are given in the curves of Figs. 11, 12, and 13.

During the interval between the addition of the loads in the case of the one-day period, several observations of strain were made, as frequently as each two to four hours in the early stages after each loading, and the rates of creep determined. Fig. 14 shows representative curves obtained in this manner at 500 deg. cent. It was an interesting discovery to find that the logarithm of the rate over the last 12 hours of each 24-hour period plotted against the corresponding stress which was acting, resulted substantially in a straight line. It was further significant in the case of the tests at 400 deg. cent. that this line paralleled the one which represented the minimum rate obtained in long-time creep tests. Such a comparison may be made by reference to the broken-line curves of Fig. 18 for 400 deg. cent. and Fig. 19 for 500 deg. cent. It is believed that some relationship may exist between the two curves so that predictions of minimum rate can be made on the basis of knowledge gained from such short-time tests and only a few regular creep tests.

The method of measuring angular deformations in all of these short-time tests was by the projectors and 96-in.-radius scale. The various devices for strain-measurement determinations have been described in Section I.

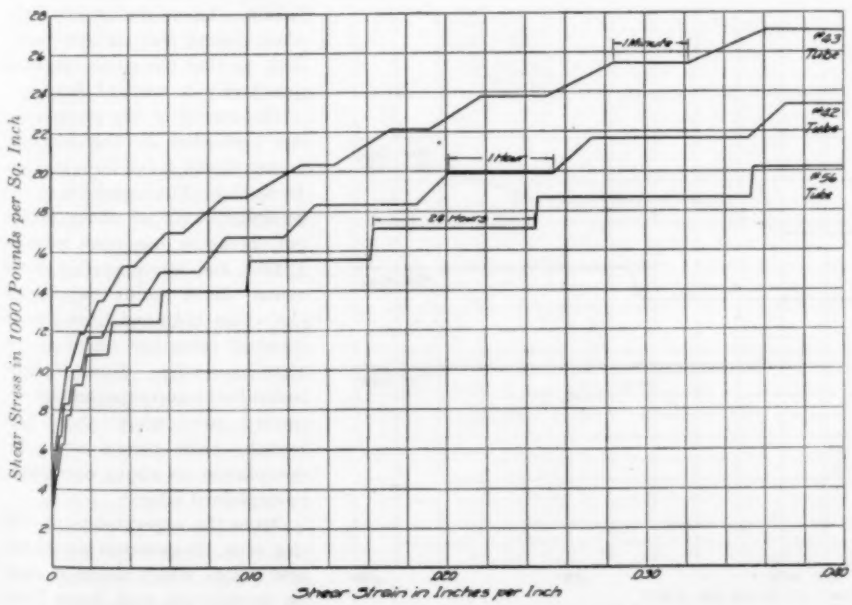


Fig. 11 Effect of Various Rates of Loading at 400 Deg. Cent.

3 Modulus of Elasticity in Shear

Modulus of elasticity in shear of the material at various temperatures from normal to 500 deg. cent. was determined, employing the specimen in Fig. 8(*a*). A curve is plotted from the resulting data in Fig. 15. This is in close agreement with studies in tension by McVetty and Mochel[4] on similar

[4] P. A. McVetty and N. L. Mochel, "The Tensile Properties of Stainless Iron and Other Alloys at Elevated Temperatures," *Trans. A.S.S.T.*, vol. II (1927), pp. 82, 100.

material by employing the relation: $G = E/2(1 + \mu)$. In this expression G is the modulus of elasticity in shear; E the modulus of elasticity in tension, and μ Poisson's ratio, which is about 0.3 for steel. It can be assumed that Poisson's ratio remains constant at this value for any temperature.

The manner of arriving at satisfactory values of the modulus of elasticity in shear at elevated temperatures was by making observations of shear strain of the specimen during unloading rather than upon loading. Angular deformations were read by use of the projectors and the 96-in. scale.

4 Ultimate Strength in Shear

Tests to determine the ultimate shearing strength at various temperatures from atmospheric to 500 deg. cent. were conducted. Short-length specimens shown in Fig. 8(*b*) were used, which precluded the possibility of buckling. The results are given in Fig. 16 in the form of a curve.

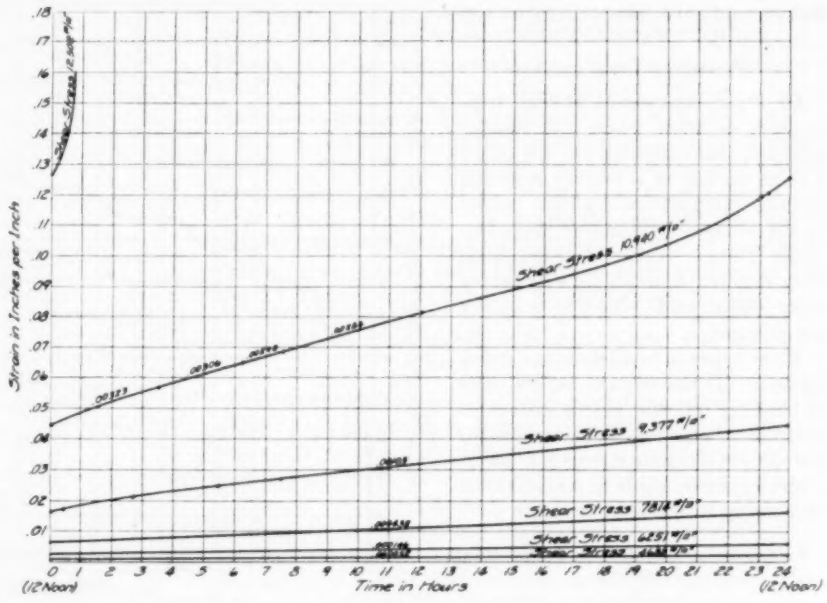


Fig. 14 Creep Rate Between 24-Hour Loadings at 500 Deg. Cent.

In loading the specimens to determine ultimate shearing strength, increments of 10 lb., and in the latter part of the test, 5 lb., were used. The rate of application of the loads was particularly important in these experiments, and a rate of 10 lb. (1600 lb. per sq. in.) per 30 sec. was carefully observed.

This curve does not look like those usually obtained for ultimate strength in tension at different temperatures. One of the reasons for the appearance of the curves in tension may be the complicated condition caused by necking. This situation is not present in experiments in uniformly pure shear produced by torsion. More experiments are required.

IV—LONG-DURATION TESTS AT HIGH TEMPERATURE

1 Results of Long-Duration Tests

Long-duration creep tests of tubular specimens of medium-carbon steel subjected to uniformly pure shear were made at temperatures of 400 and 500 deg. cent.

A number of the creep tests in shear were conducted in the usual manner, that is, the specimen was brought to the required temperature and the stress increased to a predetermined value, after which strain measurements were observed at certain time intervals. It was noted that the minimum rate of creep in each instance occurred at approximately the same total shear strain, and that buckling of the tube took place soon afterward. Strain, then, appeared in these cases to be the criterion of minimum rate of creep and also of buckling failure. Fig. 17 gives a family of time-deflection creep curves of the usual type of test made at 400 deg. cent. It is clearly seen from the first four curves that the minimum rate of creep occurs in each case at about 0.10 shear strain. The sharp upward turn in the curves denotes the initial stage of buckling failure. Large deflections take place during this sudden buckling, so that the curve becomes practically a vertical line.

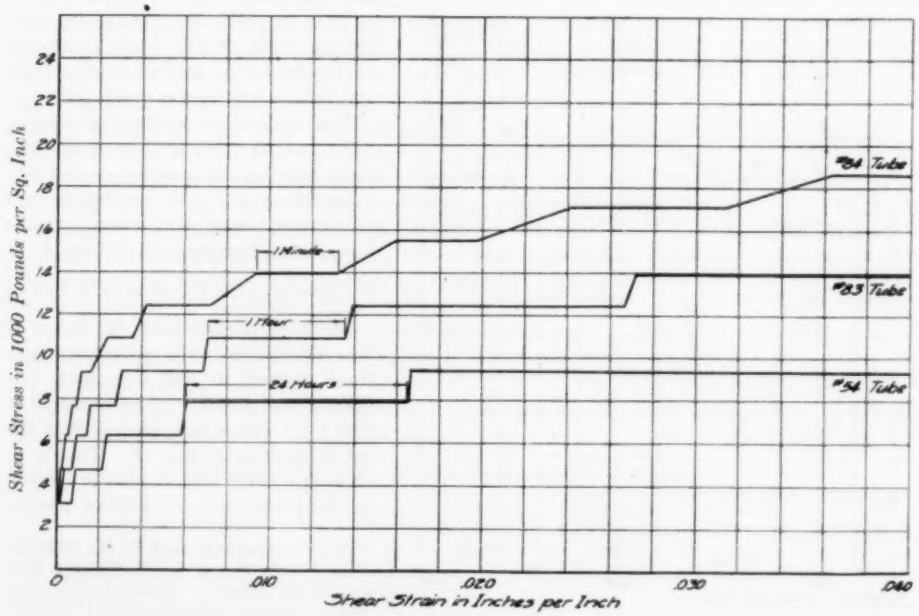


Fig. 13 Effect of Various Rates of Loading at 500 Deg. Cent.

The results of the creep tests are presented in the form of curves shown in full lines in Figs. 18 and 19. The logarithm of the minimum rate of shear strain per 24 hours has been plotted against the corresponding shear stress. Each point represents the value obtained from an individual creep test noted at the right-hand edge. Some of these individual creep experiments extended over more than one month, while others were accomplished in about one week, as explained later.

From the experiments at 400 deg. cent. the generous number of test points which describe such an unmistaken path leave little doubt as to the character of

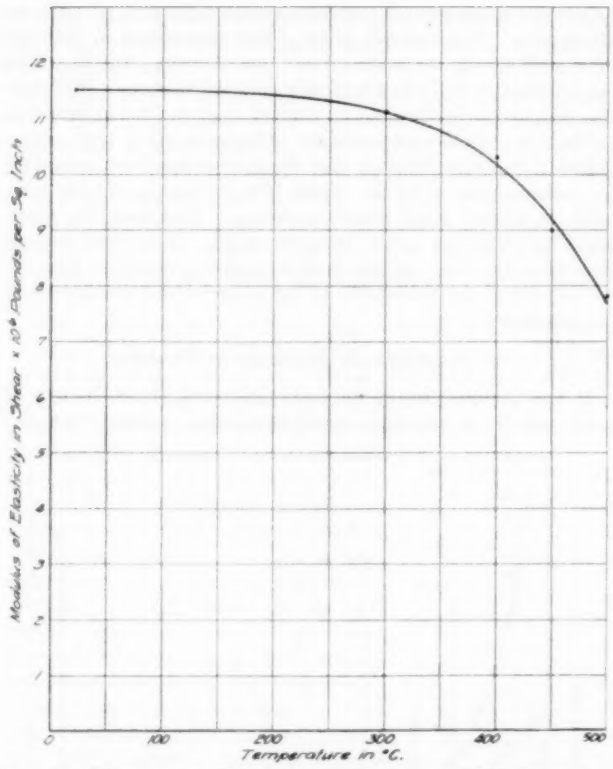


FIG. 15 MODULUS OF ELASTICITY IN SHEAR OF 0.34C STEEL AT VARIOUS TEMPERATURES

creep in shear of this material. The similar trend of the curve at 500 deg. cent. is established with reasonable assurance. Over a large portion of the curve for higher rates of creep the points fall in a straight line, but at lower rates there is a marked decrease in the creep rate for decreasing shear stress. In no creep tests has creep ceased. It is hazardous to predict beyond experimental evidence, but it appears that although there is a decided drop in the creep rate at the smaller stresses, delicate instruments which are furnished by this present method and further investigation of the problem may reveal the existence of creep very much beyond the range that has hitherto been supposed. The curve may even be asymptotic to the axis of the abscissa. Further experiments are in progress at present to establish more data for these low rates.

It may be reasoned that at the upper end of the curve very high rates of creep may be produced by large shear stresses. A physical maximum shear stress may well exist which causes instantaneous failure or, better, an infinite rate of creep. This stress is probably in the neighborhood of ultimate shearing strength, and so it seems the upper part of the curve becomes asymptotic to this practical limiting shear stress.

It was stated above that the values of the points which designated the shape of the curves of Figs. 18 and 19 represented the minimum rates of creep. From all of the tests made, a constant rate of creep has not been observed over any appreciable period of time, but rather a decreasing, or, in the case of approaching buckling failure, an increasing rate of creep. Fig. 20 is a typical curve which illustrates this point. These curves are obtained from any creep test by merely dividing the amount of shear strain which has taken place during an interval between readings by the length of the interval in hours, usually about twelve. There are three quite distinct parts of such a curve. The first portion, denoted by (*a*), is characterized by a rapidly falling rate. In some tests this occurs comparatively quickly. This part is followed by a region (*b*) of very much more uniform

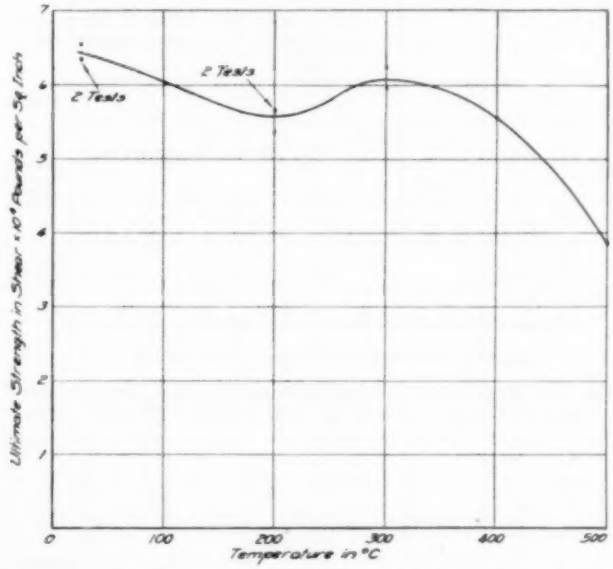


FIG. 16 ULTIMATE STRENGTH IN SHEAR OF 0.34C STEEL AT VARIOUS TEMPERATURES

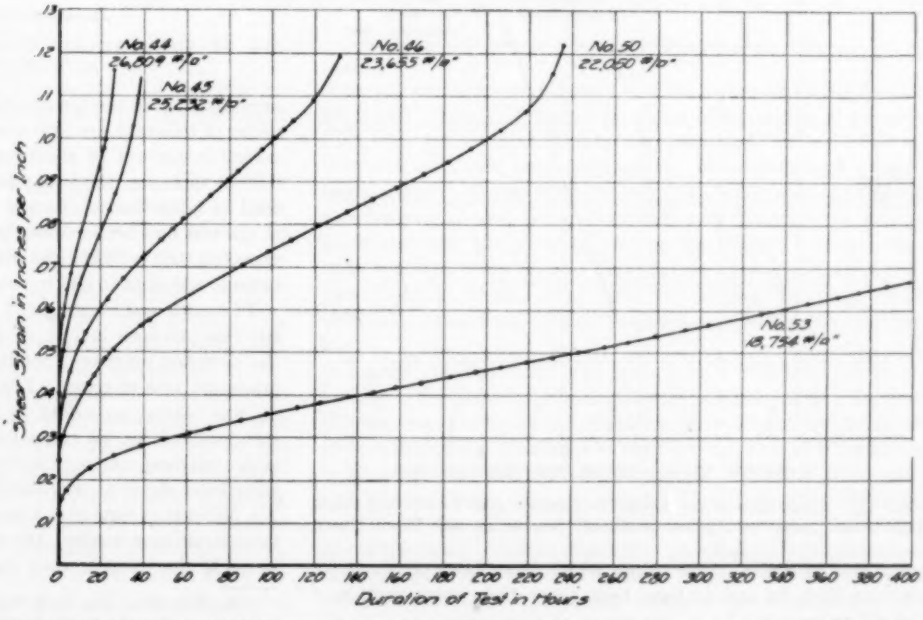


FIG. 17 SHEAR STRAIN VS. DURATION OF CREEP TEST FOR VARIOUS AMOUNTS OF SHEAR STRESS AT 400 DEG. CENT.

rate of creep, but all observations indicate it is not constant. The last stage (*c*) precedes buckling of the tube and shows a very sudden increase in rate. It can be conceived for practical purposes that the middle zone may be considered to be one of con-

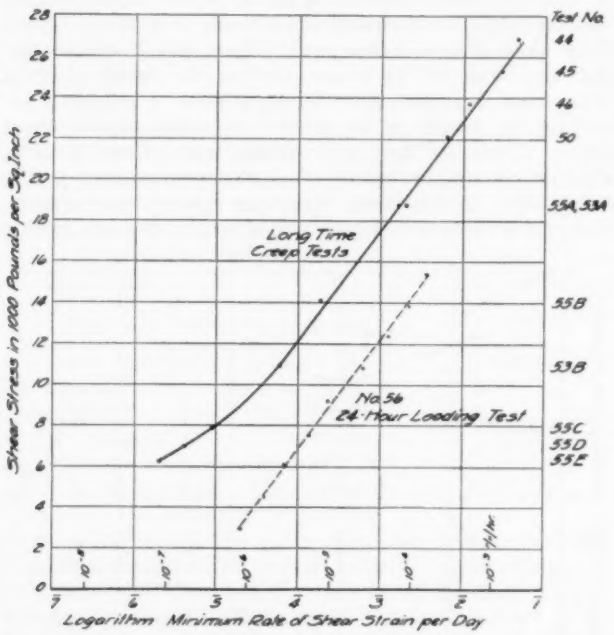


FIG. 18 COMPARISON OF SHORT-DURATION AND LONG-DURATION TESTS OF CREEP IN SHEAR OF 0.34C STEEL AT 400 DEG. CENT.

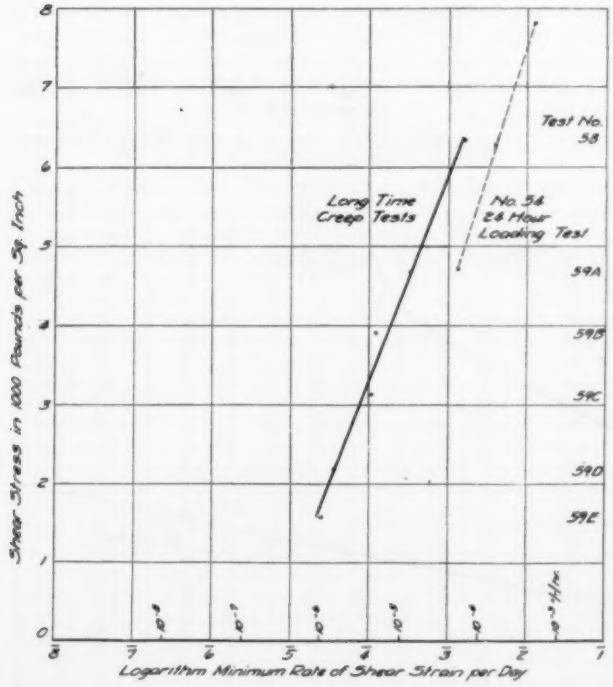


FIG. 19 COMPARISON OF SHORT-DURATION AND LONG-DURATION TESTS OF CREEP IN SHEAR OF 0.34C STEEL AT 500 DEG. CENT.

stant rate of creep. The values which have been obtained for plotting Figs. 18 and 19 have been taken from some portion of this (*b*) stage.

Mention was made earlier of the care which had been exercised to procure specimens of homogeneous material and of uniform dimensions. This was felt to be of first importance in entering upon such a program as the present one of creep. Fig. 21 shows the loading curves for five randomly chosen specimens. Although the manner of loading, as to rate of applying loads, was felt to be of no significance upon the subsequent creep test, nevertheless it was customary to load the tubular specimen according to a selected rate of 10 lb. (about 1600 lb. per sq. in.) per min. until the desired shear stress was acting. As is seen, the specimens for which the curves are given in Fig. 21 received various final loads, but the coincidence of the curves presents an interesting picture of the uniformity of the material and dimensions of the specimens.

2 ACCELERATED METHODS OF TESTING

It was postulated that the favorable point where the rate of strain was at a minimum could be reached quicker than was

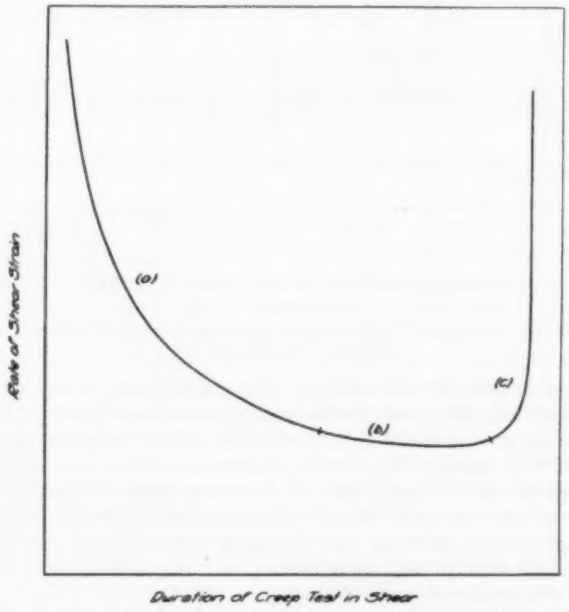


FIG. 20 CHARACTERISTIC CURVE OF RATE OF SHEAR STRAIN VS. DURATION OF CREEP TEST IN SHEAR

accomplished in the tests shown in Fig. 17, without affecting the value of this rate, by first overstressing the specimen. When a desirable amount of shearing strain had been obtained, about 0.08 in this case, the load would be reduced and creep would be read as a function of the new stress as though the entire history of the test had been according to common practice. It would be expected that a short time would be required for the specimen to become adjusted to the reduced stress.

This plan was investigated by selecting first a test (Tube No. 53, Test A[5]) seen in Fig. 18 that had been previously begun in the accepted manner of loading and which was approaching the minimum rate of creep. The new specimen (Tube No. 55, Test A) was loaded according to the same method which had been standardized for all creep tests, but the amount of the initial stress imposed was such as to produce in a few hours nearly the same total strain as was obtained with the other specimen (Tube No. 53) at that time over a period of about two weeks. When this total strain was reached, the excess load was removed so that the stress in the tubes in each experiment was the same. The rate

[5] Each tubular specimen was numbered, and when more than one test was made which employed the same specimen the experiments were designated by Test A, Test B, and so on.

of creep soon dropped in the accelerated test to values which approximated within experimental limits the rate of creep shown by the usual test.

So close was the agreement in the two tests that the points when plotted according to the logarithm of the minimum rate against shear stress nearly coincided. Fig. 18 may be referred to for comparison of the results seen as two adjacent test points. The procedure developed for this test was considered worth adopting for further tests. Two weeks only were saved in this case, but for lower values of stress, and consequently of strain, it is seen that a far greater saving of time is capable of being effected.

Furthermore, should we be willing to accept the premise that no appreciable effect upon the minimum rate of creep is caused by this accelerated method of loading, it is reasonable to expect that all of the creep data necessary to plot the curve of the logarithm of the minimum rate of creep against the corresponding stress can be obtained from a test of one specimen. The specimen would then be loaded and quickly strained to a point slightly below that for the minimum rate of creep. The load would be reduced then sufficiently to produce a rate of creep within the desired range. After the creep rate had been found for this stress, the load would be decreased and the new creep rate at this stress obtained, and so on. In this manner it seems likely that enough information could be secured within a period of several weeks or months to afford a reliable indication of the properties of the material.

Several tests of this kind have been made with apparent success, but further study is required to substantiate the degree of accuracy of this form of test procedure. The points for rates of creep of 10^{-4} shearing strain per hour, and less, in the experiments at 400 deg. cent. and all the tests at 500 deg. cent. were obtained in this manner. These data are represented by the curves plotted as stress against the corresponding logarithm of the minimum rate of creep for the particular temperature in Figs. 18 and 19.

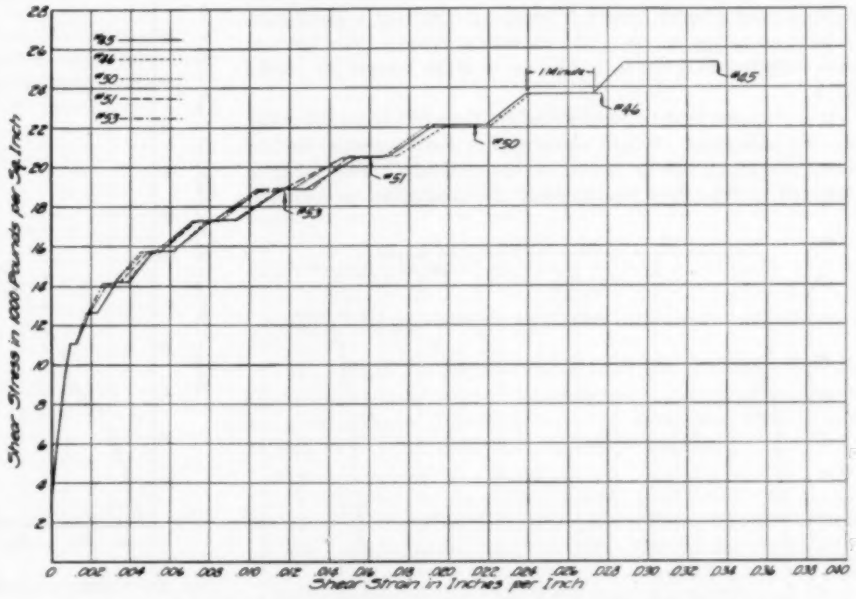


FIG. 21 LOADING CURVES FOR TUBULAR SPECIMENS AT 400 DEG. CENT.

V—SPECIAL PROBLEMS

1 INCREASE OF YIELD POINT BY CREEP AND RESTORATION OF ORIGINAL YIELD POINT BY ANNEALING

In order to investigate the effect of creep upon the physical properties in subsequent room-temperature tests, a tube (No. 56) which had already undergone considerable creep was studied. Following the test at 400 deg. cent., which was reported in Fig. 11, the specimen was unstressed and permitted to cool slowly in the furnace. At room temperature the tube was loaded in torsion until the yield point was reached at about 29,450 lb. per sq. in., and then unloaded. Then by annealing the tube at 900 deg. cent. in the usual manner, all effect of any previous cold working was removed. The tube was again tested at ordinary temperature in twist and the yield point observed to be about 24,950 lb. per sq. in. The respective curves are given in Fig. 22. From these experiments it is seen that the creep test at 400 deg. cent. caused an increase in the yield point of about 20 per cent, based on the usual tests at room temperature.

This phenomenon differs from what we know of room-temperature tests where yield point is increased only by large loading. This increase of yield point was produced by comparatively small loads. It is also significant that Hooke's law is maintained practically up to the value of the new yield point. Usually in cold working, the increased yield point is attended by a marked rounding off of the curve preceding the yield point.

From other tests on unused specimens in torsion at room temperatures, curves like the one shown in Fig. 22(*b*) have been obtained. The material, then, which was subjected to creep and later annealed, resembles the original condition of the steel.

2 NATURAL JARS AND VIBRATIONS

It was felt for some time that vibrations, and particularly jars produced in the building, caused dynamic in addition to the static loads applied to the specimen under torsion. Although all observations of suddenly high creep rates did not follow known jars, these higher values were observed when jars were known to have occurred. During the assembly of heavy apparatus in a nearby room very erratic rates of creep were observed.

In normal times, rates were usually alternately high and low in reference to a smooth curve. In analyzing this condition it was discovered that the reading taken in the morning was the lower one and the evening reading the higher. Observations were recorded twice daily. This appeared consistent, since more jars, however small, were produced during the day, which resulted in the greater rate of creep observed at the evening reading.

A more detailed study was made over a four-hour period one evening when jars and vibrations were at a minimum. Readings of creep were recorded at five-minute intervals on two specimens (Nos. 53 and 55). Reference to the curves of Fig. 23 is suggested.

No creep was observed for the first hour and a half in the specimen whose average rate was about 3×10^{-4} shearing strain per hour until the observer unintentionally caused a sudden jar across the room. This jar resulted in an apparently instantaneous large creep, after which no further change was observed in the remaining two and a half hours. Rates of the order of 4×10^{-7} strain were observed without difficulty, and form the least count for plotting the curve.

The second specimen, whose average rate of creep was slightly greater, about 8×10^{-6} shearing strain per hour, showed alternate stages of creep and of no creep. When the small jar reported above was produced, these observations also showed that a considerable movement had taken place.

It is interesting to note that there is quite an obvious average rate of creep noticeable in the case of the upper curve of Fig. 23. Just before the sudden creep, the specimen is apparently strained less than the average amount, which prepares it particularly well for the occurrence which follows. The jar, then, not only brings the strain up to the "average rate line," but causes excess straining which results in a slowing up in the rate. The period which follows appears to indicate that the material tends to maintain an average rate of creep.

From these experiments it would be concluded that creep is a discontinuous process. Whereas creep is generally read as being continuous, it is probably in reality stepwise in nature.

A special apparatus has been constructed which is designed to make a photographic record of creep by directing a small spot of light on to the mirror of the measuring device. The beam is then reflected back on to a film on a drum turned by clockwork.

It is proposed later to make a study of the effect of intentionally applied pulsations. A light motor with a small eccentric load to cause vibrations will be placed on the load pan. This study is designed to reproduce qualitatively the conditions encountered

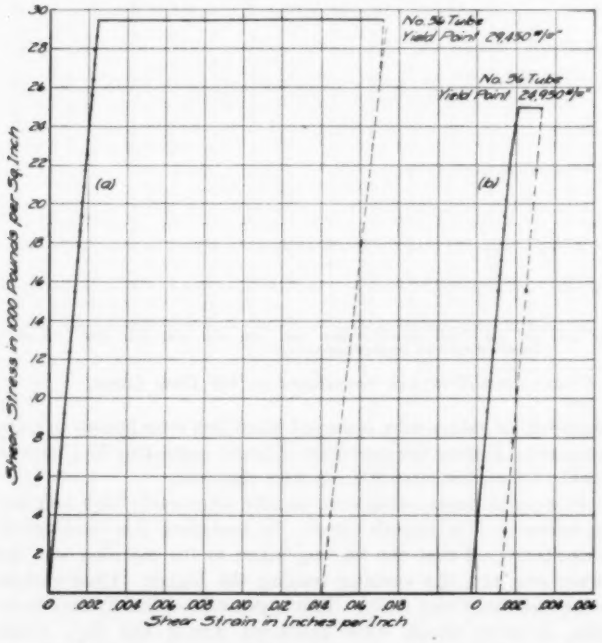


FIG. 22 TORSION TESTS AT 26 DEG. CENT.: (*a*) AFTER CREEP TEST; (*b*) AFTER SUBSEQUENT ANNEALING

in power plants where pulsations are frequently imposed in addition to static stress and high temperature.

From the study of jars which has been made, it was decided to mount the torsion-test equipment on springs. Springs were chosen in such a way as to make the frequency of the system a very low one so that outside jars and vibrations would not produce disturbances upon the torsion apparatus. The machines were therefore suspended from above, as seen by reference to Fig. 2. This eliminated the trouble caused by jars and vibrations. The temperature fluctuation effect is discussed in the following article.

3 TEMPERATURE FLUCTUATIONS

We must anticipate variations in temperature throughout any experiment of the order of one degree centigrade above and below the desired amount. In tension tests we know that these fluctuations materially affect the sensitivity of the readings of creep. They are far less serious in the case of torsion, but it is worth while to study the magnitude of their effect. An increase in temperature results in an increase in both the polar moment of inertia and the length of the specimen, and a decrease in the modulus of elasticity in shear. The resistance of the specimen to shear, then, tends to increase with a larger polar moment of

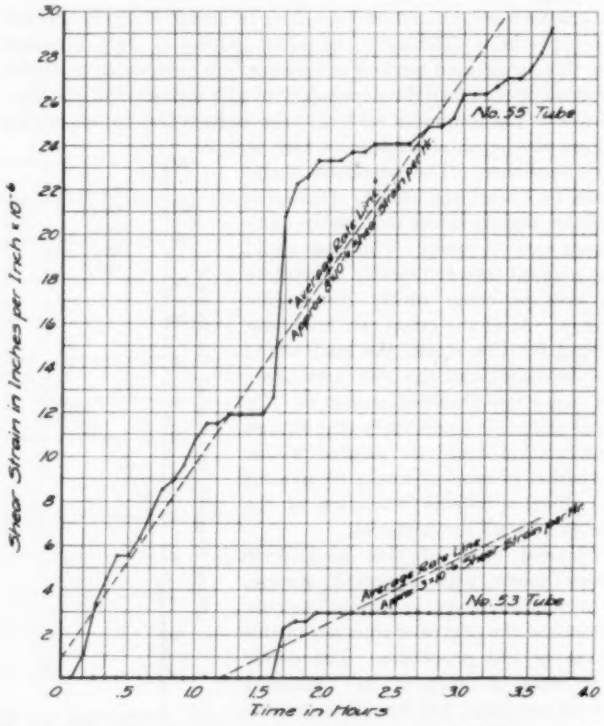


FIG. 23 CONTINUOUS RECORD OF CREEP FOR LOW RATES AT 400 DEG. CENT.

inertia, but shows a decrease with greater length and smaller modulus. The temperature effect upon modulus of elasticity in shear is very great compared with that upon the polar moment of inertia and the length. The ratio, in order, is about 150:5:1.25.

In the extreme case when the applied torque moment is large, the effect due to a two-degree-centigrade temperature fluctuation will be read as a shear strain of approximately 1×10^{-6}. With intervals of twelve hours between creep readings the effect of temperature fluctuations upon creep measurements amounts to rates of less than 1×10^{-7} shear strain per hour. The following calculations for an extreme particular case will help to make this evident.

Let:

θ = angle of twist in radians

t = 400 deg. cent., say, the desired temperature of specimen

M_t = 1×10^3 in-lb. = maximum torque moment

L = 4 in. = gage length at temperature t

G = 1×10^7 lb. per sq. in. = modulus of elasticity in shear at temperature t

I_p = 4×10^{-3} = polar moment of inertia of tubular cross-section at temperature t

c = 0.4375 in. = mean radius of tube at temperature t

$\Delta\theta$ = difference in angle of twist in radians due to temperature change

α = 1.25×10^{-6} in. per in. = change per inch of L due to 1 deg. cent. temperature fluctuation at temperature t (coefficient of expansion)

β = 150×10^{-6} lb. per sq. in. = change per unit of G due to deg. cent. temperature fluctuation at temperature t (determined experimentally; this is discussed in Section III and an experimental curve is given in Fig. 15)

γ = $4 \times 1.25 \times 10^{-6}$ in.^{4} = change per unit of I_p due to 1 deg. cent. temperature fluctuation at temperature t

δ_s = shear strain in tubular specimen.

Then, due to a fluctuation of $\pm$ 1 deg. cent. from temperature t,

$$\theta_{t \pm 1^\circ C.} = \frac{M_t L(1 \pm \alpha)}{G(1 \pm \beta) I_p (1 \pm \gamma)}$$

Therefore the difference in the angle of twist expressed in radians due to a 2-deg. cent. temperature change is, since α, β, and γ are small quantities,

$$\Delta\theta = \frac{2M_t L}{GI_p}(\alpha - \beta - \gamma)$$

and by substitution of the values for the symbols:

$$\Delta\theta = \frac{2 \cdot 1 \cdot 10^3 \cdot 4}{1 \cdot 10^7 \cdot 4 \cdot 10^{-2}} (1.25 \cdot 10^{-6} - [-150 \cdot 10^{-6}] - 5 \cdot 10^{-6})$$

so

$$\Delta\theta = 3 \times 10^{-6} \text{ radians}$$

Since

$$\delta_s = \frac{c \cdot \Delta\theta}{L}$$

Then, due to an angular deformation $\Delta\theta$ caused by a 2-deg. cent. temperature change, the resulting shear strain becomes

$$\delta_s = \frac{0.4375 \cdot 3 \cdot 10^{-6}}{4}$$

or

$$\delta_s = 3 \times 10^{-6} \text{ in. per in. strain}$$

When readings are made at 12-hour intervals, the effect of temperature fluctuations which will result in fluctuations in creep rate will be

$$\delta_s/\text{hr.} = \frac{3 \cdot 10^{-6}}{12}$$

or

$$\delta_s/\text{hr.} = 2.5 \times 10^{-7} \text{ strain per hour}$$

The figures in the results of these calculations have in several cases been rounded off to the nearest whole number. This latter value, then, represents the minimum reliable value which may be observed when temperature fluctuations of the specimen are in the order of $\pm$ 1 deg. cent. and all other conditions noted above regarding torque moment and dimensions are imposed. This temperature fluctuation registers its effect in changing the modulus of elasticity in shear, the polar moment of inertia, and the length.

It will be appreciated that as the torque is decreased, as is the case when small rates of creep are required, the magnitude of the effect due to temperature fluctuations is decreased in direct proportion. Rates of 1×10^{-6} strain per hour under such circumstances can readily be determined without difficulty when readings are made as frequently as 12 hours.

ACKNOWLEDGMENTS

Prof. S. Timoshenko conceived the importance and possibilities of investigating creep phenomena by uniformly pure shear, presented the problem to the author, and directed the entire program. To him credit is due for the vision and practical success of the project up to the present time. The author wishes to record his deep appreciation to Professor Timoshenko, moreover, for his personal interest and limitless encouragement.

To Prof. A. E. White and the Department of Engineering Research, of which he is director, the author wishes to indicate his appreciation for sympathetic interest in the problem, which included the assistance in the matter of personal expenses. The author also acknowledges with gratitude the generous support of the Department of Engineering Mechanics under the supervision of Prof. E. L. Eriksen, especially for the funds for the construction of the special equipment used in conducting the investigation.

Discussion

T. McLean Jasper.[6] This paper indicates considerable ingenuity in testing. There are, however, two very important assumptions which are made, which are very difficult to substantiate. They are that a thin tube in torsion will allow one to reach the yield point of the metal without buckling at a much lower stress, and that one can discover the ultimate strength in torsion of even a brittle material when using a thin tube and the mathematics quoted.

The problem is analogous to that of the slender column in which the apparent yield point of the same steel is reached at various apparent stresses, depending upon the slenderness ratio of the column used. The ultimate strength, in this case, is very much obscured.

It is believed that the method of testing adopted by the author should be very good in determining the shearing modulus of steel at high temperatures, provided that the whole of the gage length of the specimen is known to be at a definite temperature and that the straight portion of the stress-strain diagram is used.

The best method to determine this modulus that has been devised, it is believed, was adopted by Sutherland, in which he used the torsion pendulum for his work. This was reported in *The Philosophical Magazine* about 1891. The work by Sutherland has been summarized for steel and other metals in the magazine of 1923, pp. 617–618.

The writer is somewhat nonplused by the nonchalant assumption that the maximum-shear theory of failure has been proved.

The most exhaustive set of tests on which this is generally based was performed by Guest, who made a very detailed tabulation of his results in *The Philosophical Magazine*. The writer would call attention to the fact that Guest made as many as 20 tests on some of his thin tubes. He tested them in a simple torsion, and in torsion combined with other forms of stress. After a test in simple torsion on a tube, he made as many as 15 tests in some cases, and in others 18, in which he exceeded the straight-line stress-strain relation by other combinations of stress. When he finally tested by simple torsion, he was able to repeat the results originally obtained. Any one familiar with exceeding the elastic limit or yield point of steel knows that the new values obtained on the same specimen will not check with the original ones. The writer is not criticizing the test values obtained by Guest, because they were made in a

[6] A. O. Smith Corporation, Milwaukee, Wis. Mem. A.S.M.E.

most painstaking manner and by a man most trusted as an investigator. What he is suggesting, however, is that Guest never exceeded the elastic properties of the steel in his test tubes, but that they buckled considerably below the elastic limit of the steel because of the slenderness of these tubes. The writer has proved that slender torsion tubes will buckle in this manner by actual tests, and has obtained results in torsion of thin tubes similar to the Euler curve for columns by varying the ratio of thinness.

J. J. KANTER.[7] Incidental to his studies on creep under shear stresses, the author has considered possibilities of structural change in steel due to the effects of high temperature alone. He offers some data which indicate that in less than 30 days of continual heating at 450 deg. cent. (842 deg. fahr.) appreciable loss of tensile strength, yield point, proportional limit, and Rockwell hardness is sustained by low-carbon strip steel, normalized. It might well be inferred on the strength of these data that in a year of heating, for instance, much greater difference than measured in 30 days would be found. The author suggests that more experimental evidence on this point is to be desired.

It has been shown by several investigators that carbon steel tested in creep-elevated temperature is not seriously changed in its mechanical properties. Tensile tests have been made upon specimens subjected to as much as a year of creep testing. Tests such as these seem to show only a slight annealing at temperatures in the range of 900 and 1000 deg. fahr. (483 and 538 deg. cent.). Creep testing in the range of 750 deg. fahr. (400 deg. cent.) tends to produce a slight increase of tensile strength and yield point, presumably due to strain hardening. Carbon steel does not seem to deteriorate in any alarming degree as a result of exposure to creep conditions.

The conditions for structural change which the author considers in the present discussion, however, do not involve stress—only temperature and time. Some tests of this type involving

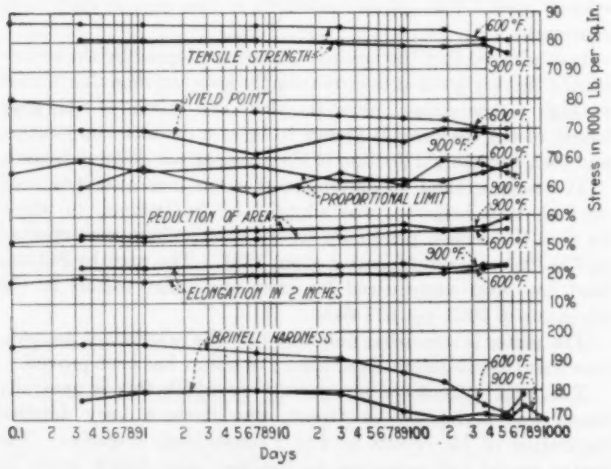


FIG. 24 MECHANICAL TESTS ON 0.45 PER CENT CARBON-STEEL ROD AT ROOM TEMPERATURE

(After various lengths of heating at 600 and 900 deg. fahr., 315 and 483 deg. cent. Analysis: C, 0.45; Si, 0.08; Mn, 0.54; S, 0.04; P, 0.012. Heat treatment: Normalized at 1650 deg. fahr., 900 deg. cent.; quenched from 1500 deg. fahr., 815 deg. cent.: drawn 2 hours at 1500 deg. fahr., 678 deg. cent.)

up to three years' exposure will probably be of interest at this time. Figs. 24 and 25 represent the mechanical test results of two carbon-steel materials given long-exposure tests at 600 and 900 deg. fahr. (315 and 483 deg. cent.). The results represent part of a comprehensive survey of various steels and alloys, under long-time heating conditions, started in 1923. The

[7] Metallurgical Department, Crane Company, Chicago, Ill.

soaking temperatures are maintained constant and uniform in two cast-iron blocks each containing a number of drilled holes to admit test specimens suitable for machining to standard 0.505-in. tensile bars. These blocks are insulated with magnesia and kept heated to temperature by means of thermostatically controlled gas burners. The variations in temperature amount to about ±25 deg. fahr. (±15 deg. cent.) of those stated. The heating blocks were kept continually at temperature, specimens being withdrawn at desired intervals.

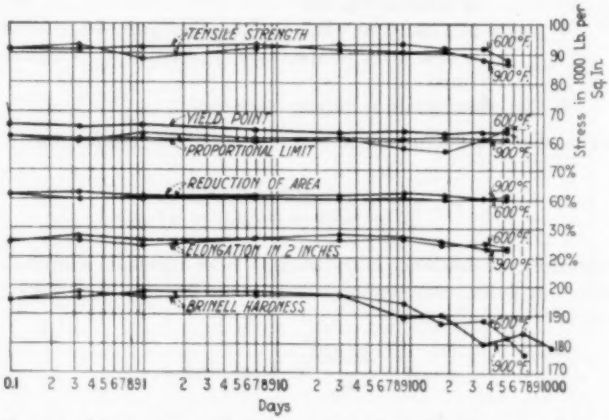


FIG. 25 MECHANICAL TESTS ON COLD-ROLLED BESSEMER-STEEL ROD AT ROOM TEMPERATURE

(After various lengths of heating at 600 and 900 deg. fahr., 315 and 483 deg. cent. Analysis: C, 0.09; Si, 0.03; Mn, 0.72; S, 0.105; P, 0.089.)

The effects of long exposure at 600 and 900 deg. fahr. (315 and 483 deg. cent.) of 1 in. diameter 0.45 per cent carbon-steel rod are shown in Fig. 24. These specimens were uniformly heat treated to 195 Brinell hardness, in the following manner:

Normalize at.....	1650 deg. fahr. (900 deg. cent.)
Quench from.....	1500 deg. fahr. (815 deg. cent.)
Drawn 2 hr. at.....	1250 deg. fahr. (678 deg. cent.)

Very little change in tensile strength was found for heating periods up to 18 months at either 600 or 900 deg. fahr. (315 or 483 deg. cent.). At 900 deg. fahr. (483 deg. cent.) after 6 months of exposure a slight loss is apparent. Changes in yield point and proportional limit, elongation, and reduction of area are similarly of small magnitude up to 18 months of heating. The changes in Brinell hardness are recorded up to 24 months at 600 deg. fahr. (315 deg. cent.) and 40 months at 900 deg. fahr. (483 deg. cent.). Neither of these temperatures produced a definite hardness change until after 30 days of heating had been exceeded. A greater percentage of loss in hardness than in tensile strength apparently is sustained in the long-time heating.

Long-time heating tests upon 1-in.-diameter cold-rolled bessemer steel rod are shown in Fig. 25.

Unlike the heat-treated 0.45 per cent carbon steel, there are marked differences in the changes in mechanical properties brought about in raising the temperature from 600 deg. fahr. (315 deg. cent.) to 900 deg. fahr. (483 deg. cent.). Gradual loss in tensile strength, yield point, and Brinell hardness, with corresponding gradual rise in elongation and reduction of area, are shown at 600 deg. fahr. (315 deg. cent.). At 900 deg. fahr. (483 deg. cent.), however, but 8 hours of heating was required to bring about the approximate total change in properties which accumulated in a year at 600 deg. fahr. (315 deg. cent.). Given time enough, therefore, 600 deg. fahr. (315 deg. cent.) is a high enough temperature to effectively relieve the hardening and internal straining due to cold work. At 900 deg. fahr. (483 deg. cent.), where the same relief is accomplished in a few hours,

further heating up to 3 years produces no drastic additional changes.

The changes which do take place at 900 deg. fahr. (483 deg. cent.) from 8 hours on are quite similar to those shown in Fig. 24 for 0.45 per cent carbon drawn steel.

These tests, together with those of the author, seem to indicate that the principal possible source of instability of mechanical properties in low- and medium-carbon steels, not hardened, is due to internal strain. The rate at which such strain is relieved apparently is a function of temperature and time, not dependent upon any critical or recrystallization temperature; 600 deg. fahr. (315 deg. cent.) is seemingly as effective over a long-time period as 450 deg. cent. (842 deg. fahr.), which is ordinarily considered to be the lowest temperature for recrystallization. The $^1/_{16}$-in.-thick strip steel used in the author's tests, although annealed at 900 deg. cent. (1652 deg. fahr.) before the exposures, may have retained some of the effects of the drastic cold working which thin strip steel is usually subjected to. The losses found by the author in the period between 5 and 30 days of heating at 450 deg. cent. (842 deg. fahr.) are of much greater magnitude than those found for similar intervals in the tests upon the two rod steels at 900 deg. fahr. (483 deg. cent.). It is quite probable that the changes take place more rapidly in $^1/_{16}$-in.-thick strip sections than in 1-in.-diameter rods.

On the strength of what evidence is at hand, there need be little anxiety over the deterioration or troublesome structural changes as a result of heating in ordinary structural carbon-steel products. It is of course known that certain heat-treated alloy stee's are affected structurally and chemically in the course of heating, but so far as low- and medium-carbon steels are concerned it is found that heating, under ordinary conditions, affects principally the state of strain hardening, whether stress is being applied or not.

GEORGE C. PRIESTER.[8] The author should be commended on the painstaking and ingenious method devised for the determination of his data. The phenomenon of creep in metals subjected to stress is of prime importance in several phases of engineering design. While most investigators have confined their analyses of this phenomenon to specimens subjected to tensile stress, this is the first paper that has come to the writer's attention which deals with creep when the specimen has been subjected to a shearing stress. The fact that the shearing deformation of a specimen subjected to torque can be measured with a much higher degree of accuracy and with simpler instruments has long been recognized by many investigators in the properties of the strength of materials, but as yet very little use has been made of this fact It remained for the author of this paper to demonstrate this, and it is the hope of the writer of this discussion that more use will be made of it.

Some six years ago in the materials-testing laboratory of the University of Minnesota, an instrument employing the use of the Ames dial was devised for measuring shearing deformation in specimens subjected to torque. The basic principle of this instrument is the same as that used by the author. The results obtained were gratifying and even beyond the expectation of the writer as to the accuracy with which these deformations could be read. This instrument is still in use.

In referring to the author's paper, in Section III-4, where he states that "This curve does not ook like those usually obtained for ultimate strength in tension at different temperatures," it is not quite clear to the writer just what significance this may have, when one compares the different manner in which an ordinary tension test is carried out with the one used by the author. In the ordinary tension test the specimen is stressed by producing a deformation in it. The author stressed his specimen by loading the specimen by direct torque, and naturally if any phenomenon of necking did take place, it would be practically impossible to measure stress and strain simultaneously in the range in which necking takes place by the method he used. Also the specimens the author used would preclude any phenomenon of necking.

In the second paragraph of Section IV, the author states that "It was noted that the minimum rate of creep in each instance occurred at approximately the same total shear strain." He also states that this occurs at a shear strain of about 0.10 in. per in., and then later on in his paper used this as a basis for "Accelerated Methods of Testing." It seems to the writer that the author would have encountered less difficulty and obtained a more reliable basis for the accelerated tests had he used the minimum rate of change of creep instead of the minimum rate. From Fig. 17 of the author's paper it will be noted that there is a minimum rate of change in all the curves at about 0.08 in. per in. shear strain. Also this minimum rate of change of creep is not affected by any buckling phenomenon and occurs at a point which could be determined with more accuracy than the one suggested by the author. By selecting the points of inflection in a set of curves there is also a greater possibility that the phenomenon will lend itself to a more rational mathematical treatment should one desire to investigate the problem from this standpoint.

On the whole the author is to be congratulated on the thorough manner in which he undertook and carried out his tests on this new phase of creep. The writer sincerely hopes that he may be able to continue productive work along this line.

A. E. WHITE.[9] This is a material contribution to the data which is accumulating on this subject. So far as the writer is aware, this is the first time that an outstanding attempt has been made to measure shear at elevated temperatures. The torsion machine which was developed for the measurement of shear represents a distinct contribution. One of the advantages which it possesses is the fact that shear creep measurements, at least in accordance with the method described in this paper, are not materially affected by the coefficient of expansion, which value always has to be considered when creep values of a high degree of accuracy are made in tension.

P. G. MCVETTY.[10] The paper is a valuable contribution to our knowledge of creep of metals at elevated temperatures. While it has not yet been proved that creep results from shear within the individual crystals rather than from some form of viscous flow of the intercrystalline material, a study of shear phenomena appears to be a step in the right direction.

Some mention should probably be made of the pioneer work of R. W. Bailey in this field, reported at the World Power Conference in Tokyo in 1929. He also used a tubular specimen and a similar method of applying a torque moment, and in addition, studied the effects of a superimposed axial stress. His conclusion that creep also occurs on planes other than those of maximum shear appears to be worthy of further consideration. This indicates that creep is even more complex than was supposed, and the necessity for further fundamental research is evident.

There are several questions which may be asked relative to the methods used in measuring and controlling temperatures. The effects of temperature fluctuations are undoubtedly less serious

[8] Professor of Materials of Engineering, University of Minnesota, Minneapolis, Minn.

[9] Professor of Metallurgical Engineering and Director of the Department of Engineering Research, University of Michigan, Ann Arbor, Mich. Mem. A.S.M.E.

[10] Mechanical Engineer, Research Laboratories, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

in a torsion test than in the usual tension creep test. However, it is assumed that the temperature is uniform over the gage length, and it would be interesting to know the extent of temperature variation within the length of the test specimen. If, for example, the specimen is 10 deg. hotter at the center than at the ends of the gage length, the angle of twist will not increase uniformly from one end to the other, and a considerable error may be introduced.

From a study of the apparatus used it is probable that considerable heat is conducted away from the specimen through the upper and lower grips and the strain-measuring device. This results in temperature gradients within the furnace, and it would be desirable to have more information about the location of the thermocouple used for temperature measurement and control.

The writer has also found a difference between readings taken in the morning and in the evening. This difficulty was experienced in some early tests before instruments were used to furnish a continuous record of furnace temperatures. This was finally traced to faulty temperature control during the night, resulting in a lack of temperature equilibrium in the apparatus at the time of the morning reading. The trouble has been completely eliminated by improvements in temperature control.

There is now much evidence to indicate that creep is a discontinuous process. The sudden slips observed by the author are probably stopped by strain hardening, after which a considerable time at the test temperature may be necessary to produce softening and further slip. In the April, 1931, issue of MECHANICAL ENGINEERING, Dr. Jeffries mentioned some interesting observations of this phenomenon as affected by grain size and method of loading. The author's proposed tests with pulsating loads should throw light on the important question as to whether the results of static creep tests are applicable to service conditions.

The apparatus developed by the author is admirably adapted to the study of very low creep rates, and the results of his further tests at low stresses will be awaited with much interest.

A. NÁDAI.[11] Experiments on the phenomenon of creep are today very important. In the industrial laboratories much work has been done during past years to determine the behavior of metals under stress at elevated temperatures. In most of these tests cylindrical specimens have been subjected to tensile loads and the extension of the specimens observed while the load was kept constant over a long period of time. The observation of a small longitudinal extension of a comparatively short gage length in a tensile test is difficult. Mr. P. G. McVetty has called attention to sources of error due to such arrangements. Small changes of the temperature of the bar may have a great effect on the observed elongation due to the inequality of the temperature between the test piece and the comparison strips of the extensometers; the thermal expansion of the latter introduces errors in the readings of the extensometer which may become serious. It is therefore a step toward the improvement of creep-testing methods used in industry to test materials, according to the suggestions of Professor Timoshenko, in torsion instead of in tension. Shear deformations are not affected by volume changes due to thermal expansion. The shear-testing apparatus designed by Dr. Everett might prove useful as it eliminates certain difficulties encountered in the usual methods in long-time testing.

On the other hand, certain new difficulties seem to be introduced by the use of thin-walled tubes subjected to torsion on account of the instability of the equilibrium of such tubes. The compressive stresses acting at 45 deg. to the axis of the tube might produce buckling. Thin-walled tubes at higher temperatures than those applied by Dr. Everett would probably buckle because of the known lowering of the critical buckling forces as soon as plastic buckling occurs, as compared with the critical values of the forces of purely elastic deformations.

The work of experimenters who have investigated the plastic range of the deformations at normal and at elevated temperatures under various stress conditions, is closely related to the problem of creep investigated in the paper. Mention should have perhaps been made of some of the elaborate tests which have been carried out under combined stress and in which the rules of plastic flow at normal temperatures have been carefully studied. Among these tests are, especially, those made in England (several years ago), more recent tests made in Germany (W. Lode[12]) and in Switzerland (Rôs and Eichinger[13]) which might also be of interest in connection with the subject of the present paper, as some of the results of these tests might have their useful applications also to the phenomenon of creep. Torsion tests with metals at elevated temperatures have been made previous to those reported here by R. W. Bailey[14] in England.

Referring now to some of the experimental results, mention should be made that the logarithmic rule as expressed by Fig. 18 was discussed by P. Ludwik as far back as 1908.[15] Presumably Ludwik was the first to propose the formula that the stress under which a metal yields slowly is proportional to the logarithm of the ratio of the velocities of deformation with which the bar is extending. This was found for metals having a low melting point and under room temperature, that is, perhaps under conditions which in some way may correspond to creep tests of steel at elevated temperatures. A logarithmic equation has been suggested independently by Cassebaum,[16] expressing the ordinates in the ordinary tensile stress-strain diagram of steel taken for the same amount of deformation, but at various speeds of extension, referred to each other. From his torsion tests, R. W. Bailey finally arrived at a similar conclusion. This last investigator distinguishes also as Dr. Everett does between three phases of the creep test under constant load, and he finds from his torsion test that if he plots the logarithms of the rate of creep for what he calls the second stage of creep (this corresponds to Dr. Everett's "minimum creep rate") as the abscissas and the stresses as ordinates, he obtains a straight line.

In Section II, in the last paragraph, the author quotes the ratio 0.6 as the value of the yield point in uniform shear to that in tension at normal temperatures,[17] finding that this is in good agreement with the maximum-energy theory. If the full amount of the maximum-strain energy, including that of the volume dilatation, was meant here, this statement needs to be corrected as it can readily be seen that the maximum-energy theory cannot be, in general, a correct criterion of the conditions of yielding.

Although it would seem at first glance as if on account of similar

[11] Research Laboratories, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

[12] Mitt. u. Forschungsarbeiten des V.D.I. Berlin, Heft 303, 1928.

[13] Proc. Eidgen, Materialprüfungsanstalt, Zurich.

[14] These tests are reported in the Transactions of the International Power Conference, 1929, Tokyo, vol. 3, p. 1089, "Creep of Steel Under Simple and Compound Stresses and the Use of High Initial Temperature in Steel Power Plant." Compare also two valuable papers by Bailey, one on "Creep of Steel Under Simple and Compound Stresses," p. 265, and the other on "Tubes and Cylinders Under High Pressures and Temperatures," p. 772, in *Engineering*, London, 1930, vol. 129. These papers not only contain very valuable experimental information but also present their applications for computing stresses under creep in cylinders at elevated temperatures.

[15] "Elemente der Technologischen Mechanik," Berlin, 1908.

[16] *Annalen der Physik*, 4 Serie, vol. 34, p. 106, 1911.

[17] The figure 0.6 mentioned may also depend on the ratio of the wall thickness of the tubes tested to their outer diameter, if computed according to the "Code for Design of Transmission Shafting," Eng. & Ind. Standards, B17c-1927.

results obtained by independent investigators, consideration should be given to apply these results at once in practical work, the writer thinks that as far as creep at elevated temperatures is concerned, this at the present time might be still premature. The observations mentioned above with regard to rules combining stress and velocity of deformation refer to various modes of yielding, namely, plastic yielding of steel at room temperature (Cassebaum) and also to creep (Ludwik, Bailey, Everett). This seems at first astonishing, as it is assumed that the mechanism of plastic flow at normal temperatures in polycrystalline materials such as steel may differ from that of such materials at elevated temperatures where the thermal agitation of the atoms has more and more influence. Observations by Mr. McVetty at the Research Laboratories seem to indicate more and more definitely that certain very slow processes taking place in a metal exposed to elevated temperatures, may, over a period of months and perhaps years, depending on the conditions, still influence the creep characteristics of a metal subjected to a long-duration test. Much experimental work will still have to be done in investigating these and similar factors until it is possible to establish quantitative rules for the slow deformations of the metallic alloys at elevated temperatures.

Although many points may need further approval in this difficult field of investigation of creep all new contributions should be much appreciated, especially the valuable work presented by Dr. Everett, in which he has attempted to attack various points from new sides. It is hoped that by repeating such efforts it may be possible in the succeeding years to arrive at a more satisfactory understanding of the slow movements and deformations of metals at higher temperatures.

Author's Closure

Mr. Jasper suggests that buckling of a thin tube in torsion may occur at stresses below the yield point of the material. The ratio of the thickness of the tubes used in the investigation discussed in the paper to the diameter was about 1 to 20, and the matter of buckling of tubes with such proportions at stresses below the yield point is absolutely impossible. In discussing the information given in Table 2 and later in connection with Fig. 22, the author explained the results of representative experiments indicating that the stability of the thin tubes was maintained at least beyond the yield point. The problem of buckling was encountered in stresses well in excess of the yield point, as has been covered in Section II-3.

The second point raised by Mr. Jasper considers the ultimate strength in torsion. Since stress concentration, although of great importance in brittle materials, does not enter into the problem of ultimate strength of a ductile material, a tubular specimen with a reduced section so short that buckling is impossible was investigated. The cross-section of this short tubular specimen was identical with that of the long specimen, but the possibility of buckling was entirely eliminated by the shortness of the small section as can be seen in Fig. 8.

The modulus of elasticity in shear at various temperatures was determined from the straight-line portion of the curve of shearing stress and shearing strain upon unloading the specimen. The effect of creep during unloading was found to be practically negligible, and a relatively long straight-line portion was obtained from which the modulus in shear was calculated with considerable accuracy. Care was exercised in providing uniform temperature conditions over the gage length of the tubular specimen. As can be seen by reference to the curve of Fig. 15, a variation of even ± 5 deg. cent., which is an extreme range, would not appreciably affect the character of the curve.

In practice many designers adopt for ductile materials the maximum-shear theory—mentioned in the Introduction of the paper—as being in most general agreement with analysis and experiment. In some cases other theories appear to fit the conditions more closely. Some of the results of the preliminary experiments made by the author coincide with the maximum-energy theory, as reported in Section II-2.

Mr. Jasper mentions the work of Guest in a complimentary manner. Guest himself favored the maximum-shear theory as a basis for determining working stresses.

In Mr. Kanter's tests, the specimens were machined subsequently to the heating period, whereas the thin specimens studied for the effect of long-time heating and reported in the paper were tested "as heated." In the latter the additional possibilities of weakening by decarburization and oxidation of the surfaces formed the reason for selecting very thin strips having a large ratio of surface area to cross-sectional area, which paralleled the condition encountered with thin tubes used in the creep tests. Mr. Kanter's tests compare as to the possibility that further releases of internal strains are not completely removed in the preliminary annealing and the growth of grains, with consequent weakening effect.

Mr. Kanter's generalization that higher temperatures have the same effect upon the physical properties as do lower temperatures operating over a longer times seems justifiable. It is known that internal strains which will become released after several months at ordinary room temperatures will be released to the same extent by placing the specimen in boiling water (100 deg. cent.).

Regarding some of the questions propounded by Professor Priester, in determining the ultimate strength of carbon steels, at least, in tension at various temperatures, it is usual to find a maximum value in the vicinity of 200 to 300 deg. cent. A gradual improvement in tensile strength frequently precedes the maximum point. Since the results obtained for the ultimate strength in shear appeared to disregard both of these characteristics, it is of interest to make this observation. We can usually look to what takes place in shear for an interpretation of tension conditions, but in ultimate strength, necking in tension tests, in addition, perhaps, to other factors, appears to contribute a further complication.

Professor Priester suggests that greater accuracy is obtainable in locating the point of minimum rate of creep by considering the minimum rate of change of creep. Curves similar to that shown in Fig. 20 and mentioned in Section IV-1 were made as the tests reported in Fig. 17 and others progressed. From such curves the minimum rate of change (zero rate) was readily observed.

Professor White has emphasized an important point in mentioning the higher accuracy obtainable by torsion methods compared with those in tension as affected by temperature variations of the test specimen.

Mr. McVetty points out, and properly so, that the investigation of creep in tubes by R. W. Bailey deserves mention. The author has studied his work with considerable interest, but it appeared only in the early part of 1930 and the investigation discussed in this present paper had progressed considerably by that time. The complete bibliography in connection with the present investigation includes the contributions of Mr. Bailey.

It appears important, as Mr. McVetty mentions, that a knowledge of the temperature at all points of the specimen be possessed. In some preliminary experiments early in the investigation as much difference as 10 deg. cent. was observed over the gage length of 4 in., representing the very worst condition that has been found. This information on temperature was the result of placing three thermocouples on the specimen, at the gage points and the mid-point, and reading on one potentiometer. The error caused by a temperature differential along the length of the tube specimen only affects strain readings as a result of a different value of the modulus of elasticity in shear, Fig. 15, as the temperature deviates from the correct amount, unless, of course, there is a

fluctuating temperature, in which case the discussion under Section V-3 applies. The error due to a constant temperature condition, admitting a small temperature differential along the gage length, is then seen to be really inconsequential. It means that the creep test is being conducted within a small temperature range, and that the creep readings are not caused to vary as a direct result of this.

There is actually a generous transfer of heat by conduction outward through the grips which hold the specimen. Both the check of the temperature at three points along the specimen as described above and the relatively long furnace, which has a ratio of about three to one compared with the specimen, assure reasonable temperature conditions. During every experiment a thermocouple attached to the mid-point of the specimen is connected to an indicating potentiometer which describes the temperature of the test. The thermocouple which leads to the automatic temperature control is fastened to the mid-point of the alundum tube of the furnace. Each furnace is equipped with its individual control instrument.

Observations from the indicating potentiometer have shown no appreciable difference in the temperature during the evening or the day. The fluctuating results of shear strain reported in Section V-2 caused by jars to the apparatus disappeared upon mounting the machines on springs.

Dr. Nádai mentions that volume changes due to thermal changes do not affect shear deformations. The author attempted to show in Section V-3 that changes in gage length and polar moment of inertia, and also modulus of elasticity in shear, combined to produce some possible error appearing in shear-strain measurements. Their combined effort is, fortunately, slight.

The question of buckling was given much careful study both preparatory to and during the long-time creep tests. In Section I-3 some remarks are made to the effect that buckling of the tube walls, if present, was not measurable on specimens after creep tests by means of a micrometer. It appears from the author's experiments that buckling takes place only after relatively large angular distortions, and that the amount of distortion during which creep data are obtained is in all cases considerably below any evident buckling condition. It is to be expected that higher temperatures will cause a lowering in the critical amount of buckling forces, but the shear stresses are also materially decreased in such creep experiments.

Referring to the value of 0.6 for the ratio of the yield point in pure shear to that in tension as found from certain room-temperature experiments mentioned in the paper, it can be said that this value, in a general way, is approximately correct, considering total strain energy and also only that part due to distortion, lying somewhere betweeh the two ratios.

Some reluctance appears to be expressed by Dr. Nádai in the matter of accepting creep data for design purposes. For industrial needs, however, information regarding creep phenomena is required and the available data, although insufficient, must be used. Such men as Baumann, Tapsell, and Hatfield believe that there are sufficient data available from which to draw some practical conclusions.

The author wishes to thank those who have contributed to the the discussion of this paper.

APM-53-11

Recent Advances in Photoelasticity

And an Investigation of the Stress Distribution in Square Blocks Subjected to Diagonal Compression

By MAX M. FROCHT,[1] ANN ARBOR, MICH.

After discussing the principle of optical equivalence of stressed isotropic bodies, the author deals respectively with current methods to determine (P — Q); the fringe method; annealing of bakelite; fringe photographs of various models; membrane method to determine (P + Q); the photoelastic investigation of stress distribution in square plates subjected to diagonal compression; isoclinics; trajectories; (P — Q) from fringe method; P and Q by graphical integration. He then summarizes the approximate theoretical analysis by the strain-energy method, gives comparative curves, discusses results, describes new equipment, and in conclusion, derives fundamental equations for the evaluation of P and Q by graphical integration.



THE growth of high-speed and large-size machinery, as well as the personal influence of a few outstanding men, is beginning to convince industrial executives of the need and value of mechanics as an engineering tool. With the general recognition of mechanics must go a special recognition of photoelasticity. Only the simplest problems in stress distribution can be determined theoretically. Generally it is necessary to employ experimental methods to this end, among which photoelasticity must rank first.

The development of this branch of experimental mechanics has been retarded by the absence of a suitable material and by a certain mistrust among engineers in the validity of its results.

The researches of Professors Mesnager, Coker, Filon, and others have gone a long way to dispel these doubts, and to build up a confidence in the results obtained by the optical method. It has been proved theoretically and substantiated experimentally that within the elastic limit and in simply connected bodies the stress distribution is independent of the physical constants of the material. Professor Filon has shown that even in the case of multiply connected bodies such as connecting rods, the errors due to the difference between the materials of the model and the corresponding machine part or structural member are very small. The introduction of celluloid as a working material made photoelastic research not only possible but practicable.

In very recent years interesting and promising developments have taken place, and it is the object of this paper to discuss some of these developments and their application. In particular the author will present, first, Dr. Tuzi's fringe method for the determination of the difference between the principal stresses at a point; second, a new experimental method for the determination of the sum of the principal stresses; third, a process for the annealing of bakelite which will restore this excellent material to photoelastic work; fourth, a photoelastic and theoretical investigation of the stress distribution in square plates subjected to diagonal compression; and fifth, photographs of new and inexpensive laboratory equipment which will demonstrate that a photoelastic laboratory is financially within the reach of industrial organizations and engineering schools.

Essentials of Photoelasticity

The fundamental optical phenomenon underlying all photoelastic investigations is that of double refraction. It explains what is transpiring in the crossed Nicols, the quarter-plates, as well as in the stressed model itself.

When a ray of monochromatic light enters a doubly refracting crystal, such as Iceland spar, it emerges transformed in two important respects: first, it splits into two plane polarized rays, the ordinary and extraordinary, which vibrate in mutually perpendicular planes; second, the two rays travel through the crystal with different velocities, and therefore emerge from the crystal with a phase difference which depends on the wave length of the light and the length of the crystal. It is an experimental fact that a stressed element in an isotropic body behaves just like a doubly refracting crystal. To generalize: *Within the elastic limit all isotropic bodies subjected to a two-dimensional stress system are optically equivalent to a set of a large number of doubly refracting crystals shaped so that their lengths at every point are directly proportional to the difference between the principal stresses at that point, and whose principal planes are set parallel to the stress trajectories.* It is a further experimental fact that the relative retardation R between the ordinary and extraordinary rays produced at a point of an isotropic body due to its temporary doubly refracting character is given by the equation

$$R = c\,(P - Q) \qquad [1]$$

where c is an optical constant depending upon the material, and P and Q are the principal stresses at the point. Due to the above-mentioned facts a stressed, transparent model in a field of *white, circularly polarized light* gives a colored image each tint of which is a measure of $(P - Q)$. This phenomenon is the foundation of the prevailing methods in photoelastic laboratories today. It is the basic idea in color matching and compensating. Both of these methods depend upon the color effect to determine the numerical value of $(P - Q)$.

---

[1] Associate Professor of Mechanics, Carnegie Institute of Technology, Pittsburgh, Pa. Dr. Frocht received a B.S. in mechanical engineering at the University of Michigan in 1922, a M.S. in physics at the University of Pittsburgh in 1925, and a Ph.D. in engineering mechanics in June of this year from the University of Michigan. At the expiration of his present leave of absence at the University of Michigan, Dr. Frocht is returning to the Carnegie Institute of Technology with which he has been connected since 1922. From 1912 to 1920 he was associated with various automobile concerns, including the General Motors Research Laboratory at Dayton, Ohio. In 1930 he was on leave of absence from Carnegie at the Autostroy, a U.S.S.R. Corporation for whom he was commissioned to make a study of the functional and structural organization of the Ford Motor Co.

Presented at the National Applied Mechanics Meeting, Purdue University, Lafayette, Ind., June 15 and 16, 1931, of The American Society of Mechanical Engineers. (From a dissertation for the degree of Doctor of Philosophy in the University of Michigan.)

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

The Fringe Method

If instead of white light a monochromatic source be used, then the relative retardation between the ordinary and extraordinary rays caused by the stressed model manifests itself not in color effects but in a change of intensity. Referring now to the rays emerging from the analyzer, let A = amplitude, A_m = maximum amplitude, I = intensity, and I_m = maximum intensity.

It can then be shown[2] that $A = A_m \sin \pi R$ ..... [2]

Since the intensity of light varies as the square of the amplitude,

$$I = A^2 = A_m^2 \sin^2 \pi R$$

But $$A_m^2 = I_m$$

hence $$I = I_m \sin^2 \pi R \quad \text{..... [3]}$$

where R is given by Equation [1].

Inspection of Equation [3] shows that I will be zero for R equal to an integral number of wave lengths. For such values of R the light will be completely extinguished and a dark spot will be obtained on the screen. In this manner the use of monochromatic light results in alternate bright and black fringes, whose number and complexity depend on the material and loading.

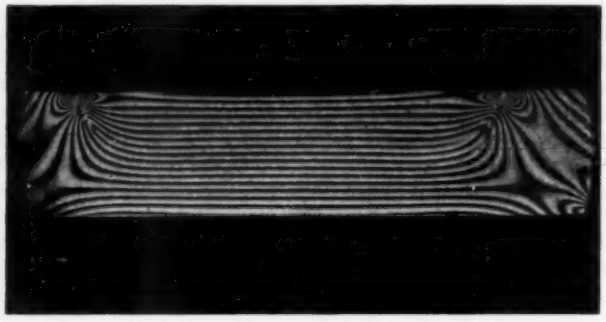


Fig. 1 Fringe Photograph of Bakelite Beam in Pure Bending

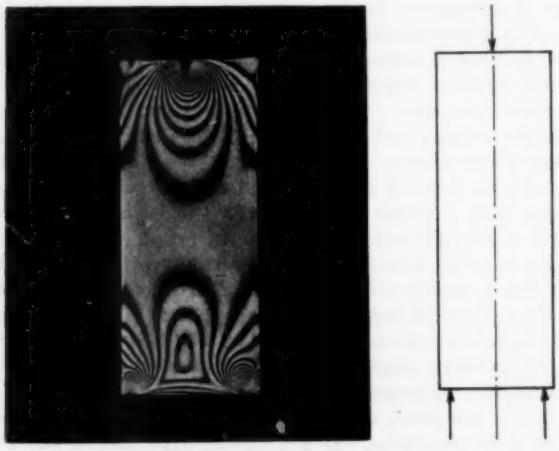


Fig. 2 Fringe Photograph of the Escaping Type

Fig. 1 shows a fringe photograph of a bakelite beam in pure bending. It will be observed that the fringes are straight, parallel, equidistant lines, this being due to the linear distribution of the stress. From the known load conditions and the dimensions of the beam, the stress value of each fringe can be computed.

[2] See article by A. L. Kimball, Jr., on "Stress Determination by Means of the Coker Photo-Elastic Method," *General Electric Review*, January, 1921, pp. 80–81.

The difference between the principal stresses corresponding to each fringe of a stressed model of the same material becomes thus a known quantity, provided we know the number of wave lengths of relative retardation responsible for the particular fringe, that is, provided we know the *fringe order*.

The Formation of Fringes

The formation of fringes is a very interesting, though somewhat complicated phenomenon. Dr. Tuzi classifies fringes into four types: the assembling, the converging, the diverging, and the escaping. The author proposes a simpler classification and will mention only two types of fringes: the escaping and the non-escaping. When all the fringes formed remain in the image, we have the non-escaping type. When some escape from the picture, we have the escaping type. The non-escaping type is best illustrated by pure bending, Fig. 1. The first fringes appear at the extreme fibers and approach the neutral axis as a limit but never reach it, since the stress at the neutral axis always remains zero. The fringes crowd together, but do not escape.

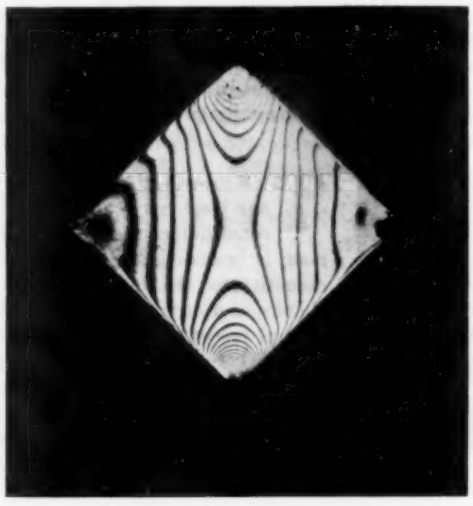


Fig. 3 Phenolite Model Showing Effect of Initial Stresses

The escaping type is the more general of the two. A simple illustration of this type is afforded by the fringes formed in a narrow, rectangular block in compression. Here the fringes are born in the interior, spread toward the edges, and escape, leaving only fragments at the top and bottom. Fringes may escape through the interior as well as the boundary. Fig. 2 shows a fringe photograph of the escaping type.

In the non-escaping type a static picture reveals the fringe order. This is not the case in the escaping type. Here the fringe formation must be studied in order to determine the fringe order. By slow and repeated loading and unloading one can determine the points where the fringes originate and vanish. It is also possible to take moving pictures and thus reproduce the fringe formation on the screen at leisure.

The fringe method for the determination of $(P - Q)$ is simpler than color matching, and more accurate than compensating. The fringes are clearer than isochromatics, and can be traced with accuracy. It eliminates the strain and tediousness that necessarily go with compensating. This method has another important merit. It not only reveals clearly the stresses in the model, but it makes it practicable to photograph them and thereby to secure permanent and convincing evidence of the stress distribution. From a pedagogical point of view the author

considers this of great importance. A picture carries a veracity which a sketch does not, and it is believed that with the help of the fringe method the dissemination of photoelastic knowledge should be materially accelerated.

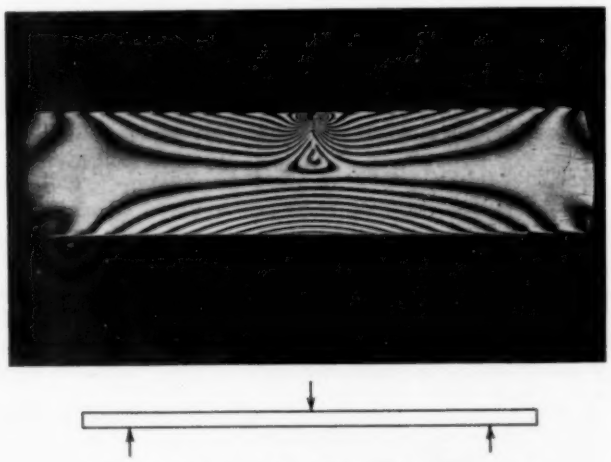


FIG. 4 BAKELITE BEAM UNDER CENTRAL LOAD

ANNEALING OF BAKELITE

The success of the fringe method depends upon having a sensitive material, which will show many fringes under comparatively small loads. Dr. Tuzi used phenolite, which is the invention of Professor Kita and Mr. Matui of the Institute of Physical and Chemical Research, Tokyo, Japan. This material is very sensitive and gives excellent results, but is not obtainable in this country.

Bakelite is a transparent material as sensitive as phenolite, and easily obtainable. Its use as a photoelastic material has been retarded because it shows pronounced initial stresses in its unstrained state. It is therefore of some interest that the author found a method which proved effective in the annealing of bakelite.[3] The main steps of this method are:

1 Heating the models to a temperature of about 70 deg. cent., and

[3] Prof. L. N. G. Filon in his paper "On the Graphical Determination of Stresses From Photo-Elastic Observations," *Engineering*, October 19, 1923, states that a method for the annealing of bakelite was discovered by Mr. Jessop. He does not, however, give an account of it, nor does he give any reference to a publication where such an account might be found.

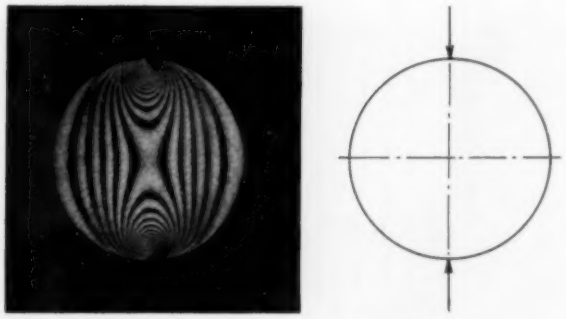


FIG. 7 PHENOLITE DISK

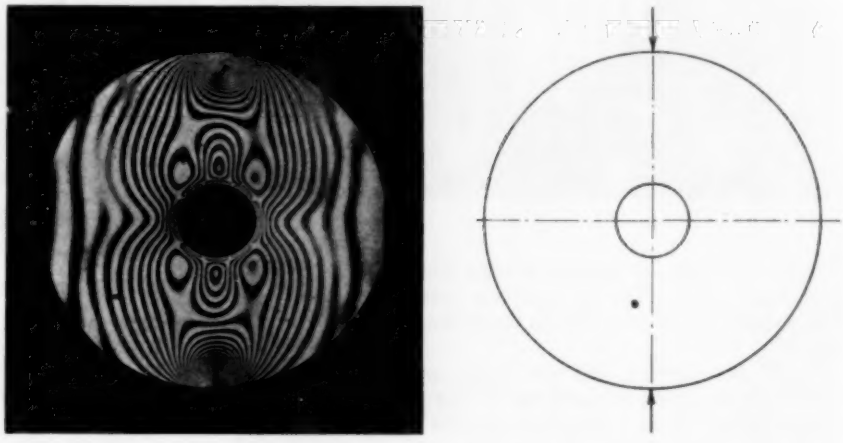
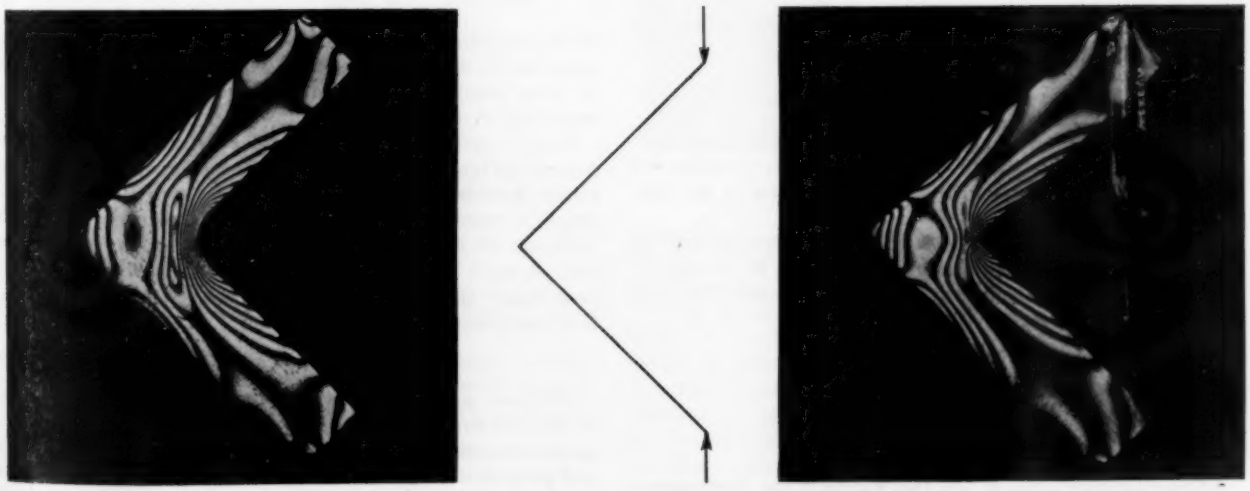


FIG. 8 BAKELITE MODEL



FIGS. 5 AND 6 BAKELITE MODELS SHOWING EFFECT OF FILLET ON STRESS CONCENTRATION, DIMENSIONS AND LOADS IDENTICAL

2 Cooling them very slowly to room temperature.

To procure uniform heating and cooling it is necessary to avoid direct contact between the models and the walls of the furnace. To this end it is advisable to place the models between glass plates and support them on a block.

The amount of time necessary for cooling depends on the furnace. In the author's case it took sixteen hours. Caution must be exercised not to remove the specimen too soon. Good models have been spoiled by premature removal. The heating can be done rapidly. It took the author only fifteen minutes

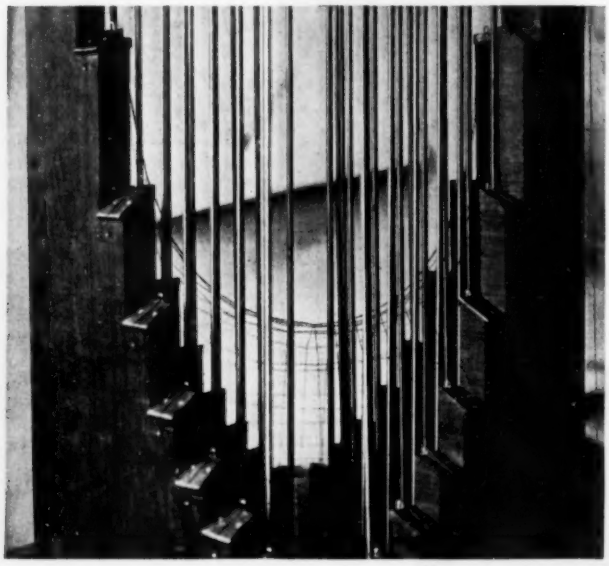


FIG. 9

to do it. In cases of stubborn stresses a second annealing will be found effective. The quality of the annealing obtained from this process can be judged from the photograph reproduced in Fig. 1.

It will be observed that the fringes are straight, parallel, equidistant lines, which would not be the case had there been initial stresses. Fig. 3 shows a phenolite model without annealing. It will be observed that there is marked absence of symmetry in the fringes, and also that they meet the boundary in directions almost parallel to them. An annealed model is shown in Fig. 16 in which the fringes are symmetrical and normal to the boundaries. Figs. 4–8 show fringe photographs of different models annealed by the aforementioned process.

A New Method for the Determination of $(P + Q)$

A method giving the same data as the lateral extensometers of Mesnager and Coker but radically different in principle and construction is being developed in the laboratories of the University of Michigan.

The sum of the principal stresses $(P + Q)$ in a two-dimensional system lends itself to representation by means of a surface.[4] This can be shown in the following way: The continuity equation of a two-dimensional system is

$$\nabla^4\phi = 0 \quad \ldots\ldots [4]$$

where ϕ represents the stress function.

This can be put in the modified form as follows:

$$\left(\frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2}\right) \times \left(\frac{\partial^2\phi}{\partial x^2} + \frac{\partial^2\phi}{\partial y^2}\right) = 0 \quad \ldots\ldots [5]$$

The second factor of this equation represents the sum of the normal component streases, i.e.,

$$\frac{\partial^2\phi}{\partial x^2} + \frac{\partial^2\phi}{\partial y^2} = \sigma_x + \sigma_y \quad \ldots\ldots [6]$$

which sum will be designated by the letter S.

The continuity equation then assumes the form:

$$\frac{\partial^2 S}{\partial x^2} + \frac{\partial^2 S}{\partial y^2} = 0 \quad \ldots\ldots [7]$$

and this is the equation of a membrane of small deflections whose boundary is subjected to uniform tension.

At a free boundary one of the principal stresses vanishes, so that the sum of the principal stresses on such boundaries equals the difference, and can therefore be determined photoelastically. If now a surface be constructed with boundary ordinates equal

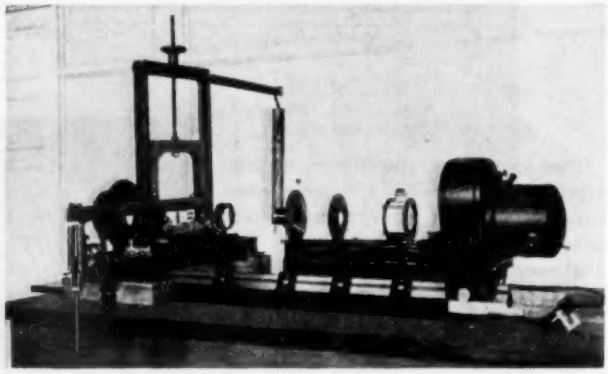


FIG. 10 APPARATUS USED IN PHOTOELASTIC INVESTIGATION

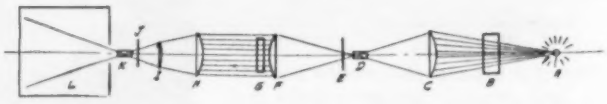


FIG. 11 ARRANGEMENT OF APPARATUS SHOWN IN FIG. 10

A—Light source
B—Water cooler
C—Lens; F = 100 mm.
D—Polarizer; 10-mm. Ahrens prism.
E—Quarter-plate
F—Lens; F = 150 mm.
G—Test model
H—Lens; F = 150 mm.
I—Lens
J—Quarter-plate
K—Analyzer; 10-mm. Ahrens prism
L—Camera or screen

to the boundary values of $(P + Q)$, then the ordinates at all other points would represent the sum of the principal stresses at those points. This affords an experimental means of determining $(P + Q)$.

Several methods can be used to construct such a surface. A soap film is one possibility; a membrane formed by a stretched rubber sheet is another. Fig. 9 shows a rough attempt to construct a woven surface. The uniform boundary tension is produced by means of small weights attached to the ends of each string. The shape of the curves thus obtained follows very closely the actual $(P + Q)$ curves. Work is in progress to obtain quantitative results.

The Main Problem

The main problem of which this paper is the outgrowth was to find the stress distribution in a square plate subjected to diagonal compression. The problem was investigated theoretically and photoelastically, and, as will be seen later, good agreement was obtained between the two sets of results. The photoelastic

[4] This suggestion is due to Mr. Den Hartog of the Westinghouse Electric and Manufacturing Company, East Pittsburgh, Pa.

investigation will be first presented. The apparatus used and its arrangement are shown in Figs. 10 and 11.

The optical solution of a problem consists of four steps:

1 Determination of isoclinic lines
2 Determination of stress trajectories
3 Evaluation of $(P - Q)$
4 Evaluation of P and Q separately.

Isoclinics. The locus of points whose principal stresses have the same directions is called an isoclinic line. The simplest and general method to obtain such lines is by removing the quarter-plates from the polariscope and to turn the crossed Nicols 5 or 10 deg. at a time. The image obtained contains the isoclinics for the particular setting of the Nicols. It is of interest that this ingenious method was proposed by Maxwell when he was only nineteen and still a student at Edinburgh.

Glass models, because of their transparency, give the clearest curves, and such models were used in the present investigation. The size of the model does not matter. The glass was $^5/_{16}$ in. thick and could sustain an average compressive stress of nearly one ton per square inch. To sustain that stress the load had to be uniformly distributed. This was attained by the use of soft copper pins which also prevented chipping. The original sketch of isoclinic curves was enlarged, and the final curves shown in Fig. 12 represent the averages of the enlargement.

The author wishes to call attention to a possible source of error in sketching isoclinics. Not all the black curves from plane polarized light are necessarily isoclinics. A point of zero stress or a point of equal principal stresses will also cause a dark spot. To eliminate the possibility of error, one should compare the pictures with and without quarter-plates. The effect of the quarter-plates, among other things, is to remove the isoclinics,

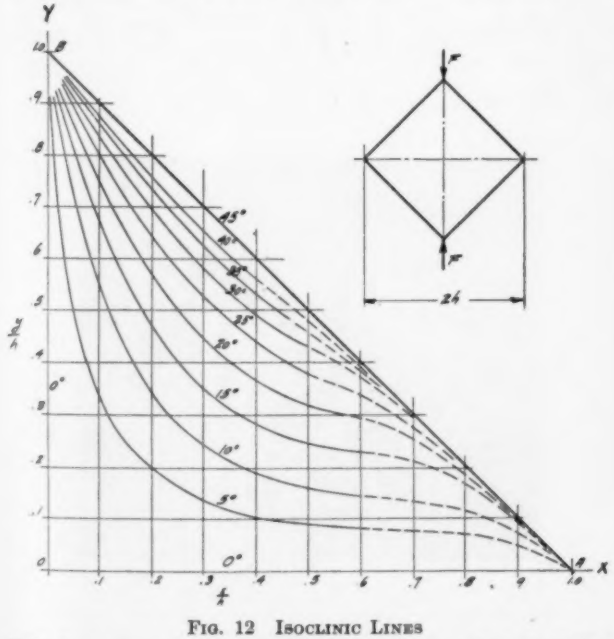


FIG. 12 ISOCLINIC LINES

(Solid lines: from direct observation; broken lines: from theoretical considerations.)

as can be seen from Figs. 13 and 14. The absence of the cross from Fig. 14 shows that it is an isoclinic, and likewise the presence of the circle in the same photograph shows that the circle is not an isoclinic.[5]

[5] Professor Muira includes the circle among the zero isoclinics, but the author believes this to be an error. See the former's "Spannungskurven in rechteckigen und keilförmigen Trägern," p. 87.

An interesting use was made of the cross representing the zero isoclinic in symmetrical models. During the investigation the Ahrens prisms became loose in their mountings and turned about the horizontal axis. They were reset by crossing the

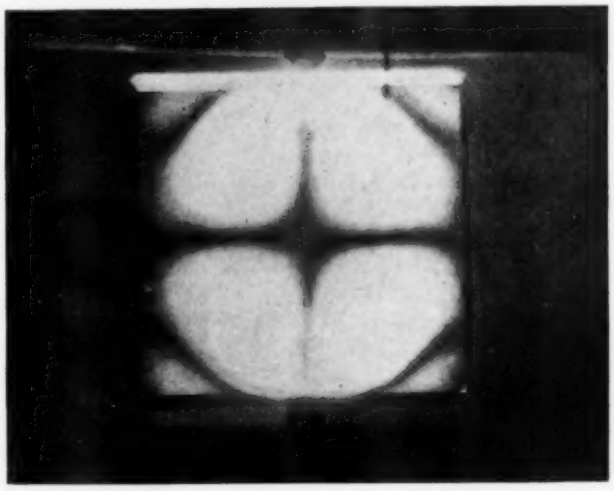


FIG. 13 WITHOUT QUARTER-PLATES

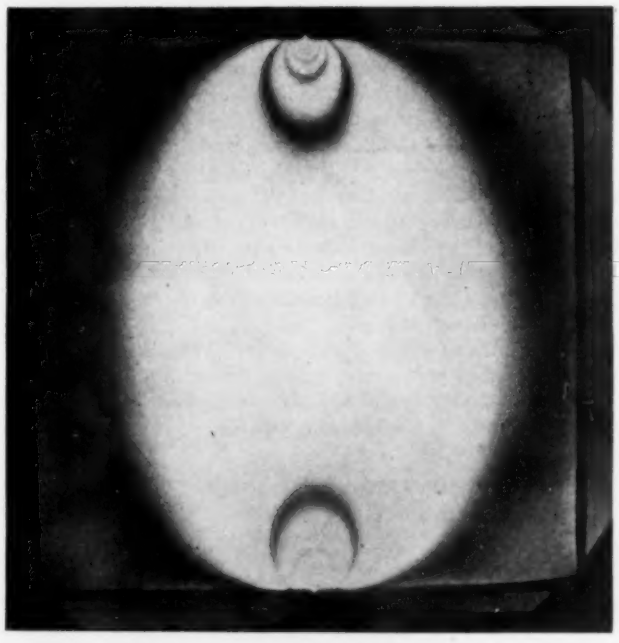


FIG. 14 WITH QUARTER-PLATES

mountings in a vertical and horizontal position, and adjusting the prisms until the isoclinic curve of the symmetrical model gave the cross.

Stress Trajectories. These were mapped in the usual manner and are shown by the full lines of Fig. 15.

Determination of $(P - Q)$. The difference between the principal stresses was determined in several ways. Celluloid, bakelite, and phenolite were used for materials, and color matching, tension compensating, and fringe counting as methods. But the author is so impressed with the simplicity and accuracy of the fringe method that the results will be presented in this form. First, two well-annealed models of the same strip of

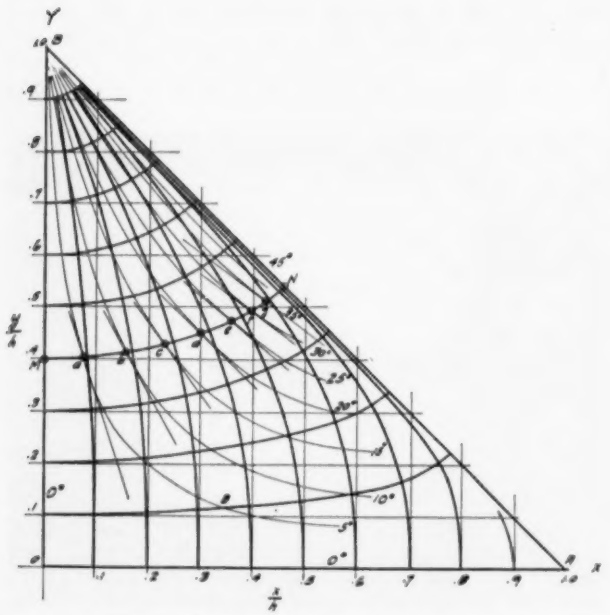


FIG. 15 STRESS TRAJECTORIES (*Heavy Lines*) AND ISOCLINICS (*Light Lines*)

bakelite were secured: a square plate and a beam, Figs. 16 and 17. The sodium lamp which was used for isoclinics was then replaced by the mercury-vapor lamp shown in Fig. 18, and the photographs shown in Figs. 16 and 17 taken. This gave all the optical data necessary to determine $(P - Q)$. As a measure of safety a free-hand sketch of the fringes was made immediately after taking the picture. This took but a few minutes and no optical creep could be observed. The clearness and thinness of the fringes enable one to sketch these with accuracy. The intersections of the fringes with the axes were noted, and curves were plotted with fringes as ordinates and axes as abscissas. Columns 2 in Tables 1 and 2 give the values of $(P - Q)$ in fringe order for the X- and Y-axes.

Determination of P and Q Separately. The principal stresses were determined separately by the use of Professor Filon's graphical-integration method. The equations used in this method are given in the following two forms. For their derivation see Appendix No. 1.

TABLE 1 COMPUTATIONS OF P AND Q ALONG OX

x/h (1)	$(P-Q)$ in fringe order (2)	$(P-Q)\Delta\phi$ $\Delta\phi = 0.0873$ radians (3)	Δy in inches (4)	$(P-Q)\frac{\Delta\phi}{\Delta y}$ (5)	Mean $(P-Q)\frac{\Delta\phi}{\Delta y}$ (6)	$(P-Q)\times\frac{\Delta\phi}{\Delta y}\Delta S_1$ (7)
0.0	4.17	0.3639	∞	0.0	0.120	0.060
0.1	4.00	0.3490	1.690	0.2065	0.268	0.134
0.2	3.58	0.3124	1.000	0.3124	0.364	0.182
0.3	3.00	0.2618	0.710	0.3687	0.382	0.191
0.4	2.25	0.1963	0.510	0.3849	0.344	0.172
0.5	1.47	0.1236	0.440	0.2809	0.260	0.130
0.6	0.88	0.0768	0.430	0.1786	0.132	0.066
0.7	0.45	0.0393	0.370	0.1062	0.064	0.032
0.8	0.20	0.0174	0.330	0.0527	0.028	0.014
0.9	0.05	0.0044	0.240	0.0183	0.007	0.003
1.0	0.0	0.0	0.0	0.0	.....	.....

x/h (1)	P (8)	Q (9)	P in lb. per sq. in., $f_1 = 283.88$ (10)	Q in lb. per sq. in., $f_1 = 283.88$ (11)	Mean Q in lb. per sq. in. (12)
0.0	0.9845	—3.1855	279.5	—904.3	—895
0.1	0.9245	—3.0755	262.5	—873.1	—837
0.2	0.7905	—2.7895	224.4	—791.9	—735
0.3	0.6085	—2.3915	172.7	—679.0	—596
0.4	0.4175	—1.8325	118.5	—520.2	—428
0.5	0.2455	—1.2245	69.6	—347.6	—277
0.6	0.1155	—0.7645	32.8	—217.0	—160
0.7	0.0495	—0.4005	14.05	—113.7	— 75
0.8	0.0175	—0.1825	4.97	— 51.8	— 25
0.9	0.0035	—0.0465	0.99	— 13.2	— 3
1.0	0.0	0.0	0.0	0.0	0

TABLE 2 COMPUTATIONS OF P AND Q ALONG OY

y/h (1)	$(P-Q)$ in fringe order (2)	$(P-Q)\Delta\phi$ $\Delta\phi = 0.08726$ radians (3)	Δx in inches (4)	$(P-Q)\frac{\Delta\phi}{\Delta x}$ (5)	Mean $(P-Q)\frac{\Delta\phi}{\Delta x}$ (6)	$(P-Q)\frac{\Delta\phi}{\Delta x}\cdot\Delta s_1$ (7)
0.0	4.17	0.3639	5.00	0.0728	0.100	0.050
0.10	4.20	0.3665	2.12	0.1728	0.240	0.120
0.20	4.30	0.3752	1.03	0.3642	0.465	0.232
0.30	4.48	0.3909	0.60	0.6515	0.800	0.400
0.40	4.79	0.4180	0.40	1.0450	1.400	0.700
0.50	5.31	0.4634	0.26	1.7820	2.350	1.175
0.60	6.20	0.5410	0.17	3.1820	4.400	2.200
0.70	7.75	0.6763	0.12	5.7800	6.950	1.740
0.75	9.30	0.8130	0.097	8.3500	11.250	2.810
0.76	9.70	0.8460	0.094	9.0000	8.700	0.435
0.77	10.40	0.9080	0.090	10.0300	9.500	0.475
0.78	11.50	1.0030	0.087	11.5400	10.800	0.540
0.79	12.50	1.0900	0.083	13.0600	12.300	0.615
0.80	13.50	1.1790	0.080	14.7500	13.900	0.695

y/h (1)	P in fringe order (8)	Q in fringe order (9)	P in lb. per sq. in., $f_1 = 283.88$ (10)	Q in lb. per sq. in., $f_1 = 283.88$ (11)	Mean P in lb. per sq. in. (12)
0.0	0.9845	— 3.1855	279.5	— 904.3	277
0.10	0.9645	— 3.2355	273.8	— 918.5	272
0.20	0.9445	— 3.3555	268.0	— 952.6	262
0.30	0.8920	— 3.5880	253.2	—1018.6	243
0.40	0.8020	— 3.9880	227.7	—1132.0	208
0.50	0.6220	— 4.6880	176.6	—1330.0	140
0.60	0.3370	— 5.8630	95.7	—1664.4	20
0.70	—0.3130	— 8.0630	— 88.8	—2289.0	—122
0.75	—0.5030	— 9.8030	—142.8	—2783.0	—100
0.76	—0.5650	—10.2650	—160.4	—2914.0	(Mean P between
0.77	—0.3400	—10.7400	— 96.5	—3050.0	0.75
0.78	0.2200	—11.2800	62.4	—3202.0	and
0.79	0.6050	—11.8950	171.7	—3377.0	0.80
0.80	0.9100	—12.5900	258.0	—3574.0	is 150)

$$1 \quad \begin{cases} (a) \quad P = P_0 - \Delta\phi \int (P - Q)\,\frac{ds_1}{\Delta y} \ldots\ldots\ldots\ldots [8a] \\ (b) \quad P = P_0 + \int (P - Q)\cot\psi d\phi \ldots\ldots\ldots\ldots [8b] \end{cases}$$

$$2 \quad \begin{cases} (a) \quad Q = Q_0 - \Delta\phi \int (P - Q)\,\frac{ds_2}{\Delta x} \ldots\ldots\ldots\ldots [9a] \\ (b) \quad Q = Q_0 + \int (Q - P)\cot\psi_1 d\phi \ldots\ldots\ldots\ldots [9b] \end{cases}$$

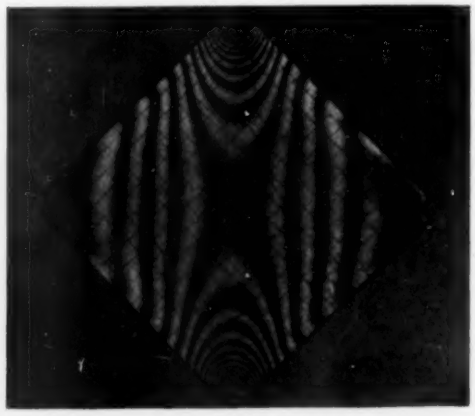


FIG. 16 ANNEALED SQUARE PLATE OF BAKELITE

The principal stresses were first determined along the diagonals, and for these lines Equations [8*a*] and [9*a*] are suitable. The notation is as follows: Referring to Fig. 19, Δx and Δy are the coordinates of any point C on the first curved isoclinic; $\Delta\phi$ is the difference between the parameters of the zero isoclinic and the first curved isoclinic. In the present case $\Delta\phi$ is 5 deg.

In the foregoing equations the directions of s_1 and s_2 are interrelated, s_2 being obtained from s_1 by a counterclockwise rotation of 90 deg.

The steps involved in the determination of P and Q will be best understood by a study of Tables 1 and 2. The mean values given in columns 6 and 12, Table 1, were taken from curves.

P was calculated first. The starting point was taken at A where P_0 is zero, Fig. 15, and integrated toward the origin 0. For s_1 positive to the left, s_2 is positive downward. It follows that Δy in Equation [8a] is negative for the 5-deg. isoclinic. In the calculations Δs_1 was taken as 0.5 in., as the author was working from an image with a diagonal of 10 in. divided into twenty equal intervals. Having determined P and knowing $(P - Q)$ at the same points, it was a simple matter to find Q.

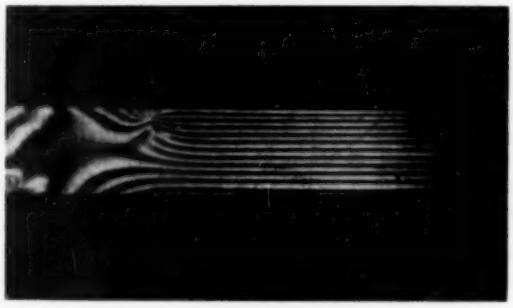


FIG. 17 ANNEALED BEAM OF BAKELITE

The stresses along the Y-axis were determined in a similar manner. The starting point was taken at 0, at which point both principal stresses were now known. Using Equation [9a] and starting with Q_o, values of Q were determined up to $0.82h$, which represents the limit of the experimental results. All the steps and the final results are given in Table 2.

The results thus far obtained are in terms of fringes. It is, however, a simple matter to convert these into pounds per square inch. The photograph of the comparison beam, Fig. 17, shows six fringes corresponding to a bending moment of 10.125 in-lb. For a loading as shown in Fig. 20, the ordinary beam equation $S = Mv/I$ is applicable and gives $S_{max.} = 1703.28$ lb. per sq. in., or 283.88 lb. per sq. in., for fringe No. 1.

By multiplying the fringe value of the principal stresses by 283.88 we obtain these stresses in pounds per square inch.

Verification of Results. The external load on the model was

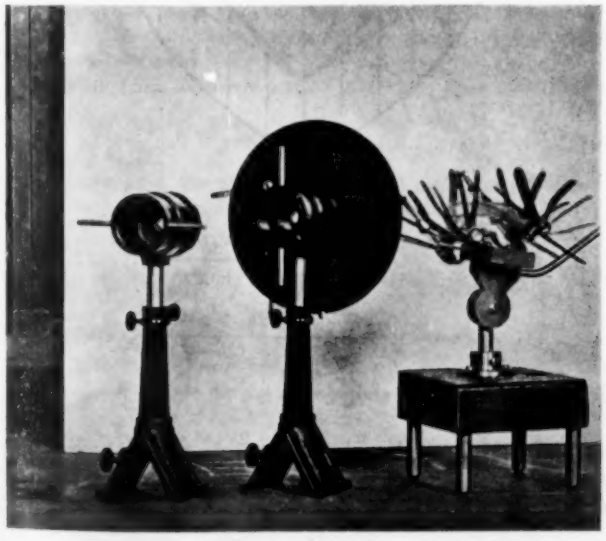


FIG. 18 SHOWING MERCURY-VAPOR LAMP AND CROSSED AHRENS PRISMS

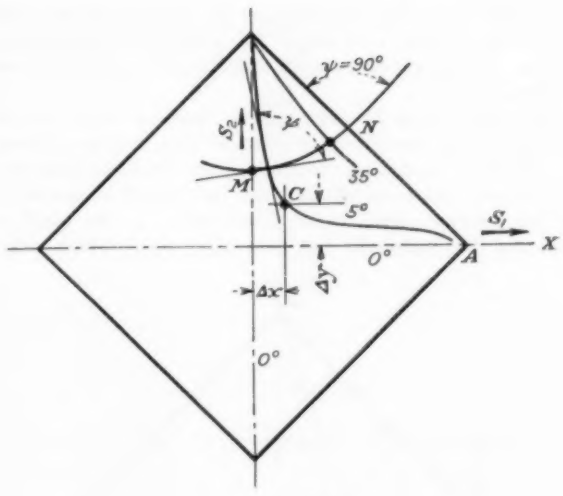


FIG. 19

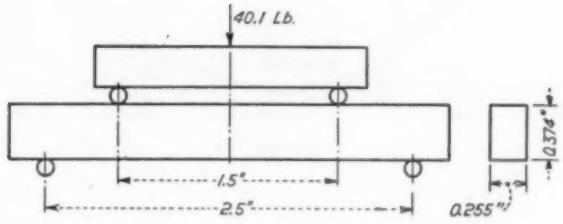


FIG. 20

179.66 lb. If, therefore, a section be made through the horizontal diagonal and one half of the block be considered as a free body, a static check of the work can be obtained. For equilibrium

$$\Sigma Y = 0$$

or

$$179.66 = \int_{-h}^{h} Qtdx \qquad [10]$$

where $2h$ is the diagonal and t the thickness of the model. For $2h = 1.855$, $t = 0.255$, and the mean values of Q as given in Table 1 the right side of Equation [10] works out to be 191 lb., which gives a check within 5.7 per cent. A portion of the discrepancy is probably due to the small initial stresses at the edges of the comparison beam.

DETERMINATION OF PRINCIPAL STRESSES ON A GENERAL STRESS TRAJECTORY

To show the generality of the photoelastic method, the principal stresses were also computed along the stress trajectory MN, Fig. 15. An enlargement of the photograph shown in Fig. 16 was made and the stress trajectory MN superimposed upon it. The intersections of the fringes with this trajectory were noted and a curve plotted giving $(P - Q)$ as a function of MN. The intersections of the isoclinics with MN are marked by the small letters a, b, c, d, e, f, and g, Fig. 15.

To determine P and Q separately, Equation [8b] was used. $\Delta\phi$ in this case is the difference between the parameters of two successive isoclinics. Thus in going from N to g, $\Delta\phi$ is -10 deg., Fig. 15.

ψ for any point on the stress trajectory is the angle between the tangents to the stress trajectory and the intersecting isoclinic measured counterclockwise. These were determined from Fig. 15.

The stress trajectory was followed in the direction of NM, starting from N where P_o is zero. This made $\Delta\phi$ negative and P positive. The details of the work and the final results are given in Table 3.

Check From Stress Trajectory MN. Another check of the correctness of the work from a point of view entirely different from statics is afforded by the results from the stress trajectory MN, Fig. 15. At point M there are two sets of values of the principal stresses. One set of these values is the result of integrating first along $A0$ and then along $0M$. These values

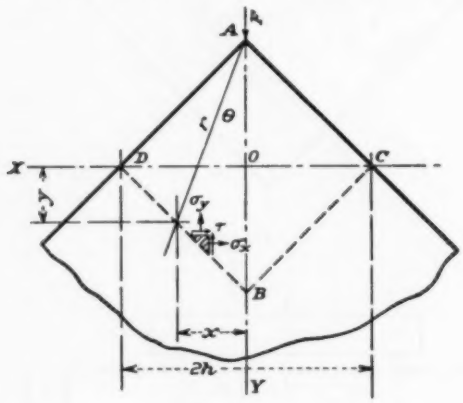


FIG. 21

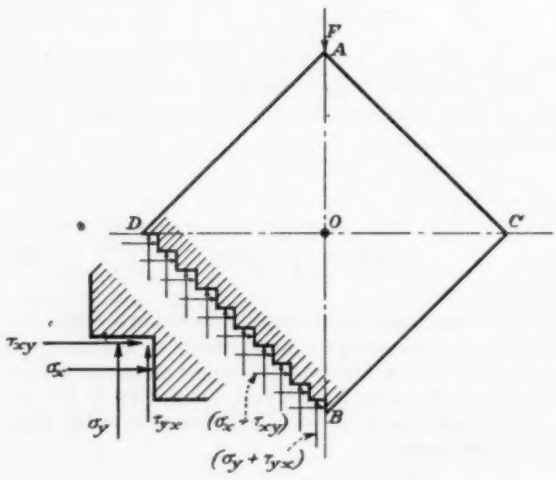


FIG. 22

in fringe order are: $P = 0.8020$, and $Q = -3.988$, see Table 2. The other set of values is the result of integrating along the path NM, which gave $P = 0.824$, and $Q = -3.976$ in fringe order, see Table 3.

The P's agree within 2.8 per cent and the Q's within 0.5 per cent. This agreement between the values of the principal stresses at a point obtained from different paths of integration constitutes the best proof of the correctness of the work.

SUMMARY OF APPROXIMATE THEORETICAL SOLUTION OF PROBLEM

The problem discussed is approached from the point of view of an infinite wedge, and the author assumes the stress function:[6]

[6] It is shown in texts on elasticity that this function satisfies the boundary conditions and the continuity equation.

TABLE 3 COMPUTATIONS OF P AND Q ALONG STRESS TRAJECTORY MN

$P = P_0 + \int_{\phi_0}^{\phi} (P - Q) \cot\psi d\phi$. In going from N to M, $\Delta\phi$ is negative, and $P_0 = 0$. $\therefore P = \int_{\phi_0}^{\phi} (P - Q) \cot\psi d\phi$, a positive number

Deg.	$(P-Q)$	Cotψ	$(P-Q)$ × cotψ	Mean $(P-Q)\cot\psi$	$(P-Q)$ cotψ. $\Delta\phi$	P in fringe order	Q in fringe order
0	4.80	0	0	—0.69	—0.0602	0.8240	—3.976
5	4.68	—0.231	—1.080	—1.270	—0.1110	0.7638	—3.916
10	4.35	—0.325	—1.415	—1.510	—0.1320	0.6528	—3.697
15	3.82	—0.404	—1.540	—1.590	—0.1390	0.5208	—2.309
20	3.15	—0.504	—1.580	—1.480	—0.1290	0.3818	—2.768
25	2.70	—0.466	—1.260	—1.200	—0.1050	0.2528	—2.450
30	2.42	—0.425	—1.030	—0.940	—0.0820	0.1478	—2.270
35	2.25	—0.335	—0.755	—0.400	—0.0658	0.0658	—2.180
45	2.00	0.0	0	...	...	0.0	—2.000

NOTE: $\Delta\phi = 5° = 0.0873$ radian everywhere except at 45°, where $\Delta\phi = 10°$ or 0.1646 rad.

$$\phi = Ar\,\theta \sin\theta \quad [11]$$

If from the infinite wedge we cut a square plate of a diagonal $2h$ as shown in Fig. 21, then the sides BD and BC would be subjected to stress components which in terms of cartesian coordinates and boundary conditions reduce to:

$$\sigma_x = -\frac{F}{(\pi+2)} \frac{(y+h)(h-y)^2}{(h^2+y^2)^2} \quad [12]$$

$$\sigma_y = -\frac{4F}{(\pi+2)} \frac{(2h-x)^3}{[x^2+(2h-x)^2]^2} \quad [13]$$

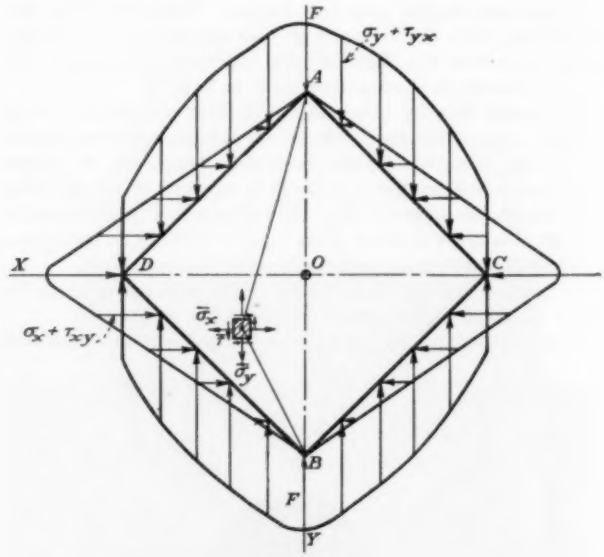


FIG. 23

$$\tau = -\frac{F}{(\pi+2)} \frac{(y+h)^2(h-y)}{(h^2+y^2)^2} \quad [14]$$

$$= -\frac{4F}{(\pi+2)} \frac{(2h-x)^2 x}{[x^2+(2h-x)^2]^2} \quad [15]$$

Combining σ_x with τ_{xy}, and σ_y with τ_{yx}, we obtain

$$\sigma_x + \tau_{xy} = -\frac{F}{(\pi+2)} \frac{2h(h^2-y^2)}{(h^2+y^2)^2} \quad [16]$$

and

$$\sigma_y + \tau_{yx} = -\frac{4F}{(\pi+2)} \frac{(2h-x)^2\,2h}{[x^2+(2h-x)^2]^2} \quad [17]$$

Upon plotting curves for [16] and [17] we find that good approximations can be obtained with simpler and symmetrical functions: namely,

$$\sigma_x + \tau_{xy} = -0.195\left(1 + \cos\frac{\pi x}{h}\right)\frac{F}{h} \quad [18]$$

$$\sigma_y + \tau_{yx} = -\left(0.564 - 0.1945\frac{x^2}{h^2} - 0.1031\cos\frac{\pi x}{h} - 0.0564\cos\frac{2\pi x}{h} - 0.01556\cos\frac{4\pi x}{h}\right)\frac{F}{h} \quad [19]$$

Returning now to the infinite wedge, a square plate *ABCD* is cut from it whose free-body diagram is shown in Fig. 22. Suppose now that to the system of forces in Fig. 22 there is added a symmetrical system and boundary forces are obtained as shown in Fig. 23. The stress components for the combined system are:

$$\bar{\sigma}_x = -\frac{4Fx^2}{(\pi+2)}\left\{\frac{(h+y)}{[x^2+(h+y)^2]^2} + \frac{(h-y)}{[x^2+(h-y)^2]^2}\right\} \quad [20]$$

$$\bar{\sigma}_y = -\frac{4F}{(\pi+2)}\left\{\frac{(h+y)^3}{[x^2+(h+y)^2]^2} + \frac{(h-y)^3}{[x^2+(h-y)^2]^2}\right\} \quad [21]$$

and

$$\bar{\tau}_{xy} = \bar{\tau}_{yx} = \frac{4Fx}{(\pi+2)}\left\{\frac{(h-y)^2}{[x^2+(h-y)^2]^2} - \frac{(h+y)^2}{[x^2+(h+y)^2]^2}\right\} \quad [22]$$

If to $\bar{\sigma}_x$, $\bar{\sigma}_y$, and $\bar{\tau}$ are added the stress components due to a system of boundary forces equal and opposite to those shown in Fig. 23 the required stress distribution will be obtained, that is, the stress system in a square plate subjected to diagonal compression. This will be done by the use of Professor Timoshenko's Approximate Strain-Energy Method.[7]

The required stress function is assumed in the form of a series, thus

$$\Phi = \Phi_0 + \alpha_1\phi_1 + \alpha_2\phi_2 + \ldots\ldots\ldots\alpha_n\phi_n \quad [23]$$

where Φ_0 is a function so chosen that its corresponding stress system satisfies the boundary conditions, ϕ_1, ϕ_2.....ϕ_n are functions whose corresponding stress systems vanish on the boundary, and $\alpha_1, \alpha_2 \ldots\ldots \alpha_n$ are constant coefficients determined by the conditions that the strain energy at equilibrium must be a minimum.

We shall limit ourselves to the first two terms of the series [23], i.e.,

$$\Phi = \Phi_0 + \alpha_1\phi_1 \quad [24]$$

The expression for the strain-energy V, assuming Poisson's ratio $\mu = 0$, and the modulus of elasticity equal E is

$$V = \frac{1}{2E}\int\int(\sigma_x^2 + \sigma_y^2 + 2\tau_{xy}^2) \times dxdy \quad [25]$$

and for equilibrium

$$\frac{\partial V}{\partial\alpha_1} = 0 = \int\int\left(\sigma_x\frac{\partial\sigma_x}{\partial\alpha_1} + \sigma_y\frac{\partial\sigma_y}{\partial\alpha_1} + \frac{2\tau\partial\tau_{xy}}{\partial\alpha_1}\right)dxdy \quad [26]$$

It is proved in the theory of elasticity that in terms of a stress function F the stress components are:

$$\sigma_x = \frac{\partial^2 F}{\partial y^2}; \qquad \sigma_y = \frac{\partial^2 F}{\partial x^2} \quad \text{and} \quad \tau = -\frac{\partial^2 F}{\partial x\partial y}$$

[7] See his article in *Phil. Mag.*, vol. XLVII, June, 1924.

It follows that the stress components in terms of our approximate stress function Φ are:

$$\sigma_x = \frac{\partial^2\Phi_0}{\partial y^2} + \alpha_1\frac{\partial^2\phi_1}{\partial y^2} \quad [27]$$

$$\sigma_y = \frac{\partial\Phi_0}{\partial x^2} + \alpha_1\frac{\partial^2\phi_1}{\partial x^2} \quad [28]$$

$$\tau = -\left(\frac{\partial^2\Phi_0}{\partial x\partial y} + \alpha_1\frac{\partial^2\phi_1}{\partial x\partial y}\right) \quad [29]$$

It will be found that

$$\phi_1 = \frac{1}{16}[(x+y)^2 - h^2]^2[(y-x)^2 - h^2]^2 \quad [30]$$

gives a stress system which vanishes on the given boundaries whose equations are

$$x + y = \pm h \quad [31]$$

$$y - x = \pm h \quad [32]$$

and therefore satisfies the conditions imposed upon it.

Upon substitution of Equations [27], [28], and [29] and their respective derivatives with respect to α_1 in Equation [26] and upon collecting terms in α_1 we get

$$\int\int\left\{\frac{\partial^2\Phi_0}{\partial y^2}\times\frac{\partial^3\phi_1}{\partial y^3} + \frac{\partial^2\Phi_0}{\partial x^2}\times\frac{\partial^3\phi_1}{\partial x^3} + 2\frac{\partial^2\Phi_0}{\partial x\partial y}\frac{\partial^3\Phi_1}{\partial x\partial y} + \alpha_1\left[\left(\frac{\partial^2\phi_1}{\partial y^2}\right)^2 + \left(\frac{\partial^2\phi_1}{\partial x^2}\right)^2 + 2\left(\frac{\partial^2\phi_1}{\partial x\partial y}\right)^2\right]\right\}dxdy = 0 \quad [33]$$

Letting X and Y stand for $-(\sigma_x + \tau_{xy})$ and $-(\sigma_y + \tau_{yx})$, respectively, we remove first the boundary forces defined by Y. In this case Φ_0 is a function of X only. It will also be found that within the given limits of integration

$$\int_0^h\int_0^{h-y}\frac{\partial^2\phi_1}{\partial x^2}dxdy = \int_0^h\int_0^{h-y}\frac{\partial^2\phi_1}{\partial y^2}dxdy \quad [34]$$

Taking these into consideration, Equation [33] reduces to

$$2\alpha_1\int_0^h\int_0^{h-y}\left[\left(\frac{\partial^2\phi_1}{\partial x^2}\right)^2 + \left(\frac{\partial^2\phi_1}{\partial x\partial y}\right)^2\right]\times dxdy = -\int_0^h\int_0^{h-y}\frac{\partial^2\Phi_0}{\partial x^2}\frac{\partial^2\phi_1}{\partial x^2}dxdy \quad [35]$$

It is from this equation that we determine α_1. The left side of Equation [35] contains about eighty integrals of the type

$$\int\int x^m y^n dxdy$$

These have been reduced to only sixteen by observing that within the given limits of integration

$$\int\int x^m y^n dxdy = \int\int x^n y^m dxdy \quad [36]$$

With this simplification

$$2\int_0^h\int_0^{h-y}\left[\left(\frac{\partial^2\phi_1}{\partial x^2}\right)^2 + \left(\frac{\partial^2\phi_1}{\partial x\partial y}\right)^2\right]dxdy = \int_0^h\int_0^{h-y}$$

$$(25x^{12} - 78x^{10}y^2 - 106h^2x^{10} + 222h^2x^8y^2 + 87x^8y^4 + 175h^4x^8 - 116h^2x^6y^4 - 220h^4x^6y^2 - 34x^6y^6 - 140h^6x^6 + 109h^4x^4y^4 + 55h^2x^4 + 76h^6x^4y^2 + 5h^2x^2y^2 - 10h^{10}x^2 + 0.5h^{12})dxdy = 0.1044997h^{14}. \quad [37]$$

The right side of Equation [35] is

$$-\int_0^h\int_0^{h-y}\frac{\partial^2\Phi_0}{\partial x^2}\frac{\partial^2\phi_1}{\partial x^2}\,dxdy = -\int_0^h\int_0^{h-y}(3.5x^6+4.5x^2y^4 + 4.5h^4x^2 - 7.5x^4y^2 - 7.5h^2x^4 - 0.5y^6 + 3h^2x^2y^2 + 0.5h^2y^4 + 0.5h^4y^2 - 0.5h^6) \times \left(0.564 - 0.1945\frac{x^2}{h^2} - 0.1031\cos\frac{\pi x}{h} - 0.0564\cos\frac{2\pi x}{h} - 0.01556\cos\frac{4\pi x}{h}\right)\frac{F}{h}\,dxdy = -0.0070967\,Fh^7. \quad [38]$$

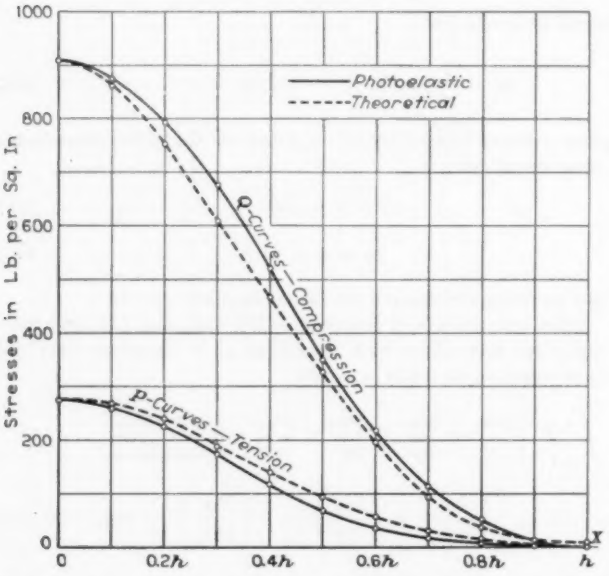


FIG. 24 COMPARATIVE CURVES OF THEORETICAL AND PHOTOELASTIC VALUES OF P AND Q ALONG OX

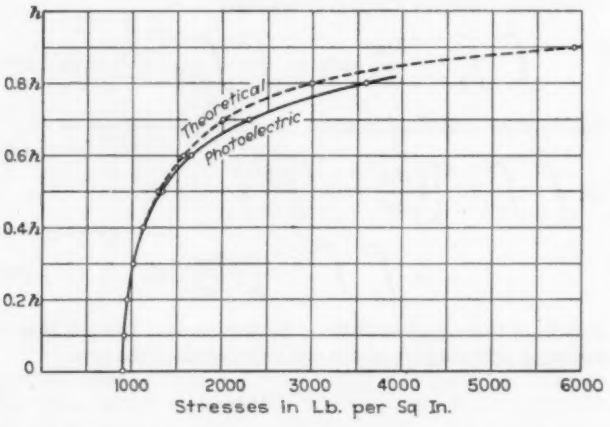


FIG. 25 COMPARATIVE CURVES OF THEORETICAL AND PHOTOELASTIC VALUES OF P ALONG OY

Solving for α_1

$$\alpha_1 = -\frac{0.0070967\,Fh^7}{0.1044997\,h^{14}} = -0.06788\,\frac{F}{h^7} \quad [39]$$

The approximate stress function for the Y boundary forces thus becomes

$$\Phi = \Phi_0 - 0.06788\,\frac{F}{h^7} \times \frac{1}{16}\,[(x+y)^2 - h^2]^2 \times [(y-x)^2 - h^2]^2 \quad [40]$$

In precisely the same way the approximate stress function for the X boundary forces is found to be

$$\psi = \psi_0 + 0.114249\,\frac{F}{h^7} \times \frac{1}{16}\,[(x+y)^2 - h^2]^2 \times [(y-x)^2 - h^2]^2 \quad [41]$$

where ψ and ψ_0 replace Φ and Φ_0, respectively. Upon combining the two stress systems Φ and ψ and designating the resultant stress components by P_0, Q_0, and S_0, we obtain

$$P_0 = \frac{\partial^2\psi_0}{\partial y^2} + 0.04637\,\frac{F}{h^7}\frac{\partial\phi_1}{\partial y^2} \quad [42]$$

and

$$Q_0 = \frac{\partial^2\Phi_0}{\partial x^2} + 0.04637\,\frac{F}{h^7}\frac{\partial\phi_1}{\partial x^2} \quad [43]$$

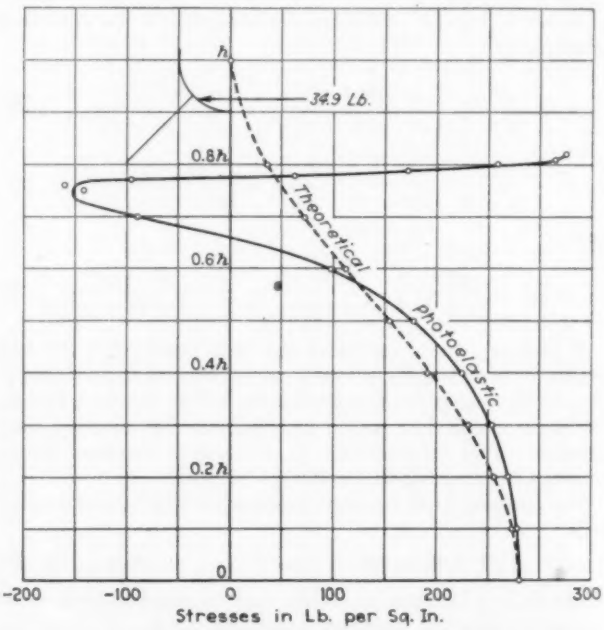


FIG. 26 COMPARATIVE CURVES OF THEORETICAL AND PHOTOELASTIC VALUES OF Q ALONG OY

Denoting the resultant stress components in the square block by P, Q, and S, respectively, we obtain

$$P = P_0 + \bar{\sigma}_x \quad [44]$$

$$Q = Q_0 + \bar{\sigma}_y \quad [45]$$

Table 4 gives the values of P and Q for the axes of symmetry; $r = x/h$ or y/h.

TABLE 4 GIVING P AND Q ALONG X- AND Y-AXES

	X-axis		Y-axis	
r	P in $\frac{F}{h}$	Q in $\frac{F}{h}$	P in $\frac{F}{h}$	Q in $\frac{F}{h}$
0.0	0.3668	—1.1900	0.3668	—1.1900
0.1	0.3520	—1.1300	0.3593	—1.2040
0.2	0.3100	—0.9850	0.3379	—1.2530
0.3	0.2510	—0.8000	0.3025	—1.3420
0.4	0.1860	—0.6130	0.2573	—1.4800
0.5	0.1250	—0.4300	0.2048	—1.7000
0.6	0.0741	—0.2600	0.1492	—2.0540
0.7	0.0380	—0.1270	0.0950	—2.6700
0.8	0.0151	—0.0498	0.0476	—3.9360
0.9	0.0045	—0.0170	0.0134	—7.7930
1.0	0.0010	0.0116	0.0	∞

DISCUSSION OF RESULTS

The extent to which the theoretical results substantiate the

experimental ones is graphically represented by the curves of Figs. 24–26, inclusive. The only marked divergence is shown by the curve of Fig. 26 beyond $0.6h$. The explanation for this discrepancy is found in the approximate character of the stress functions Φ and ψ. It is felt that if more terms of the series [23] were taken this divergence would be minimized. All the other curves substantiate the photoelastic results. At the center and the extremities of the horizontal diagonal the agreement is almost perfect. It is therefore considered that the results from the photoelastic investigation represent a correct picture of the stress distribution in the square block.

Referring now to this stress distribution it is observed first of all that fairly large tensile stresses are produced as a result of the diagonal compression. From an engineering point of view this is perhaps the most interesting aspect of the problem. The large tensile stress could hardly have been predicted on the basis of the ordinary theories of mechanics. At the center the maximum tension is 278.6 lb. per sq. in., or nearly one-third of the corresponding compressive stress. The maximum value of the tension, however, is at least 350 lb. per sq. in., this being the value obtained on the Y-axis at $0.82h$. Square blocks weak in tension and subjected to diagonal compression should therefore show signs of failure at points on the loaded diagonal about $h/5$ from the corners.

It may also be worth noting that the P-stresses are all tensile along what might be called the horizontal stress trajectories, but are of variable signs on the vertical stress trajectories. Thus on the Y-axis the P is positive from 0 to $0.655h$, negative from this point to $0.775h$, and again positive from there on to at least $0.82h$. No computations were made beyond this point due to the inaccuracies in the measurements of the very small values of Δx. It is of interest that $(P - Q)$ in fringe order could be distinctly read even at $0.9h$.

The P-curve cannot rise much beyond the value of 350 lb. per sq. in. without having to make a sudden return and cross the axis again to assume compressive values. Assuming that the P-curve stays positive to $0.9h$, and that the stress distribution in the tip beyond $0.9h$ is essentially that of an infinite wedge, we get a static check on the loaded diagonal from horizontal summation of the forces, first, however, cutting out the loaded tips up to a radius of $0.1h$, and replacing them by the horizontal thrusts which they exert, and which for our loads were found to be 34.9 lb.

$$\Sigma X = 0, \text{ or } 34.9 = \int_0^h Ptdy \ldots\ldots\ldots [46]$$

Taking $2h$ again as 1.855 in., t as 0.255 in., and the mean values of P from Table 2, we get for the right side of Equation [46] 33.96 lb., or a static check within 2.7 per cent.

Photoelastic Equipment

The main equipment of a photoelastic laboratory consists of

1 A polariscope
2 A compensator
3 A loading frame
4 A camera
5 A monochromatic lamp
6 An electric furnace, and
7 A special table for drawing isoclinics.

Fig. 10 shows a photograph of the polariscope used in the Photoelastic Laboratory of the Mechanics Department at the University of Michigan. It is imported from the Winkel-Zeiss firm at Göttingen. It uses Ahren prisms instead of Nicols, and gives satisfactory results.

Fig. 27 shows a photograph of a compensator developed at the laboratory. The carriage of the compensator was designed by Dr. Donnell, formerly of the Mechanics Department of the University of Michigan. It is capable of a vertical, horizontal, and rotary motion. The spring, the beam compensator (Fig. 28), the setting-up fixture, and other features were designed by the author. Considerable difficulty was encountered in the design of a spring which would be free from energy losses. The adopted design gives a linear relation between load and deflection, and is the same during loading and unloading. In the fringe method the compensator is not used to determine $(P - Q)$. It is used, however, to determine the sign of the stress.

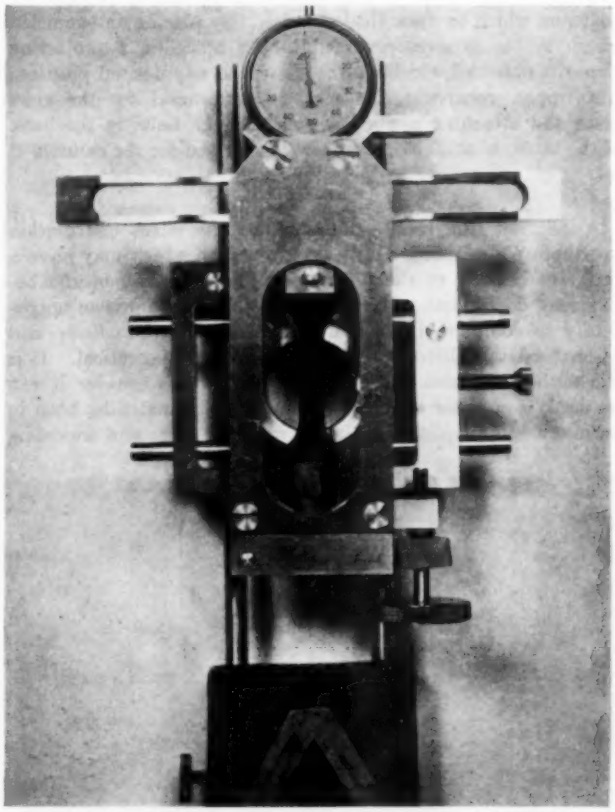


Fig. 27 Tension Compensator

It also makes an excellent tension machine, and the one shown could produce a tension of 150 lb., the accuracy of the reading being $^1/_4$ lb. The spring constant is 1615 lb.

The Loading Frame. The frame shown in Fig. 29 can produce tension, compression, and bending. The central frame is capable of a vertical and a horizontal motion. It can accommodate many sizes of models and has means to produce perfect alignment of the supporting pins. When producing bending provision is made for self-centering, which automatically gives equal moment arms and loads. By means of a small eccentric operated by a wooden handle visible in the picture, the load can be quickly removed from the model. The springs reduce impact forces and absorb vibrations. The main lever rests against a knife edge.

Camera. An ordinary Eastman 5 × 7-in. camera was used. The lens was removed, the front of the camera opened, and the image projected directly upon the film. To retain the use of the shutter it was mounted on a special pedestal and placed close to the analyzer. In selecting a camera for the fringe

method special attention should be paid to the provisions for adjusting the focal plane, as it is of utmost importance to secure good focusing of every part of the image.

Lamp. For the fringe method a monochromatic lamp must be had. The one used was a mercury-vapor lamp, shown in Fig. 18. With the aid of a No. 77 Wratten filter, the yellow rays were stopped and a fairly pure green light obtained. Orthochromatic Eastman films were found suitable for the author's purpose.

Furnace and Isoclinic Table. The annealing work was done in a small electric furnace working at $12^1/_2$ amperes and 220 volts. Fig. 30 shows a three-in-one piece of equipment. The main function of this table was to provide a vertical glass drawing board on which to draw the isoclinics. By placing an enameled screen in the grooves provided in the uprights, a projecting screen is obtained which can be moved to any desired position. Also, upon removing the special frame used for the glass plate and attaching a wooden shelf through bolts in the back of the table, a satisfactory support is obtained for the camera.

Conclusion

Aside from the results obtained for the stress distribution in square plates under diagonal compression which may have a practical bearing in the design and understanding of the behavior of double knife edges, this investigation affords an opportunity to compare the several methods in photoelasticity and to contrast the photoelastic method with the theoretical. It is the author's opinion that the fringe method, whenever it can be used, is superior to compensating or color matching both in accuracy and simplicity, and that with the successful annealing of bakelite a suitable material is made available for this method.

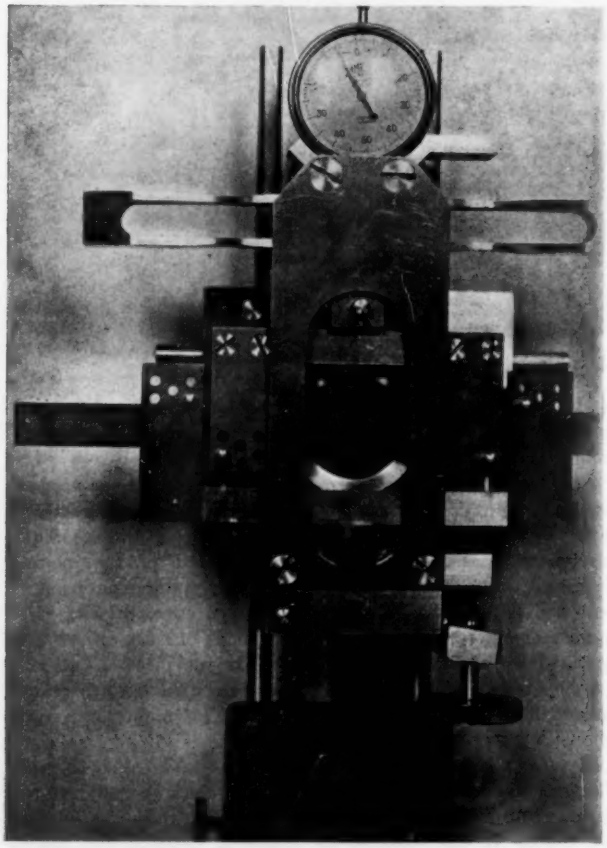


Fig. 28 Beam Compensator

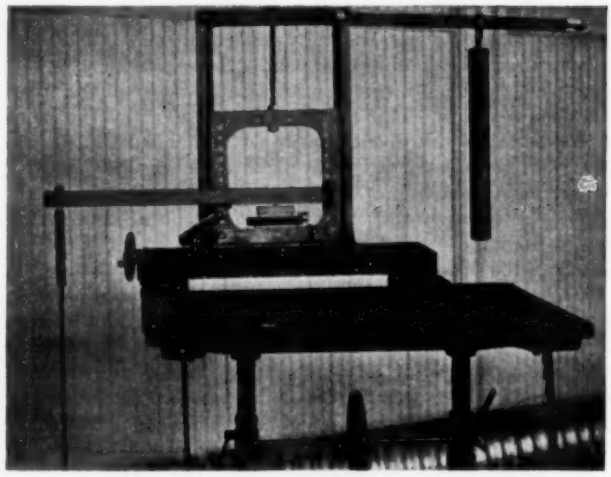


Fig. 29 Loading Frame Showing Arrangement for Bending (See Fig. 10 for compression.)

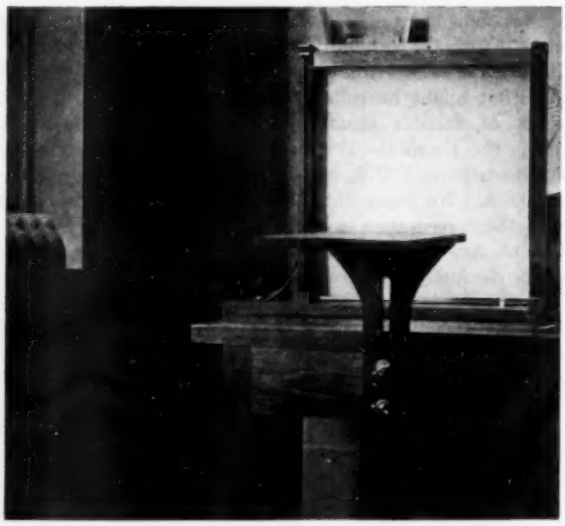


Fig. 30 Combined Projecting Screen, Vertical Drawing Board, and Camera Support

That the photoelastic results are reliable and agree with those from the mathematical theory of elasticity has been proved long before this. It is only desired to call attention to the vast difference in time and labor between the two methods. The summary of the approximate theoretical solution submitted fails to convey the amount of work necessary to get the results. Special tables of integrals have to be constructed, and many laborious and tedious computations have to be made which increase the danger of errors. It has been the author's experience that the photoelastic method takes about as many weeks as the theoretical method takes months.

The prevailing impression seems to be that the cost of a photoelastic laboratory is too high for the average engineering college or industrial organization. It may therefore be of interest to note that all the equipment described in this paper cost less than $1500, and it is believed that $2000 is sufficient to equip a moderate but efficient laboratory for quantitative work.

The main problem of this paper was suggested by Professor Timoshenko, who has kept in constant touch with this investi-

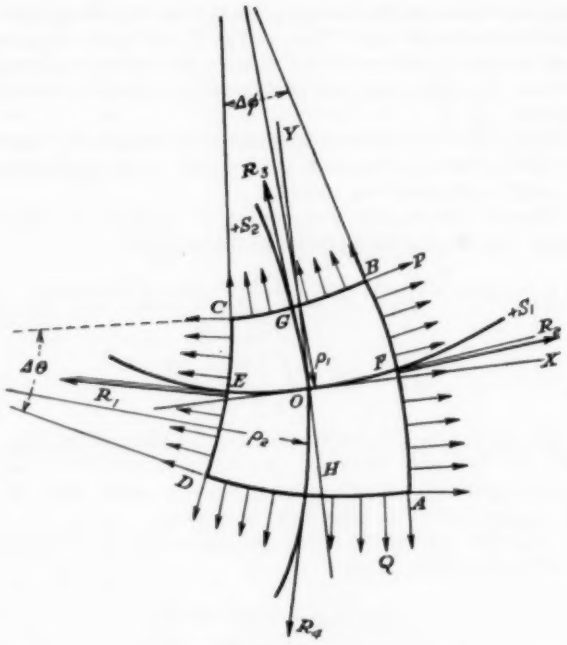


FIG. 31

gation and to whom the author is indebted for valuable advice. He also wishes to thank Dr. Tuzi for the sample of phenolite used in the investigation, and the Research Department of the Westinghouse Electric & Manufacturing Company for the bakelite which they were good enough to supply for the research.

Appendix No. 1

DERIVATION OF FUNDAMENTAL EQUATIONS FOR THE METHOD OF GRAPHICAL INTEGRATION

Let s_1 and s_2 denote two stress trajectories meeting at point O, Fig. 31.

Let s_1 be measured positive to the right of O and s_2 upward.

Let ρ_1 and ρ_2 be radii of curvature of s_1 and s_2, respectively, being measured positive if the tangent to the curve rotates counterclockwise as s_1 and s_2 increase.

In the element of Fig. 31 $OE = OF = \Delta s_1$, and $OG = OH = \Delta s_2$; also the boundary consists of arcs of the stress trajectories passing through points H, F, and G, and E.

Let P and Q represent average boundary stresses, and let R_1, R_2, R_3, and R_4 represent the resultant forces on the four sides of the element.

Lastly, let P_0 and Q_0 represent the resultant forces on GOH and EOF, respectively.

If $\Delta\phi$ is a small angle, the resultants R_1 and R_2 may be treated as tangents to EOF at E and F, respectively. Similarly R_3 and R_4 may be taken as tangent to GOH at G and H, respectively, the errors being infinitesimals of higher order. It follows that R_1 and R_2 make angles with the X-axis equal to $\Delta\phi/2$.

The Y-component of R_1 is $R_1 \sin \frac{\Delta\phi}{2}$

The Y-component of R_2 is $R_2 \sin \frac{\Delta\phi}{2}$

and the sum, which will be designated by R_y, is

$$R_y = (R_1 + R_2) \sin \frac{\Delta\phi}{2} = (R_1 + R_2) \frac{\Delta\phi}{2} \quad \ldots\ldots [47]$$

$\Delta\phi$ being a small angle.

For ρ_1 a positive radius of curvature R_y is positive that is upward, as can be seen from the figure. For equilibrium it is necessary that

$$\Sigma Y = 0 \quad \ldots\ldots [48]$$

Hence

$$(R_4 - R_3) = R_y \quad \ldots\ldots [49]$$

$$\therefore (R_4 - R_3) = (R_1 + R_2) \frac{\Delta\phi}{2} \quad \ldots\ldots [50]$$

Now

$$R_1 = \left(P_0 - \frac{\partial P_0}{\partial s_1} \Delta s_1\right) \quad \ldots\ldots [51]$$

and

$$R_2 = \left(P_0 + \frac{\partial P_0}{\partial s_1} \Delta s_1\right) \quad \ldots\ldots [52]$$

hence

$$R_1 + R_2 = 2P_0 \quad \ldots\ldots [53]$$

and [47] becomes

$$(R_1 + R_2) \frac{\Delta\phi}{2} = 2P_0 \frac{\Delta\phi}{2} = P_0 \Delta\phi \quad \ldots\ldots [54]$$

Also

$$R_3 = Q_0 + \frac{\partial Q_0}{\partial s_2} \Delta s_2 \quad \ldots\ldots [55]$$

and

$$R_4 = Q_0 - \frac{\partial Q_0}{\partial s_2} \Delta s_2 \quad \ldots\ldots [56]$$

$$\therefore R_4 - R_3 = -2 \frac{\partial Q_0}{\partial s_2} \Delta s_2 \quad \ldots\ldots [57]$$

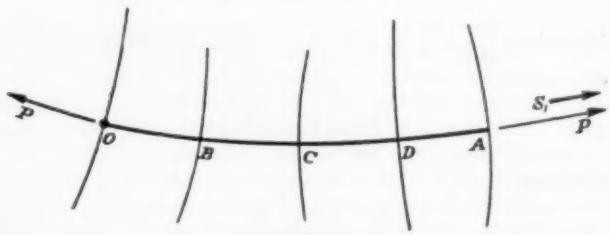


FIG. 32

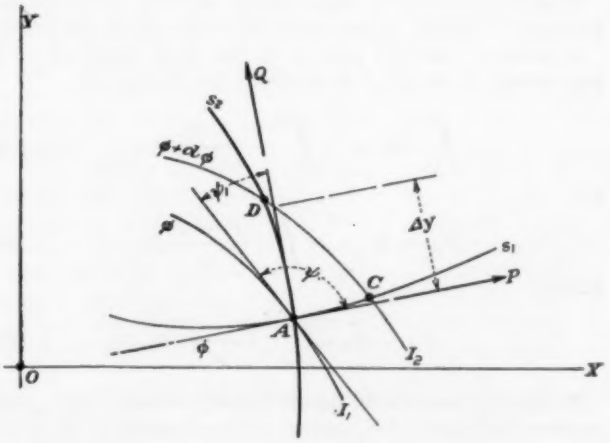


FIG. 33

Combining with [53],

$$-2\frac{\partial Q_0}{\partial s_2}\Delta s_2 = P_0\Delta\phi \quad [58]$$

or

$$\frac{\partial Q_0}{\partial s_2}\Delta s_2 = -\frac{P_0}{2}\Delta\phi \quad [59]$$

But as a first approximation

$$Q_0 = 2\Delta s_1 Q \quad [60]$$

and

$$P_0 = 2\Delta s_2 P \quad [61]$$

On substitution in [59],

$$2\frac{\partial}{\partial s_2}(\Delta s_1 Q)\Delta s_2 = -2\Delta s_2 P\frac{\Delta\phi}{2} \quad [62]$$

but

$$\Delta s_1 = \rho_1\frac{\Delta\phi}{2}$$

hence [62] becomes

$$2\frac{\partial}{\partial s_2}\left[\left(\rho_1\frac{\Delta\phi}{2}\right)Q\right]\Delta s_2 = -\Delta s_2 P\Delta\phi \quad [63]$$

But $\Delta\phi$ may be taken as a constant, so that [63] becomes

$$\frac{\partial}{\partial s_2}(\rho_1 Q)\Delta s_2\Delta\phi = -\Delta s_2 P\Delta\phi \quad [64]$$

or

$$\frac{\partial}{\partial s_2}(\rho_1 Q) = -P \quad [65]$$

$$\therefore \rho_1\frac{\partial Q}{\partial s_2} + Q\frac{\partial \rho_1}{\partial s_2} = -P \quad [66]$$

In the limit $\frac{\partial \rho_1}{\partial s_2} = -1$, since $\Delta\rho_1 = -\Delta s_2$ and we have $\rho_1\frac{\partial Q}{\partial s_2} - Q = -P$

$$\therefore \frac{\partial Q}{\partial s_2} + \frac{P-Q}{\rho_1} = 0 \quad [67]$$

Similarly,

$$\frac{\partial P}{\partial s_1} + \frac{P-Q}{\rho_2} = 0 \quad [68]$$

It will be observed that Equations [67] and [68] do not contain the elastic constants. Therefore they hold beyond the elastic limit.

If we start from any point O on the stress trajectory of P and follow it to any point A we obtain upon integration

$$\int_{P_0}^{P_A} dP = -\int_0^A \frac{(P-Q)}{\rho_2}ds_1 \quad [69]$$

or

$$P_A - P_0 = -\int_0^A \frac{(P-Q)}{\rho_2}ds_1 \quad [70]$$

Hence

$$P_A = P_0 - \int_0^A \frac{(P-Q)}{\rho_2}ds_1 \quad [71]$$

It must be remembered that in this derivation ρ_1 and ρ_2 were assumed positive if the rotation of the tangent to the curve is counterclockwise as the curves s_1 and s_2 increase and that s_2 and s_1 are interrelated, s_2 being obtained from s_1 by counterclockwise rotation of 90 deg. Thus in Fig. 32 if O is the origin and s_1 is measured positive to the right, ρ_1 is positive at all points between O and A, and ρ_2 is positive at O, B, and A, and negative at C and D.

In Fig. 32 if A is taken as an origin and s_1 is measured positive toward O, then ρ_1 is negative at all points; ρ_2 is negative at A, B, and O and is positive at D and C.

Professor Filon[8] gives two methods to evaluate the integrals on the right-hand sides of Equations [71] and [72]:

$$P_A = P_0 - \int_0^A \frac{(P-Q)}{\rho_2}ds_1 \quad [71]$$

$$Q_A = Q_0 - \int_0^A \frac{(P-Q)}{\rho_1}ds_2 \quad [72]$$

Let point A be a point in a body under plane stress, Fig. 33. Through this point there are two stress trajectories s_1 and s_2 and an isoclinic I_1 of parameter ϕ, where ϕ is the angle which the tangent to s_1 at A makes with the X-axis.

Let I_2 be a nearby isoclinic of parameter $\phi + d\phi$, intersecting s_1 at C and s_2 at D. Then

$$AD = \rho_2 d\phi; \text{ and } \frac{1}{\rho_2} = \frac{d\phi}{AD} \quad [73]$$

Also

$$ds_1 = AC$$

hence

$$\frac{1}{\rho_2}ds_1 = d\phi\frac{AC}{AD} \quad [74]$$

If ψ be the angle through which the stress trajectory of P has to be rotated (counterclockwise) in order to bring it upon the isoclinic I_1, then

$$\frac{AD}{AC} = -\tan\psi \quad [75]$$

the negative sign being necessary because tan ψ is negative and AD/AC is positive.

Upon substitution in [74], we have

$$\frac{ds_1}{\rho_2} = \frac{-d\phi}{\tan\psi} = -\cot\psi\, d\phi \quad [76]$$

Substituting in [71]

$$P_A = P_0 + \int_{\phi_0}^{\phi}(P-Q)\cot\psi\, d\phi \quad [77]$$

Also

$$\rho_1 d\phi = AC, \text{ and } \frac{1}{\rho_1} = \frac{d\phi}{AC}$$

$$\therefore \frac{ds_2}{\rho_1} = d\phi\frac{AD}{AC} = d\phi\cot\psi_1 \quad [78]$$

Upon substitution in [72],

$$Q_A = Q_0 - \int_{\phi_0}^{\phi}(P-Q)\cot\psi_1\, d\phi \quad [79]$$

The equations just discussed give good results when the angles ψ are large. For small values of ψ, Equations [77] and [79]

[8] L. N. G. Filon, "On the Graphical Determination of Stress From Photo-Elastic Observation," *Engineering*, Oct., 1923, p. 511.

become inaccurate as a small error in ψ makes a large error in cot ψ. In such case the intervals between successive isoclinics become large, and this adds to the inaccuracy of the method.

Professor Filon therefore gives another treatment for such cases. Let Δy, Fig. 33, be the intercept measured perpendicular to the path of integration between two near isoclinics in the neighborhood of the point considered whose parameters differ by $\Delta\phi$. Then

$$\frac{1}{\rho_2} = \frac{\Delta\phi}{\Delta y} \text{ (approximately)} \quad [80]$$

The isoclinics are usually drawn for constant differences of say, 5 or 10 deg., so that $\Delta\phi$ is a constant.

Substituting this in [67] and [68],

$$P = P_0 - \Delta\phi \int_0^A \frac{(P - Q)}{\Delta y}\, ds_1 \quad [81]$$

which again enables us by graphical integration to determine P and Q.

This last method is especially useful in cases of symmetry, in which the straight lines of symmetry are also isoclinics. Δy is then the intercept between this line and the nearest isoclinic, measured perpendicular to the path of integration.

SHORT BIBLIOGRAPHY

E. G. Coker, *General Electric Review*, November and December, 1920; January, March, May, 1921. (Five fundamental articles.) Also *Journal Franklin Institute*, October, 1923, and March, 1925.

A. L. Kimball, Jr., "Stress Determination by Means of the Coker Photo-Elastic Method," *General Electric Review*, January, 1921, pp. 73–81.

L. N. C. Filon, "On the Graphical Determination of Stress From Photo-Elastic Observation," *Engineering*, October, 1923, pp. 511–512.

Akira Miura, "Spannungskurven in Rechteckigen und Keilförmigen Trägern."

Z. Tuzi, Scientific Papers of the Institute of Physical and Chemical Research, Tokyo, Japan. Nos. 112–114, October, 1927; Nos. 149 and 156, August, 1928; Nos. 209 and 210, October, 1929.

Discussion

T. McLean Jasper.[9] The writer is grateful to know that some one has extended the work with polarized light and has developed its use with bakelite, which offers a material much more suitable than celluloid. Glass, which is probably the most suitable material from the optical standpoint, is so difficult to shape and therefore is not likely to yield answers to the most difficult problems in two-dimensional stress.

Most of our complicated machinery problems are three-dimensional in degree. It would be very gratifying if the photoelastic method could be applied to such problems. The photoelastic method is attractive and elegant and should draw very good support. However, its limitations should be carefully stated and understood.

The writer would like to suggest that the effect when using polarized light should be given by the following formula:

$$R = c\,(P - Q)\,t$$

in which the thickness of the material t is not considered as a part of the constant c. In this way corrections for the thickness change due to stress or variation in the thickness of models can be considered.

The writer also believes that a compensator such as the Babinet should be considered as part of the equipment of measurement and not trust to a tension specimen of the material for the work of evaluating the stress differential. In this way a much more refined gradation of stress measurement can be obtained.

[9] Director of Research, A. O. Smith Corporation, Milwaukee, Wis. Mem. A.S.M.E.

J. N. Goodier.[10] The writer would like to add his congratulations to the author on his achievements in obtaining such excellent fringe pictures and on his convincing demonstration of the virtues of this method of interpreting the photoelastic images.

Criticism of the method can hardly be directed toward the determination of maximum shear stress, a quantity so important in practice and so readily revealed by the apparatus. But in the isolation of the principal stresses, as the author points out, the same accuracy may not be preserved, especially on account of the difficulties of graphical integration. This may conceivably be the cause of the discrepancy between the theoretical and photoelastic curves in Fig. 26. It is true that if the actual distribution were similar to the photoelastic version, the theoretical approximation (Fig. 24) would be by no means adequate to represent it, but on the other hand experience of similar stress systems leads one to regard the photoelastic curve, representing the tensions across the vertical diagonal of the square, improbable as a consequence of the boundary tractions represented in Fig. 23, and therefore to doubt the conclusions based on it, particularly those regarding the situation and magnitude of the maximum tension. It would be very interesting to have an estimate of the possible error of the photoelastic results in the region $0.6h$ to $0.8h$ in Fig. 26.

Arshag G. Solakian.[11] The paper is an important contribution to the science of the optical-mechanical method of stress determination. It is very encouraging that during the last ten years important research work along this line has been carried on in different institutions of the United States. However, based on his experiences and observations in the Photoelastic Laboratories of Dr. E. G. Coker, of University College, London, and of Prof. A. Mesnager, of L'École des Ponts et Chaussees, Paris, and also including the considerable amount of research work in and publications about photoelasticity contributed by other European institutions, the writer is of the opinion that this interesting field should deserve more importance than is given to it by our physicists and engineers

Some time last November, Prof. G. B. Karelitz and the writer built up a simple optical set at the Physics Laboratories of Columbia University for the study of shearing stresses in keys and keyways of different types. Bakelite was adopted as the material used for the models, on account of its high optical sensitiveness in producing several color fringes for stresses within the elastic limit of the material. The method adopted for measuring the shearing stresses was exactly the same as that used by the author, the only difference being that a white source of light was used which produced color fringes of yellow, red, and green of several repeating orders. The author's method of black-and-white fringes due to the use of monochromatic light undoubtedly has the advantage of the formation of an image that can be easily and cheaply photographed and printed. On the other hand, the color fringe method has the following merits:

(*a*) With the black-and-white-fringe method it is very difficult to determine the order of a certain fringe for the evaluation of its corresponding stress value. It requires slow and repeated loading and unloading of the material and counting of the number of the fringes passing a certain point. Tension tests run by the writer at the Materials Testing Laboratories of Columbia University have revealed the fact that bakelite under repeated loading and unloading is not free from the influences of permanent set and of creep phenomena, both of which are functions of time and intensity of stress. Hence, stress conditions of the elements in a bakelite model under such loading are of a constantly changing nature.

[10] Commonwealth Fund Fellow, University of Michigan, Ann Arbor, Mich.

[11] Engineering Department, Columbia University, New York, N. Y.

With color fringes, the first-order yellow, red, and green bands are easily recognized by their brightness, relatively larger width of bands, and their location in the specimen. Hence the order of the others can be quickly determined by starting to count from the respective colors of the first order.

(*b*) The use of color fringes with three-color bands for each order against two bands in the black-and-white-fringe method has the advantage of increasing the optical sensitivity of bakelite in the ratio of 3 to 2.

(*c*) Another difficulty encountered in the use of the black-and-white-fringe method is due to the presence of black bands corresponding to $P - Q = 0$, which, intermixing with the stress fringes, cause certain confusion in their determination. The use of quarter-wave mica plates to produce circular polarized light eliminates only the direction bands called isoclinics. In color fringes the black bands corresponding to $P - Q = 0$ are easily recognized and eliminated.

The method of heat-treating bakelite, as adopted by the author to free the model from the effects of initial stresses, is similar to that used by the writer. Several trials at different temperatures have shown that a heating temperature of 75 to 85 deg. cent., with a heating time of at least one hour and uniform cooling down to room temperature during a period of about 12 to 14 hours, is very satisfactory to obtain the desired results. The writer is of the opinion that the presence of stubborn stresses requiring a second heat treatment for their removal, as suggested by the author, is greatly due to the very short time allowed for heating (15 minutes) and the lower value of temperature used (70 deg. cent.). In this connection it might be of interest to point out that under no circumstances should one try a temperature of 110 deg. cent. or above this value during the annealing process of bakelite on account of the change of transparency of the material. At temperatures above 180 deg. cent. a piece of perfectly transparent bakelite became red and non-transparent.

Both mechanical and optical tests run in the laboratories of Columbia University have indicated that the properties of bakelite are affected by annealing. Hence the stress-fringe order relations will no more be the same for the material in its initial and heat-treated stages. However, by annealing the calibrating beam under the same temperature and time conditions as that of the model, any discrepancy of this nature will be eliminated.

Observations of the stress fringes in the problem studied by the author give the impression to the writer that their very close spacing at points of stress concentrations might have produced stresses very close to the elastic limit of the material. Although bakelite, like most brittle materials, possesses neither a true elastic limit nor a distinct yield point, however time-creep tests of specimens of bakelite, both annealed and non-annealed, show that stresses within about 35 per cent of the maximum strength of the material do not produce an appreciable permanent set.

H. F. Moore.[12] At the University of Illinois a photoelastic apparatus has been developed using a petrographic microscope fitted with a Babinet compensator. The general appearance of the set up is shown in Fig. 34. The microscope is attached to the framework of a milling machine, and the specimen to be stressed and studied is mounted on the table of the milling machine and

[12] Research Professor of Engineering Materials, University of Illinois, Urbana, Ill. Mem. A.S.M.E.

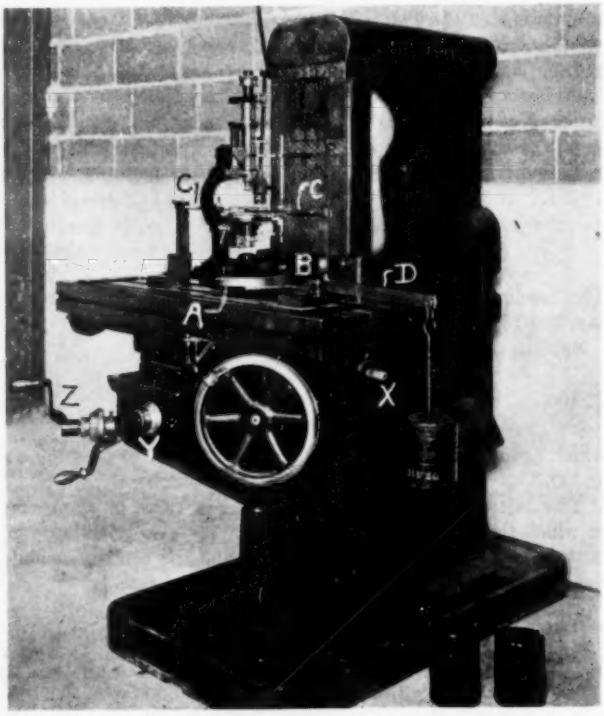


Fig. 34 (Right Top) Petrographic Microscope Mounted on Milling-Machine Frame
(Loading apparatus shown at *D*, grips for specimen at *C*.)

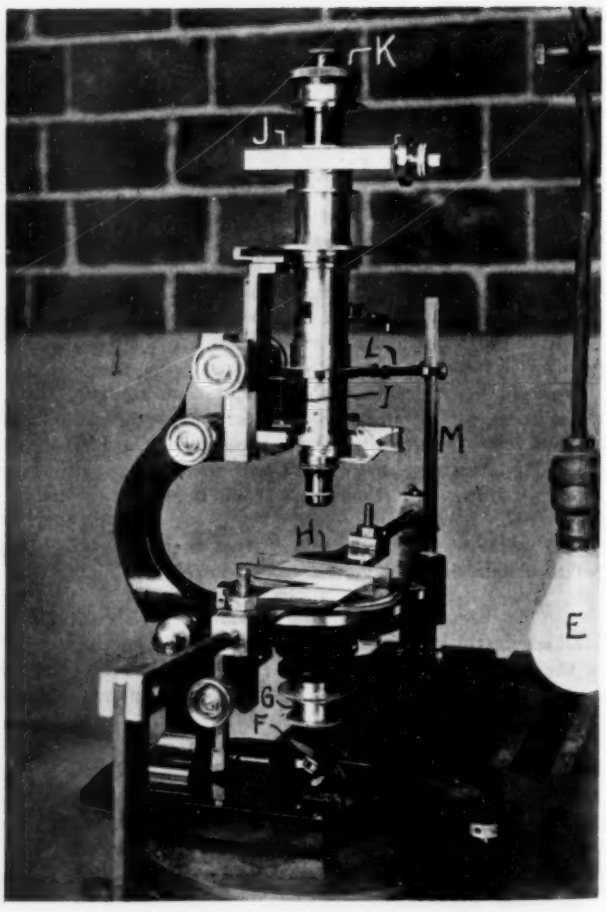


Fig. 35 (Right Bottom) Petrographic Microscope With Babinet Compensator *J* and Cap Nicol *K*
(Specimen shown at *H*.)

can thus be moved in three directions by micrometer control. Fig. 35 shows the microscope in somewhat greater detail. The polarizer is located at G, and at I is the ordinary analyzer, which can be moved out of the way. The Babinet compensator, which consists of two quartz wedges which can be slid over each other by a micrometer screw, is shown at J. In use the specimen is placed under the microscope, and the microscope is focused with the cross-hairs intersecting on the point at which the stress is to be determined. The Babinet compensator is not in place when this is done, and the ordinary eyepiece is used. A general notion of stress distribution can be obtained by the use of the analyzer I, noting the color bands (or if monocromatic light is used, the fringes). After the microscope is thus focused, the analyzer I is moved out of the way, and the Babinet compensator J is put in place, as is a cap Nicol K which is put in place of the eyepiece of the microscope. Looking through K and J, cross-hairs are seen and bands of colored light with a dark line about in the middle. Using the micrometer on J, this dark line is brought to the intersection of the cross-hairs. A definite load is then put in the specimen. The position of the dark line shifts, and it is again brought to the cross-hairs by means of the micrometer. The difference of micrometer readings with and without load is a measure of the difference of principal stresses at the point studied. The stress constant of the micrometer is obtained by a test on a simple tension specimen made of the same material and of the same thickness as that in which the stress is determined, and the direction in which *tensile* stresses cause the black line to move is also determined in this calibration test. The Babinet compensator was used for stress-distribution study some years ago by Mesnager. The testing rig used at Illinois has been developed by Prof. T. M. Jasper and later by Prof. R. Michel and Mr. S. G. Hall.

M. F. Sayre.[13] Great credit is due to Professor Kita and Mr. Matui in Japan and to the Westinghouse Research Laboratory and to Professor Frocht of the University of Michigan, in this country, for the development of satisfactory methods of preparing phenolite and bakelite for use in photoelasticity studies. Professor Frocht in his fringe photographs has admirably brought out the advantages of these materials; for it is only the high sensitivity of these materials which has made possible the fringe method. With celluloid or glass the photoelastic sensitivity is so much less that two or possibly three fringes represent the maximum that can be obtained without exceeding the elastic limit. It is this fact which in practice has compelled the use of the color rather than the fringe method to date.

One word of caution might be here interpolated for the benefit of novices. When dealing either with celluloid or bakelite, stresses must be kept within the elastic limit of the material to obtain accurate results. It is therefore desirable when starting work to find out experimentally the unit stress and the number of fringes corresponding to the elastic limit for one's particular material, and to avoid going beyond that number of fringes, regardless of the temptation to do so in order to obtain an attractive picture.

G. B. Karelitz.[14] The difficulty of an analytical solution of complicated cases of a stressed state makes all methods facilitating a visual study of stress distribution rather valuable. A number of analogies, such as the famous membrane analogy of twist, have been developed recently. Investigations of stresses in structures by the use of models have been also employed with success. However, the photoelastic method has given to date the most powerful and illuminating presentation of stress distribution in loaded members, provided that cases of two-dimensional stress are considered.

By the introduction of bakelite as a working material, the author has made a notable contribution to the art and should be congratulated on his achievement. The writer, together with Mr. A. Solakian, has been working at Columbia University on questions of photoelasticity, and they have also used bakelite as their material. Our observations regarding the ease of annealing, procedure, and temperatures coincide with those of the author, which is highly gratifying.

It seems to the writer that the author has not brought out with sufficient force the point that the fringe method, as described by him, is a powerful tool for designers. The main value of the photoelastic method is not in the possibility of determining the exact values of the principal stresses, but in obtaining at a glance the stress concentrations caused by irregularities in cross-section or loading. It helps also to form a correct idea about stress distribution in loaded members, which makes it an indispensable aid to instruction. From the standpoint of "practical" use the isoclinics are of no particular value, once bakelite is employed, the fringe picture itself giving sufficient information.

It also may be added that there is some justification for the use of colored images instead of monochromatics. When a complicated fringe diagram is analyzed, there may be regions where a monochromatic (such as Fig. 8) does not show whether a valley or a hill is encountered in the shearing stress. With a colored image, the succession of colors gives this information. The colors of the fringes change, with increase of shearing stress, from red to yellow, green, gray, red, yellow, etc. It is therefore clear that stress increases in the direction from yellow to green and decreases in the direction from yellow to red.

In conclusion, the writer wishes to express his admiration for the perfect photographs produced by the author.

Author's Closure

The author has read with interest and appreciation the discussions of T. McL. Jasper, J. N. Goodier, A. G. Solakian, H. F. Moore, M. F. Sayre, and G. B. Karelitz.

Mr. Jasper's suggestion to write the stress-optic formula in the form $R = c(P - Q)t$ so that the thickness of the material would be shown explicitly would be of benefit in models of variable thickness. In models of constant thickness the variation due to stress can be neglected for all engineering purposes. In such cases it is simpler to include the thickness in the constant of proportionality.

The author agrees that a Babinet compensator would be a valuable addition to a well-equipped photoelastic laboratory and also that it is not desirable to trust the tension-strip compensator. He wishes, however, to emphasize that in the fringe or monochromatic method employed by him the difference between the principal stresses is determined without resort to such a tension compensator (see Fig. 1). The accuracy of this method is good, the procedure simple, and there is considerably less labor involved then either with the Babinet or the tension compensator.

The author shares Mr. Goodier's opinion that it would be desirable to check the values of the normal stresses on the vertical diagonal beyond $0.6h$ by a method other than that of graphical integration, and he plans to do so when suitable equipment is available. The 5-deg. isoclinic is asymptotic to the vertical axis of symmetry, Fig. 19. The distances marked ΔX thus become very small beyond $0.5h$, thereby increasing the per cent error in the values of the normal stresses the accuracy of which depends on the degree of precision with which these distances are measured.

The author was very much interested in the comments made by

[13] Associate Professor of Mechanical Engineering, Union College, Schenectady, N. Y. Mem. A.S.M.E.

[14] Associate Professor, Columbia University, New York, N. Y. Mem. A.S.M.E.

Mr. Solakian, as he raises the fundamental question of the superiority of the fringe or monochromatic method over that of white light. In paragraph (*a*) Mr. Solakian states: "With the black-and-white-fringe method it is very difficult to determine the order of a certain fringe for the evaluation of its corresponding stress value. It requires slow and repeated loading and unloading of the material and counting of the number of the fringes passing a certain point." Leaving for the moment the question of difficulty involved in determining the fringe order, the author wishes to point out that even in the white-light method essentially the same difficulties exit. The process of counting is as necessary with white light as it is with monochromatic light, since the order of a color band must be known before a numerical value can be attached to it. The same is true about the need for slow loading. If the fringe pattern is of the escaping type, so is the color pattern, and to determine the number of escaping isochromatics, one has to watch their formation during slow loading. The determination of fringe No. 1, as well as of the whole fringe order, presents in reality no difficulties.

It is the author's opinion that the existence of optical creep is really a strong point against the white-light method and in favor of the fringe or monochromatic method. It takes only a fraction of a second to take an accurate fringe photograph, and it may take an hour or more to sketch isochromatics with even a partial degree of accuracy. The existence of creep is therefore bound to affect the drawing of isochromatics vastly more than it does a picture. As a matter of fact the sharpness of the fringe photographs is evidence of the absence of effect of creep. For best results the fringe photograph should be taken immediately after the load is applied, and the determination of the fringe order

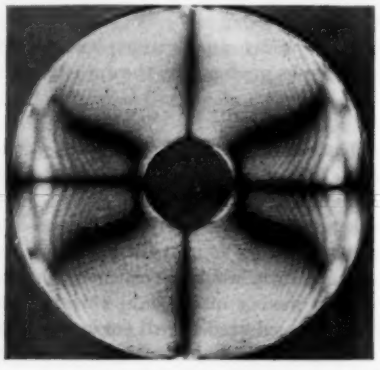


FIG. 36 BAKELITE-MODEL WHITE-LIGHT PHOTOGRAPH
(Showing the vagueness of isochromatics of higher order. Compare with Fig. 8. The black curves represent the zero isoclinic.)

should be carried out later. The presence of creep during the determination of the fringe order is evidently of no importance.

In paragraph (*b*) Mr. Solakian concludes that the use of white light increases the optical sensitivity of bakelite in the ratio of 3 to 2 by furnishing three bands of yellow, red, and green to two bands of black and white. It seems to the author that it would be more correct to say that the fringe method increases the optical sensitivity rather than the white-light method. Mr. Solakian has undoubtedly observed that isochromatics become narrower and narrower as their order increases. In fact, beyond the fifth green the colors become so crowded and washed out as to be of little value. Assuming that all isochromatics can be traced with accuracy (a task not always easy to perform, especially in the presence of creep), and taking as a practical limit the sixth green, we have a total of 18 isochromatic bands. With the fringe method the patterns are distinct even beyond the 15th fringe, which represents approximately the elastic limit of bakelite. We thus have within the elastic limit 30 bands with the fringe method, against 18 with white light, or an effect equivalent to an increase in optical sensitivity of more than 3 to 2 (see Fig. 36).

The author agrees with Mr. Solakian that the use of white light brings out very clearly the points of $P - Q = 0$ by providing a colored background for the dark points or bands. A little experience, however, enables one to tell these points with relative ease when a mercury-vapor lamp is used. All one needs to do is to remove the color filter, and the dark bands come into relief.

The author has had few occasions to go to temperatures higher than 70 deg. cent. to obtain well-annealed models, and because the changes produced in the bakelite increase with the temperatures, he prefers to anneal at 70 deg. cent., and only in exceptional cases to go a few degrees above that.

The points at which the loads were acting in the problem studied by the author were undoubtedly stressed beyond the elastic limit. That, however, is inevitable if an appreciable optical effect is to be produced on the horizontal diagonal. Before closing, the author wishes to point out an aspect of the monochromatic method which has thus far been neglected—i.e., the directional effect of stress when white light is used.

Consider two elements A and B in a body subjected to a two-dimensional stress system (see Fig. 37). Let us assume that the

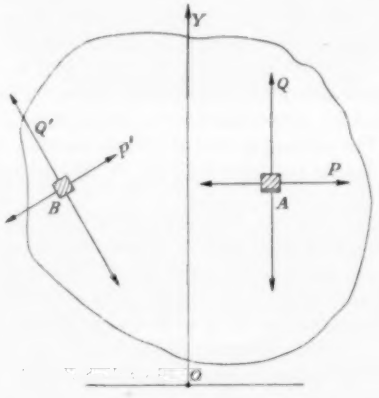


FIG. 37

principal stresses on these elements are equal in magnitude but differ in direction; i.e., $P = P'$; $Q = Q'$. If a ray of plane polarized light be transmitted through this body, it will be resolved into rectilinear components in directions of the principal stresses the magnitudes of which will depend on the direction of the plane of vibration of the impinging ray. The character of the emerging ray will thus depend on the direction of the principal stresses. If, for concreteness, the impinging-plane polarized ray be vibrating in a vertical plane, then the emerging ray would still be plane polarized at point A, while the one at B would generally be elliptic. In such cases the optical effect of a stress system acting upon an element would depend on the direction of the stresses as well as their magnitudes. To obviate such a condition the plane-polarized ray is replaced by a circularly polarized one, which breaks up into equal components regardless of the directions of the principal stresses. Such a condition is implied in the stress-optic law $R = c\,(P - Q)$, which does not contain the directions of the stress system. Now, the most effective way to produce circular polarization is by means of a quarter-plate. But this quarter-plate has meaning only when used with monochromatic light. There can evidently be no quarter-plate for white light. If white light be merely the combined effect of the fundamental colors having different wave lengths, then obviously a quarter-plate can be made to fit one but not all of the colors

simultaneously. So that white light gives elliptic polarization and not circular, with a consequent result that there is a directional effect of the stresses at a point.

The author has made some experiments to verify this theory. Strips of celluloid and bakelite were subjected to tension and placed in the path of plane and circular rays, respectively. They were then rotated in the plane normal to the axis of propagation of the ray of white light, and the changes of color were observed.

In the plane-polarized field, both the celluloid and the bakelite showed marked changes in brightness, the maximum brilliance being at an inclination of 45 deg. to the plane of vibration. In the circularly polarized field the celluloid strip seemed to hold its color, but the bakelite strip showed pronounced changes. Starting with a deep green in the vertical position, the color changed and showed a clear tinge of red in the horizontal position, indicating a possible error of approximately 20 per cent. What seemed to be evidence of circular polarization with celluloid thus proved to be evidence of its small optical sensitivity. There is therefore a directional effect when white light is used. Such, however, is not the case with monochromatic light. Here a quarter-plate can be made to produce true circular polarization and thereby to eliminate directional effect and to increase the accuracy of the work.

The fringe or monochromatic method has also the advantage that it makes the taking of clear moving pictures practicable, and thereby contributes an effective tool for many fields of investigation, especially for the study of impact stresses.

The author was glad to learn from H. F. Moore and M. F. Sayer of the photoelastic work which is being done at their respective institutions. He also wishes to thank G. B. Karelitz for emphasizing the practical value of the fringe method for designers. While the author champions the fringe or monochromatic method, believing that it has genuine merits and that it will facilitate the dissemination of knowledge of photoelasticity, there are two occasions on which he uses white light as a source: one of these is when sketching isoclinics and the other when investigating the signs of boundary stresses.

APM-53-12

Influence of Damping in the Elastic Mounting of Vibrating Machines

By E. H. HULL,[1] SCHENECTADY, N. Y.

This paper considers the elastic material for, and design of, elastic mountings intended for the prevention of transmission of vibration from electrical apparatus into the structure of buildings in which comparative quiet is desired. The effectiveness of such a support depends on the square of the ratio of the frequency of the impressed disturbance to the natural frequency of the machine on its mounting. Thus the natural frequency of the machine on its mounting must be made low in comparison with the impressed frequency. It is shown analytically that damping, or internal friction, in an elastic mounting causes additional transmission through the mounting. An apparatus was built to compare directly the effectiveness of materials proposed for elastic support. The detrimental effect of damping is clearly shown at high ratios of impressed frequency to natural frequency, causing the transmission to be doubled in the case of some samples. Therefore when choosing the elastic material for a support it is necessary to consider the damping present in the substance. Drift tests were also made to aid in determining the best material for this work. In several ways rubber appears to be well suited to use as an elastic support material. As an example, its incorporation in a mounting for a 15-hp. induction motor is described.



THE present tendency toward lighter building construction creates structures which transmit noise and vibrations throughout their framework more readily than do buildings of the older types. Recently, elastic mountings have been placed under apparatus which creates objectionable disturbances in these modern buildings. Excellent results are obtained in preventing the transmission of vibration to the building by the use of such mountings, provided that these supports are correctly designed and applied.

Elastic supports may be divided into two types: (1) Mountings intended for isolation from disturbances of any frequency or from shocks, and (2) mountings designed for isolation from vibration of one, or several, known and constant frequencies. Mountings of the first type are exemplified by the automobile suspension in which damping of some sort is known to be beneficial since road shocks may occur at the natural frequency of the vehicle on its springs. The present investigation deals with the effect of damping in the elastic material of mountings of the second type. A mounting constructed to prevent the transmission of electromagnetic noise from a three-phase induction motor to its foundation is described as an example of this second type.

Demonstration

An apparatus sketched in Fig. 1 gives an idea of the function of an elastic support. The mass M is supported through a spring k from a crosshead P moving with constant amplitude in approximately simple harmonic motion. At a very low cranking speed M moves through the same amplitude and in phase with P. As the speed increases the amplitude of M increases to a maximum at the resonant speed.

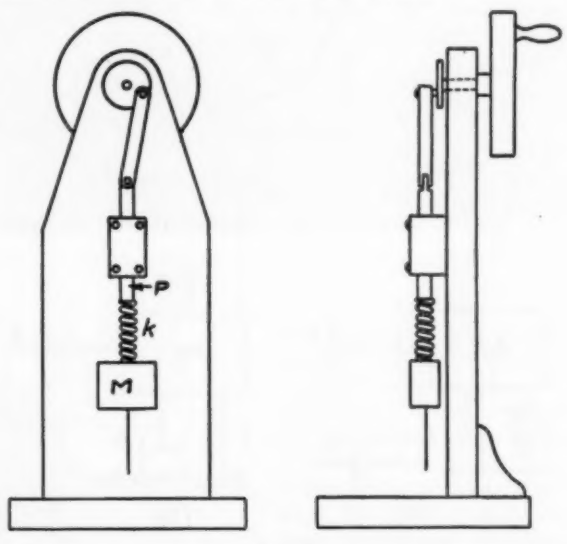


Fig. 1 Apparatus for Demonstrating the Effect of Elastic Mounting

In this range the motions of P and M, which have been in phase, shift through 180 deg. and continue out of phase at higher speeds. Any increase in speed above the resonant speed tends to decrease the amplitude of M until it is found, at a speed several times the resonant speed, that the mass remains practically stationary in space.

The curve marked $\delta = 0$ in Fig. 2 indicates the relation between speed and amplitude, showing clearly that the elastic support of M is harmful until a speed of 1.41 times the resonant speed is reached, but becomes increasingly effective as the frequency is raised above that speed. At a speed of several times that at which resonance occurs, M moves with a small fraction of the impressed amplitude.

Simple Theory

Fig. 3(*a*) shows schematically the essentials of the demonstration apparatus with damping added. k is the stiffness constant of the spring in force per unit deflection, the mass $M = W/g$,

---

[1] General Electric Co. Jun. A.S.M.E. Edwin H. Hull was born at Derby, Conn., in 1902; was graduated from the Sheffield Scientific School of Yale University in 1924, and since then has been in the Research Laboratory of the General Electric Company.

Presented at the National Applied Mechanics Meeting, Purdue University, Lafayette, Ind., June 15 and 16, 1931, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

and the damping, assumed to follow the viscous-friction law, contains a constant b having the dimensions force per unit linear velocity. The disturbing force of frequency ω and the resulting motions are assumed to be sinusoidal.

If the transmissibility ϵ of the system is defined as the ratio of the maximum amplitude of M to the maximum impressed amplitude, then the Appendix gives

$$\epsilon = \sqrt{\frac{b^2\omega^2 + k^2}{b^2\omega^2 + (k - M\omega^2)^2}} \quad \ldots\ldots [6]$$

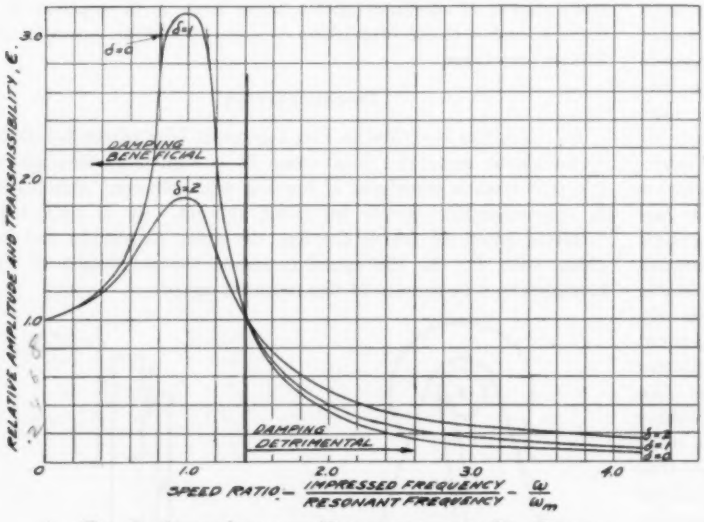


FIG. 2 AMPLITUDE AND TRANSMISSIBILITY VS. SPEED

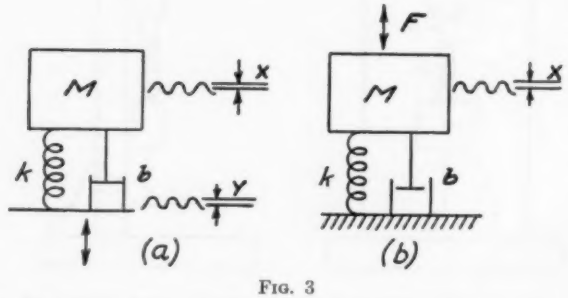


FIG. 3

When $b = 0$,

$$\epsilon = \frac{1}{1 - \left(\frac{\omega}{\omega_e}\right)^2}$$

or, since we are not interested in phase,

$$\epsilon = \frac{1}{\left(\frac{\omega}{\omega_e}\right)^2 - 1} \quad \ldots\ldots [7]$$

where $\omega_e = \sqrt{(k/M)}$, the resonant speed of the system.

Elastic mountings of the second type are usually represented by the system shown in Fig. 3(b). This case includes vibrating machines isolated from buildings by an elastic suspension. Here the transmissibility will be given by the ratio of the maximum value of the force transmitted to the foundation to the maximum value of the impressed force. In the Appendix it is shown that Equations [6] and [7] hold in this case as well as in the former.

In order to obtain comparisons between various systems it is convenient to have the damping term given by a dimensionless quantity. For the cases in which the viscous-friction law holds (frictional resistance proportional to velocity) it is easily shown[1] that the logarithmic decrement, $\delta = \pi b/\sqrt{(Mk)}$. Substituting in [6],

$$\epsilon = \sqrt{\frac{1 + \frac{\delta^2}{\pi^2}\left(\frac{\omega}{\omega_e}\right)^2}{\left(1 - \left(\frac{\omega}{\omega_e}\right)^2\right)^2 + \frac{\delta^2}{\pi^2}\left(\frac{\omega}{\omega_e}\right)^2}} \quad \ldots\ldots [8]$$

Fig. 2 shows that the transmissibility in the range from $\omega/\omega_e = 3$ and up may be more than doubled by common values of δ. Thus damping is beneficial in suspensions of the first type in which the speed ratio ω/ω_e may vary over an unknown range and hence be less than 1.41. On the other hand, damping is detrimental in supports of the second type, in which with proper design the speed ratio will never be near unity.

APPARATUS AND EXPERIMENTS

Since it is not known what law is followed by the friction in the various materials available for elastic support, or whether the stiffness of the material remains constant, it was thought advisable to make tests measuring the transmissibility directly. The usual troublesome frequency encountered in electrical machinery is twice the fundamental electrical frequency or 120 cycles per second for 60-cycle power supply. By experience we have found that a speed ratio ω/ω_e of 6 or 8 is desirable in isolating this vibration. In order to make the test results as applicable as possible to practical problems, it was decided to make measurements over this range. Unit pressure and thickness of the tested materials were to be kept equal or near to those used in practice.

In directly determining transmissibility in a system similar to that in Fig. 3(a), three measurements are necessary, namely, two amplitudes and a speed. Likewise, in the system of Fig. 3(b) two forces and speed are necessary. Since for this case it is easier to measure small amplitudes than small alternating forces, the former system was chosen for test purposes. It has been shown in the Appendix that both systems give the same expressions for transmissibility.

Figs. 4 and 5 give some idea of the construction and arrangement of the test apparatus. The horizontal shaft S, motor driven through a belt and pulley at one end, carries a flywheel F at the other. Sunk in the face of F is a circular hole, eccentric with the shaft S. A disk D is held in this hole by the nut N. D carries a ball-bearing support at a distance from the disk center equal to the eccentricity of the hole in F. Hence, by changing the angular position of D, the eccentricity of its ball-bearing support may be varied from zero to twice the eccentricity of the support in D. A connecting rod R leads to a crosshead carrying a table T. Any slight rotation of T is prevented by the horizontal phosphor-bronze strip Q.

Samples of isolating material O are placed on the table T, and support the $48^1/_2$-lb. weight W which is prevented from tipping by six horizontal wires. Speed is measured by an electric tachometer U having a range up to 10,000 r.p.m. The two cathetometers, C_1 and C_2, supported at points some distance from the apparatus, measure the amplitudes of the table and weight, respectively, by reading the apparent width of a point against a calibrated eyepiece scale. Distinct points were best

[1] A. L. Kimball, Trans. A.S.M.E., 1929, APM-51-21.

obtained by strongly illuminating the surface tangentially and selecting one of a number of minute, brilliant points of light reflected by dust or metal particles adhering to the surface. A dial indicator (not shown) was used to read the weight amplitude near resonant speed.

In performing the tests, the eccentricity was adjusted to produce a double amplitude of the platform of 12–15 mils. The thickness of the sample was initially chosen as that recommended by the manufacturer of the material and the area adjusted to conform with his recommended loading pressure. Readjustments of these dimensions were often necessary to give proper isolation. The prepared sample was placed centrally on the platform and the weight lowered upon it.

As the speed was slowly raised, the platform and weight amplitudes and speed were read at frequent intervals. The first occurrence of interest was the maximum-amplitude point of the weight, which was carefully determined. Readings were taken at frequent intervals from this point up to the highest speed which would produce a readable amplitude of the weight, or until vibration in the microscope supports made further reading impossible. In certain cases the energy dissipation due to friction in the sample raised its temperature sufficiently to change its stiffness and friction coefficient. When this occurred the sample was run at a high speed until a temperature equilibrium was reached, the readings then being taken as the speed was decreased.

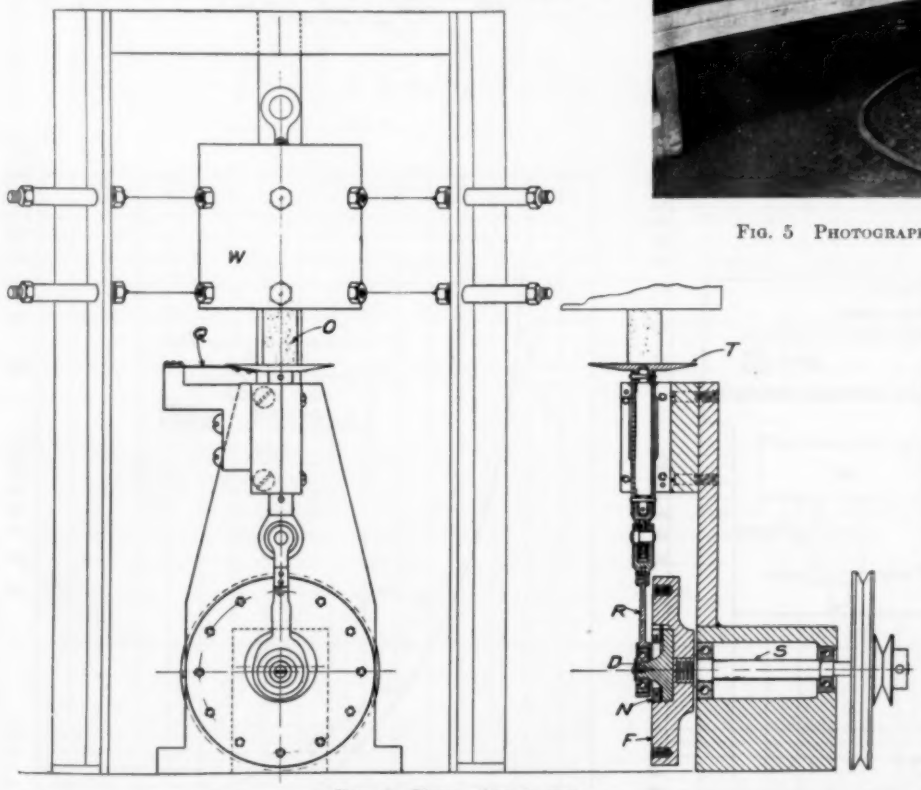


FIG. 4 TEST APPARATUS

RESULTS AND DISCUSSION

Results were plotted as shown in Figs. 6–11. The transmissibility ϵ is, as before, the ratio of the amplitude of the weight to the platform amplitude and is plotted on a logarithmic scale in order to magnify the situation at high ratios of impressed frequency to frequency of maximum amplitude of the weight, ω/ω_m. f_m, noted on each set of curves, is the frequency in vibrations per second corresponding with ω_m, an angular frequency. The solid line on each of the plots is the $\delta = 0$ curve from Fig. 2 representing the transmissibility which should be obtained with constant stiffness and no damping in the material. Steel springs follow the latter curve very closely.

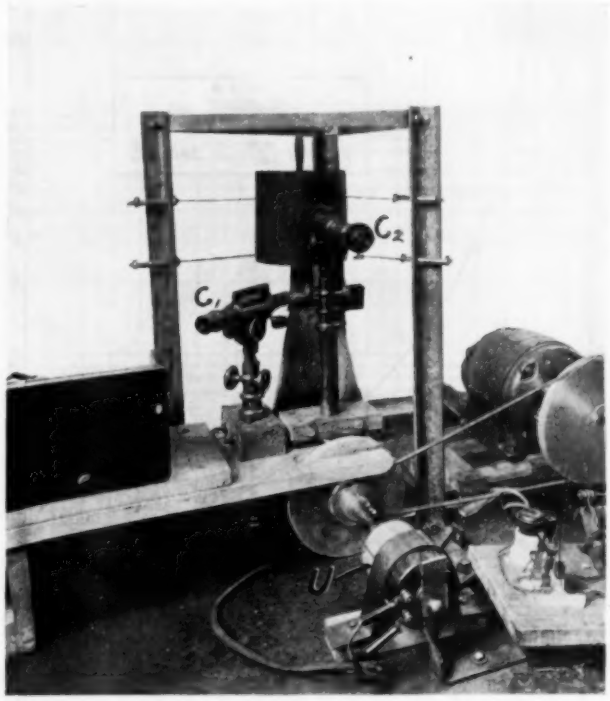


FIG. 5 PHOTOGRAPH OF TEST ARRANGEMENT

Experimental results from a high-quality rubber compound, containing about 90 per cent pure rubber, are given in Fig. 6. Damping in the material is low, as is shown by the large amplitude at resonance; therefore the increase in transmissibility due to damping is small.

Fig. 7 indicates the results obtained from coarsely ground, compressed cork, and also disproves the popular fallacy that interfaces of unlike materials reflect "vibration waves" analogous to the partial reflection of light on passing between bodies of different refractive indices. Data taken with and without the steel plate between the cork samples fall along the same curve.

The broken line of Fig. 8 shows results obtained from a natural cork, intended for isolation purposes, loaded in accordance with the manufacturer's recommendations. Amplification occurs at frequencies up to 2900 r.p.m. Tripling the load per unit area gave isolation at frequencies above 2000 r.p.m. as shown by the solid curve. Fig. 9 shows the latter data plotted for comparison with the transmissibility which could be obtained for the same stiffness with no damping.

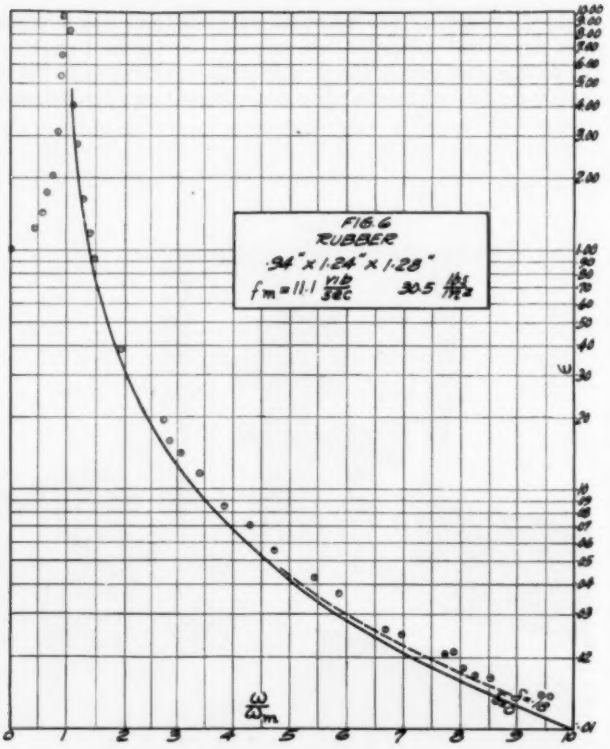


FIG. 6

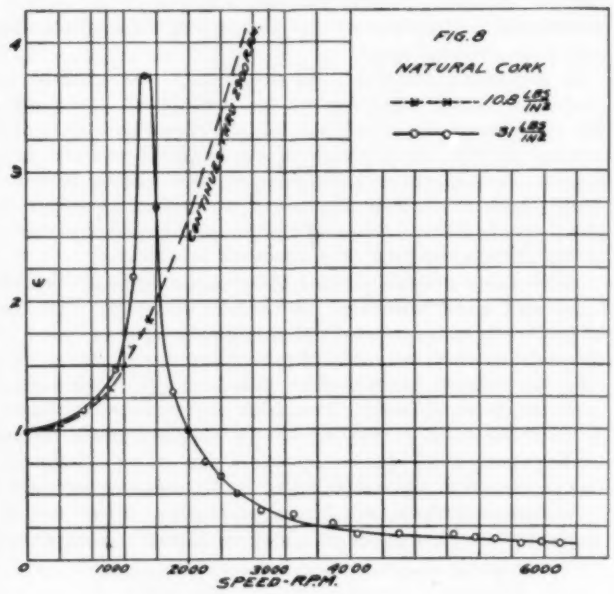


FIG. 8

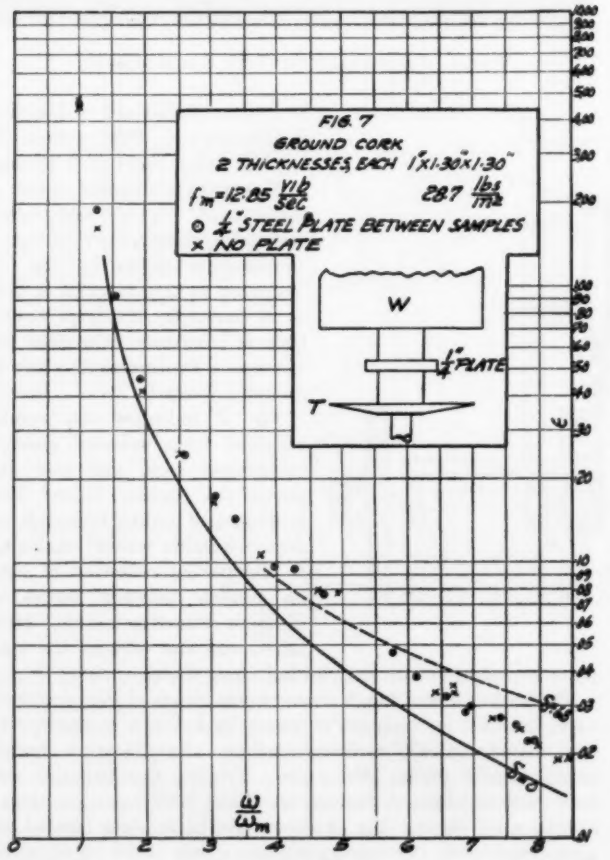


FIG. 7

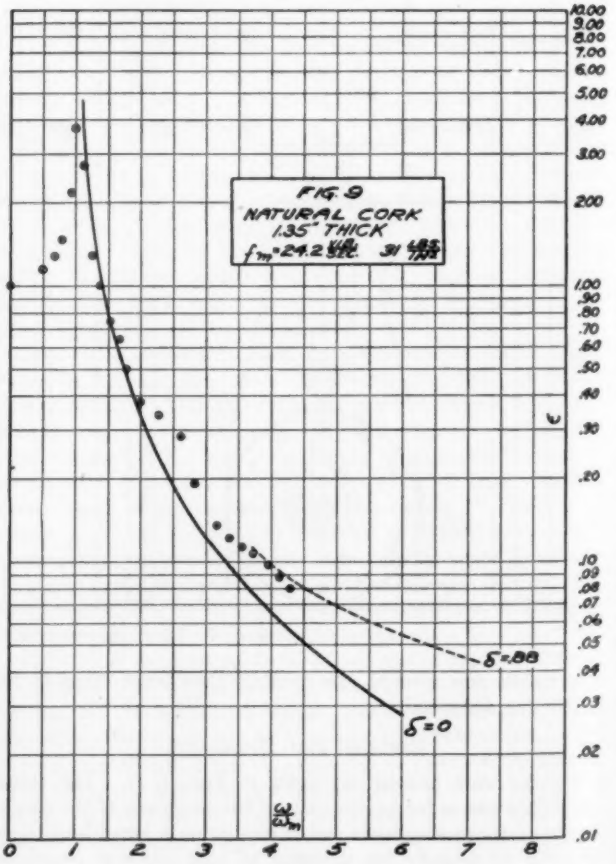


FIG. 9

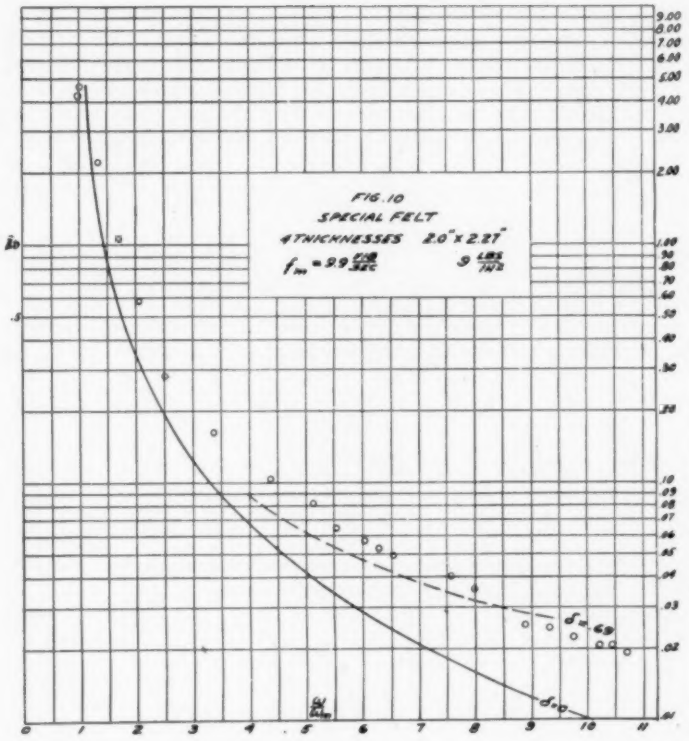


FIG. 10

A material similar to hektograph-pad gel required four thicknesses at 30 lb. per sq. in., rather than the maker's recommendations of one thickness, to produce satisfactory isolation (see Fig. 11).

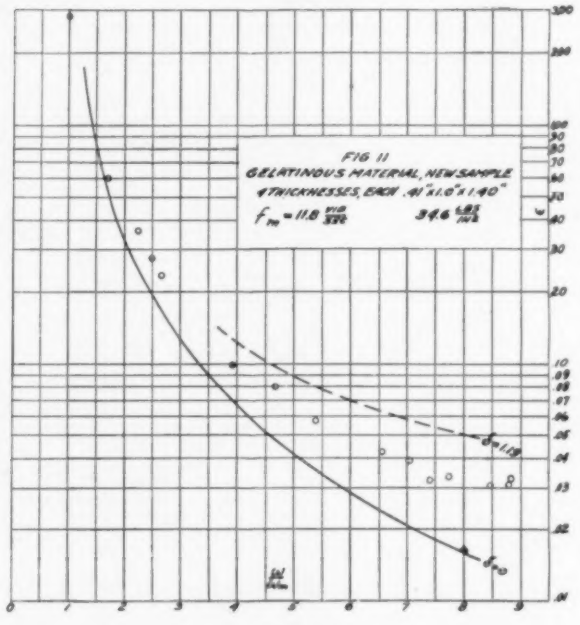


FIG. 11

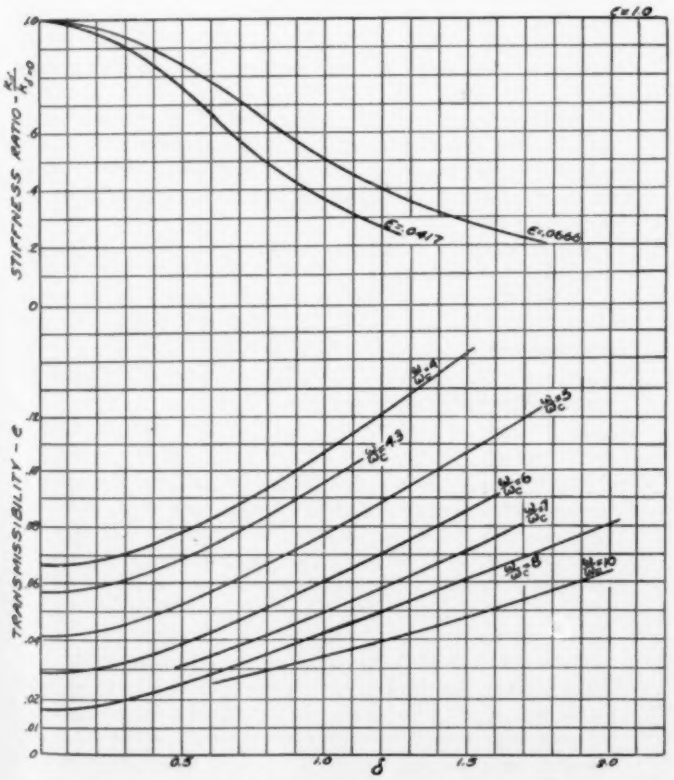


FIG. 12 COMPARISON OF MOUNTINGS FOR VARIOUS DECREMENTS

In the case of a special hair felt made for isolation purposes, Fig. 10 shows that the transmissibility is doubled at a speed ratio of 8.

In the analyses given in the Appendix, friction was assumed proportional to the velocity of extension and compression in the material. Some idea of the degree of approximation in this assumption may now be obtained.

When there is considerable damping present in a vibrating system of one degree of freedom, maximum amplitude does not occur at the value of speed given by $\omega_c = \sqrt{(k/M)}$, but is displaced in a direction and amount dependent on the type of system considered.

In this case Equation [6] gives

$$\frac{x}{y} = \sqrt{\frac{b^2\omega^2 + k^2}{b^2\omega^2 + (k - M\omega^2)^2}} \quad \ldots\ldots\ldots\ldots [6]$$

Solving for x, differentiating with respect to ω, and simplifying,

$$\omega_c^2 = \omega_m^2 + \frac{b^2\omega_m^4}{2k^2} \quad \ldots\ldots\ldots\ldots [9]$$

Thus, for this case, friction reduces the speed at which maximum amplitude occurs.

A first approximation of k and b may be obtained by assuming that $\omega_c^2 = \omega_m^2$. Then

$$k = M\omega_m^2$$

Equation [6] may now be solved for b, using the measured value of x/y at ω_m. Substituting these approximate values of k and b in [9] gives a value for ω_c^2, from which more exact values of k and b may be obtained by repeating the above process. It is found that this second determination of b is sufficiently accurate. The logarithmic decrement for the material may now be found using the previously given expression

$$\delta = \frac{\pi b}{\sqrt{Mk}}$$

Values for some of the substances tested are given in Table 1.

TABLE 1 VALUES OF δ FOR VARIOUS SUBSTANCES

Material	δ
Rubber	0.18
Special felt for supporting purposes	0.69
Cork, coarse ground and compressed	0.68
Gelatinous material similar to hektograph pad { 6 mo. old	1.93
Gelatinous material similar to hektograph pad { new	1.19
Natural cork loaded perpendicular to grain	0.88

The broken-line curves in Figs. 6–11, excluding Fig. 8, show the values of transmissibility which would be obtained from these various materials based on the calculated δ at ω_m, if these materials obeyed the viscous-friction law.

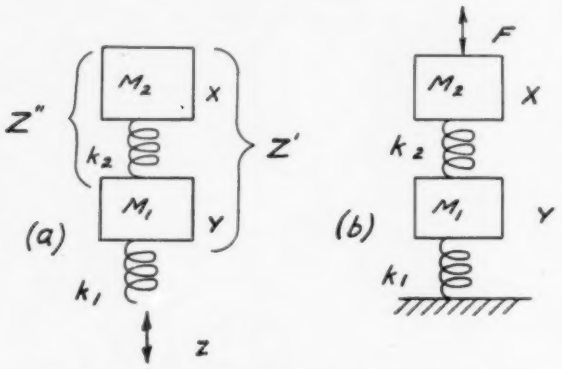


FIG. 13 TWO-DEGREE-OF-FREEDOM SYSTEMS

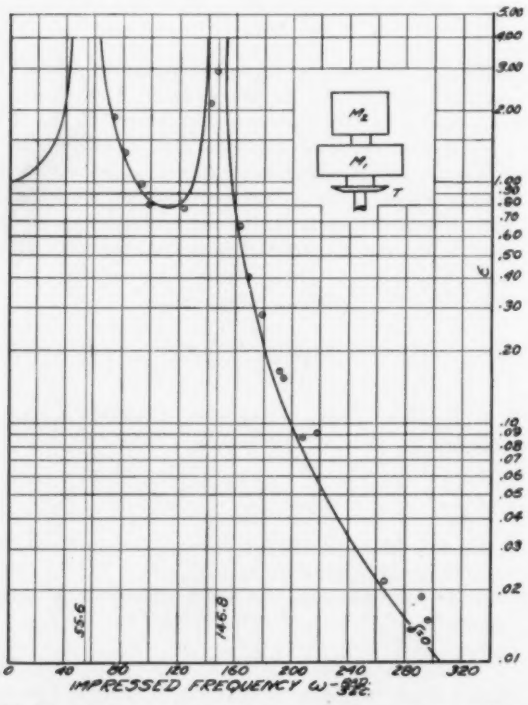


FIG. 14 TEST RESULTS IN A SYSTEM OF TWO DEGREES OF FREEDOM

Apparently this law gives a general idea of the increased transmissibility for an average case in the frequency range of $\omega/\omega_m = 6$ to 8, but may vary somewhat for particular cases.

A comparison may be made on the basis of this theory between two mountings, one in which $\delta = 0$ and the other in which $\delta = 1$.

The lower set of curves given in Fig. 12 shows the increase in transmissibility with increasing δ for constant speed ratios, and brings out the fact that δ has an increasingly greater effect as the speed ratio is raised. The upper group of two curves shows the amount by which the stiffness of a system having a certain δ must be reduced below the stiffness of a system in which $\delta = 0$, in order to maintain a constant transmissibility. When elastically mounted machines are required to transmit torque to apparatus on a separate base, the mounting stiffness should be as high as possible, consistent with good isolation, thus requiring a low value of δ.

Other damping laws might be considered, the solid-friction law,[3] for instance, in which the friction loss per cycle is independent of frequency but proportional to the square of the maximum stress in the material. Kimball[3] has shown in this case that for a sustained vibration

$$b' = \frac{\xi E k}{\pi \omega}$$

and that $\delta = \xi E$. Combining,

$$b' = \frac{\delta k}{\pi \omega}$$

This value when substituted for b in Equation [6], after dividing numerator and denominator by k^2, gives

$$\epsilon = \frac{x}{y} = \sqrt{\frac{\frac{\delta^2}{\pi^2} + 1}{\frac{\delta^2}{\pi^2} + \left(1 - \frac{\omega^2}{\omega_c^2}\right)^2}} \quad \ldots\ldots [10]$$

The values of δ computed by this expression and that of Equation [8] from values of x and y at ω_m will be very nearly the same since $(\omega_m/\omega_c)^2$ is close to unity. On the other hand, a large difference in the value of ϵ does occur at speeds well away from resonance.

The point marked with a triangle at $\omega/\omega_m = 8$ on Fig. 11 indicates the expected increase in transmissibility according to this law. The predicted increase is negligible, hence the law should be rejected for substances such as cork, gelatinous material, and felt.

The damping law for each material tested is probably different. Assuming that the stiffness for each material remained constant, these laws for the particular conditions of the test could be discovered from the data presented. This is not thought worth while since the assumption of the viscous-friction law gives an indication of the amount which friction raises the transmissibility in the most important range.

SYSTEM OF TWO DEGREES OF FREEDOM

Elastic mountings in the form of systems of two degrees of freedom offer interesting possibilities. Such a system was set up on the test apparatus using an additional elasticity and mass arranged as sketched in Fig. 13(*a*), rubber being used as the elastic medium. Fig. 14 shows the test points as well as a curve indicating the expected result when $b = 0$, obtained from Equation [14] developed in the Appendix.

The steep slope of the curve above the second resonant speed indicates that the support is very effective at a short distance above this speed. For instance, the two-degree system gives a calculated ϵ of 0.02 at 266 radians per sec. A one-degree-of-freedom system would require a resonant speed of 38 radians per sec., which is less than either of the two-degree-of-freedom resonant speeds, to produce the same transmissibility.

[3] A. L. Kimball and D. E. Lovell, *Phys. Rev.*, vol. 30 (1927), pp. 948–959.

Drift Tests

Drift tests to determine the loss in thickness with time under load were made on samples of materials used in the above experiments. A simple apparatus was used (see Fig. 15) consisting of a 112-lb. weight, held in a vertical position by horizontal wires, under which the sample S is placed. The dial indicator I gives the drift in mils. Variation in stiffness with drift is obtained from measurements of the resonant frequency of the weight on the sample at several times as the test progressed.

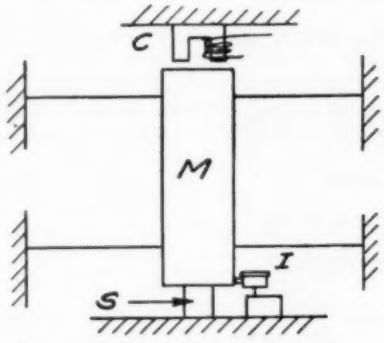


Fig. 15 Drift-Testing Apparatus

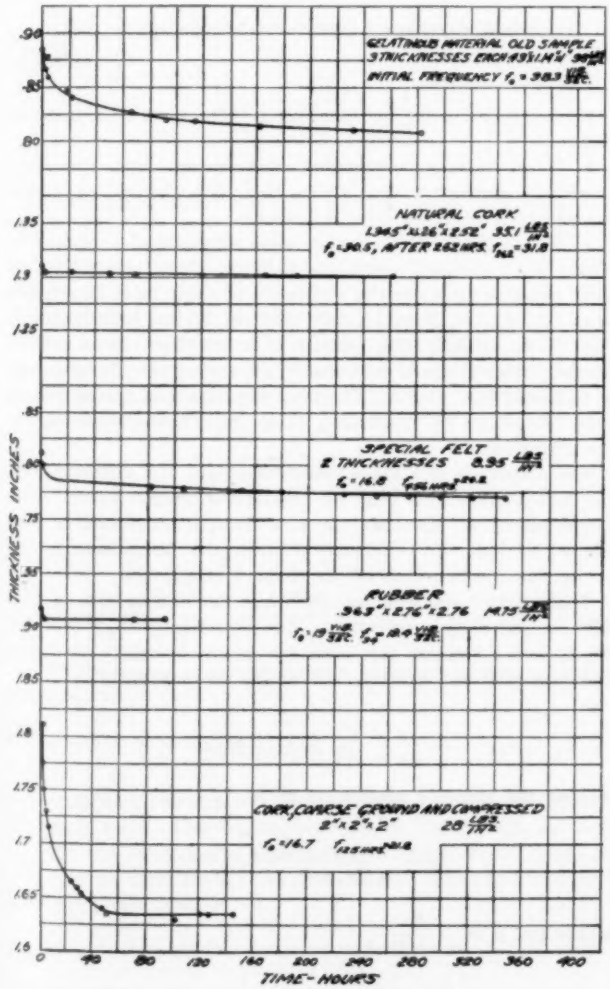


Fig. 16 Drift-Test Results

Samples were proportioned to give about the same load per unit area as was found necessary for satisfactory transmissibility.

The upper curve of Fig. 16 shows the loss in thickness with time for a six-months-old sample of the gelatinous material, which had become 1.6 times as stiff as the newer sample. Even at this high loading and consequent rapid drift, the resonant frequency is too high for proper isolation.

Natural cork shows a small drift, but the resonant frequency is high. Two thicknesses should be used to reduce this frequency and obtain better isolation, in spite of the fact that the drift would be doubled in so doing.

Two thicknesses only of the special felt were tested. Even with this low loading, the drift is great and the stiffening with time fairly large. If the test had been made with four thicknesses the drift would have been twice as large and the percentage change in stiffness the same.

Rubber, at this low loading, shows practically no drift, although at higher loadings some loss in thickness would begin to appear.

Ground cork, when loaded to a value just below the manufacturer's recommended range, shows a very large initial drift. The frequency increase is considerable also.

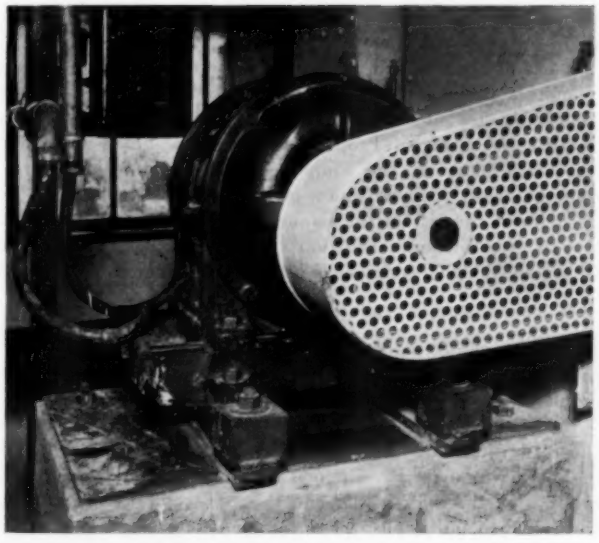


Fig. 17 Photograph of an Installation

Rubber in a Motor Support

In choosing the elastic material for mountings several considerations point to rubber as the most advantageous. Its damping is low, much energy may be stored in a small volume, protection from oil can be arranged, stiffness in various directions may be controlled by shape, drift is small at low pressures, and finally, its life, when compounded with antioxidants, is fairly long.

As an example of the use of rubber, Fig. 17 shows a three-phase induction motor driving a ventilating fan in an office building. The sliding base under the motor is isolated from the foundation by rubber placed in the four symmetrically disposed metal boxes, two of which are visible in the foreground. An induction motor may have vibration components along each of its three coordinate axes, therefore the mounting must be elastic in all directions. A motor on such a mounting has six modes of vibration, and hence six natural frequencies, all of which should fall outside the running range of the motor. Since a floor is most sensitive to vibration in a vertical direction, the natural frequency of the motor on its mounting must be low in that direction. The fundamental frequency of the impressed force is 120 cycles per sec. for a 60-cycle power supply, hence the vertical natural frequency referred to above should be about 15–20 cycles per sec. The mounting shown isolated

the motor very satisfactorily, and also appeared sufficiently stable for good belt operation.

It has not been found possible to make the mounting stiffness of motors transmitting torque to apparatus on separate bases sufficiently low to isolate from unbalance vibration unless the motor speed is high. Proper mechanical balancing will nullify this disturbance at the source. In machines which share a common base, such as motor-generator sets, the natural frequencies of the set on the mounting may be made sufficiently low to give adequate isolation from the unbalance vibration. Of course all higher frequencies are isolated even more effectively.

Summary

Supports can be made to prevent the transmission of vibration to or from the mounted machine through its base, but cannot reduce the noise radiated directly to the air from the machine.

In an elastically supported machine the speed ratio, that is, the ratio of the disturbing frequency to the natural frequency of the machine on its support, must be greater than 1.41 in order that the support be effective. For vibrations, such as those due to unbalance, having a frequency below the audible range, a speed ratio of 2 or 3 is sufficient. Vibration occurring at frequencies in the audible range, due to electrical disturbances for instance, requires a speed ratio of 6 or 8 for proper isolation.

Damping has been shown, both by theory and experiment, to be detrimental to this type of support. For comparison purposes, damping has been assumed to follow the viscous-friction law. A table of δ, the logarithmic decrement of the material, is given for several substances in order that such comparisons may be made. All tests were carried out at values of pressure and frequency approximating service conditions.

Several considerations point to rubber as the material best suited to supports of the type described.

In conclusion the author wishes to acknowledge the suggestions and assistance given by Messrs. A. L. Kimball and W. C. Stewart of the Research Laboratory of the General Electric Company.

Appendix

TRANSMISSIBILITY OF VARIOUS SYSTEMS OBTAINED BY THE MECHANICAL-IMPEDANCE METHOD

MECHANICAL vibration problems may be solved for the steady-state conditions by a complex quantity method analogous to that used in solving electrical circuits.[4,5,6] The electrical equivalents of mechanical quantities are tabulated below.

Electromechanical Equivalents

Mechanical		Electrical	
v	velocity	i	current
F	force	E	voltage
Z	velocity impedance	Z	impedance
x	amplitude	q	charge
k	stiffness	$\frac{1}{C}$	$\frac{1}{\text{capacity}}$
b	friction coefficient	R	resistance
Z_m	mechanical impedance	$j\omega Z$	

$$j = \sqrt{-1}$$

Ohm's law states that

$$i = \frac{E}{Z} \qquad [1]$$

and analogously

$$v = \frac{F}{Z} \qquad [2]$$

In our problem the instantaneous values[7] of displacement and velocity are

$$x_i = x \sin \omega t$$

and

$$v_i = \frac{dx}{dt} = x\omega \cos \omega t$$

Returning to maximum values,

$$v = j\omega x$$

Substituting in [2] and solving for x,

$$x = \frac{F}{j\omega Z} = \frac{F}{Z_m} \qquad [3]$$

Mechanical impedance, Z_m, is found from the following known constants of the system in question:

Frictional impedance (proportional to velocity)	$= jwb$
Mass impedance	$= -M\omega^2$
Elastic impedance	$= k$

Three general rules for combining these quantities may be given:

1 When various components of a system have the same amplitude impressed upon them, their impedances are added directly to obtain the total impedance.

2 If it is possible for two or more elements of a system to divide an impressed amplitude between them, depending on their impedances, then the reciprocal of their total impedance equals the sum of the reciprocals of the individual impedances.

3 When dealing with systems of more than one degree of freedom, impedances of each part of the system are first obtained as indicated in 1 and 2. The total impedance of the system is then discovered by combining these impedances according to the above rules.

The amplitude of a mass means, of course, the distance from the mid-position to a peak on the time vibration curve. The amplitude of an elasticity, or of a damping medium, is considered as half the difference between the displacements impressed on the two ends of the element in question.

This method will first be applied to the simple case of Fig. 3(*b*) by way of demonstration. The sinusoidal force F vibrates M, k, and b through the same amplitude, therefore their impedances will add directly. From [3] we may write

$$x = \frac{F}{jb\omega + k - M\omega^2} \qquad [4]$$

Taking the absolute magnitude gives

$$x = \frac{F}{\sqrt{b^2\omega^2 + (k - M\omega^2)^2}}$$

where $\tan \phi = \dfrac{b\omega}{k - M\omega^2}$

Returning to the problem of the paper, the transmissibility ϵ of this system will be given by the ratio of the maximum force passing through to the rigid foundation to the maximum value of the impressed force f/F.

[4] A. G. Webster, Proc. Am. Acad. of Science, vol. 5 (1919), p. 275.

[5] A. E. Kennelly and Nukiyama, Trans. A.I.E.E., vol. 38 (1919), pp. 651–699.

[6] C. A. Nickle, Trans. A.I.E.E., vol. 44 (1925), p. 1277.

[7] I. B. Crandall, "Theory of Vibrating Systems and Sound," 1926, p. 9.

$$f = x(j\omega b + k)$$

Solving [4] for F and taking the ratio f/F gives

$$\epsilon = \frac{f}{F} = \frac{x(j\omega b + k)}{x(j\omega b + k - M\omega^2)} = \sqrt{\frac{b^2\omega^2 + k^2}{b^2\omega^2 + (k - M\omega^2)^2}}. \quad [5]$$

In the system of Fig. 3(a) in which the foundation is moved through a constant amplitude y, causing a certain amplitude x to appear in M, the transmissibility will be given by the ratio x/y. A certain force F is required to move the foundation ends of k and b through the amplitude y. This same force is applied to M through k and b. Therefore from [3] we may write

$$x = \frac{F}{Z_m(\text{mass})}, \quad y = \frac{F}{Z_m(\text{total})}$$

$$\epsilon = \frac{x}{y} = \frac{Z_m(\text{total})}{Z_m(\text{mass})}$$

$$Z_m(\text{mass}) = -M\omega^2$$

From rule 2,

$$\frac{1}{Z_m\ (\text{total})} = \frac{1}{-M\omega^2} + \frac{1}{k + j\omega b}$$

whence

$$\epsilon = \frac{x}{y} = \frac{\dfrac{-M\omega^2(k + j\omega b)}{j\omega b + k - M\omega^2}}{-M\omega^2}$$

Canceling and taking the ratio of the magnitude of the vectors,

$$\epsilon = \frac{x}{y} = \sqrt{\frac{b^2\omega^2 + k^2}{b^2\omega^2 + (k - M\omega^2)^2}} \quad \ldots\ldots [6]$$

Equations [5] and [6] are identical.[8]

The transmissibility of the two-degree-of-freedom system shown in Fig. 13(a) is x/z. Assuming the displacements x, y, and z positive, also that $x > y > z$, and, for brevity, that $b = 0$, we have from [3],

$$x = \frac{-k_2(x - y)}{-M_2\omega^2} \quad \ldots\ldots [11]$$

$$y = \frac{-k_1(y - z)}{Z'} \quad \ldots\ldots [12]$$

$$Z'' = \frac{-k_2M_2\omega^2}{k_2 - M_2\omega^2}, \qquad Z' = \frac{M_1M_2\omega^4 - k_2M_1\omega^2 - k_2M_2\omega^2}{k_2 - M_2\omega^2}$$

Solving [12] for y, substituting in [11] and bringing z to the left-hand side,

$$\epsilon = \frac{x}{z} = \frac{k_1k_2}{k_1k_2 - \omega^2(k_1M_2 + k_2M_1 + k_2M_2) + M_1M_2\omega^4}. \quad [13]$$

To simplify, the expression may be divided through by the constants k_1k_2. The resonant speeds of the system, ω_c' and ω_c'', are the roots of the denominator when it is set equal to zero. Then breaking up the denominator, using the sum and product of the roots, gives

$$\epsilon = \frac{1}{\left[\left(\frac{\omega}{\omega_c'}\right)^2 - 1\right]\left[\left(\frac{\omega}{\omega_c''}\right)^2 - 1\right]} \quad \ldots\ldots [14]$$

An identical expression may be obtained for the transmissibility of the system shown in Fig. 13(b). Damping will increase the transmissibility in either case.

[8] C. R. Soderberg, *Elec. Jl.*, vol. 21 (1924), pp. 160–165.

Discussion

STUART H. HAHN.[9] The writer is much interested in hearing of the investigation that the author has been making, especially so because this work appears to antedate a somewhat similar investigation with which he has been connected for some months past. Thus the user of a material has, as often before, anticipated its manufacturer in a study of some of its properties.

The author is to be congratulated upon having designed and built a machine for general testing of elastic materials as mountings for the isolation of vibrating machines. The design is simple and the method of use is quite direct. So far as may be judged from the paper itself, however, the author appears to have neglected to consider entirely what we may term the form factor for comparatively incompressible materials which plays an important part in determining the properties of an elastic mounting. It is well known that rubber and more or less similar elastic materials, such as the gelatinous substance tested by the author, are only slightly compressible, so that when subjected to compression, as in a machine mounting, the material must bulge at the sides, undergoing tensile strains in planes perpendicular to the axis of compression. The elastic properties of a block of such material are thus dependent on more than the two factors of unit loading and thickness mentioned by the author; they depend very markedly upon the configuration and extent of the surface area. This discussion, of course, leaves out reference to other than solid elastic soft bodies with sides perpendicular to the loaded face.

Thus the author's figure of about 14 lb. per sq. in. for rubber may be applied only to blocks of such shape and composition as that used by him in his test set-up. In general, however, our work has pointed to loading figures for the various materials tested by the author, of about the same order of magnitude as the values he found most suitable for isolation of high-frequency vibrations. Our work, although so far performed on an entirely different scale from that adopted by the author, also indicates that damping factors of the various materials may be arranged in the same order as he has shown in his Table 1. Our work also corroborates his findings on the so-called gelatinous substance, which is, it is believed, a vulcanized vegetable oil. This material shows exceptionally high hysteresis and was of practically no use as an elastic mounting in the thickness recommended by its manufacturers.

Rubber has been used as an elastic support in many forms for at least the past hundred years. It has long had a few special uses such as for the rubber buffers and springs in European railroad equipment (in which it is said to have longer life than steel springs) and for the landing gears of aeroplanes. This use has been developed widely in the past ten years with its general adoption as an elastic mounting for motors and other parts of automobiles. Its application to motor supports has been studied only by somewhat empirical methods, and as a result there are almost as many rubber mounting designs as there are manufacturers. One of the most important types of motor mounts is that in which the rubber is stressed in shear. The shear mounting often has a considerable advantage of compactness for a given value of the elastic constant.

The damping factor given by the author in Table 1 is of course a characteristic only of the particular rubber compounds which he tested. Both the damping factor and the elastic properties of rubber compounds may, as most engineers realize, be adjusted over quite a wide range by selection of compound. The writer's company has, for instance, made rubber balls for incorporation into a rubber-metal combination universal joint and torsional vibration damper which had such high damping characteristics that they would rebound to a height of only about 10 per cent

[9] Physical Research Laboratory, B. F. Goodrich Company, Akron, Ohio. Jun. A.S.M.E.

of the fall when dropped on a hard surface, as contrasted to a rebound of about 50 per cent for a solid ball of pure rubber composition. The elastic properties of soft rubber may also be varied over almost as wide a range.

The method of studying elastic supports used in the writer's own laboratory has been to apply the support to a fairly large special machine weighing about four tons which is used for punching out small pieces from a wide web of rubber-fabric composition and to measure the building vibrations set up by the heavy impact, in several locations with a sensitive vibrograph of his own design and construction. Other full-scale tests have been and will be run, and laboratory studies on general properties of elastic supports are contemplated.

FRANK M. LEWIS.[10] The author has brought to attention the very interesting and important problem of the elastic mounting of generators. He divides elastic supports into two classes; those designed to protect against shock and those designed to isolate against a single frequency or a relatively small number of frequencies all higher than a certain fundamental.

The author concludes that for mountings of the second type the transmission will be increased with damping and that the best type of elastic support would be one entirely without damping.

The writer is rather hesitant to draw such a conclusion.

The dynamic system which actually exists in the case of a generator or other machine transmitting vibration to a structure is in actuality much more complicated than the simple system postulated by the author in Fig. 3 (*a*) or (*b*). It is assumed in Fig. 3 (*b*) that the structure on which the generator is mounted is of infinite rigidity or mass. If such were actually the case, there would be no need to solve the problem, for no vibration would be transmitted to the structure.

The essence of the problem is the elasticity of the structure and the resonant conditions which may exist in it due to this elasticity.

The actual set-up more nearly resembles that of the author's Fig. 13 (*b*). In this let M_1k_1 represent the elastic system of the building on which the motor M_2 is mounted by the spring k_2. In practice the frequency of the M_1k_1 system alone will be in the vicinity of the disturbing periodic force and is not under control. The frequency of M_2k_2 can be made low as compared with that of the disturbing force.

Assume for the moment no damping in k_1.

If the damping in k_2 were made to approach infinity, it is evident that M_2 and M_1 would vibrate together as a single mass and that a condition of synchronism between the periodic forces and the natural frequency of the elastic system would exist, resulting in excessive vibration. If the damping in k_2 were made zero, the system would now have two natural frequencies. One of these would be relatively low, but the other might be in the vicinity of the frequency of the periodic force, and excessive vibration might again exist. Excessive vibration could then occur if the damping in the machine support was either very large or very small. There must exist, then, some optimum value of damping in the machine support for which a minimum amplitude of vibration would exist in the structure.

This optimum value of damping may be found by setting up the equations for a system of two degrees of freedom, with damping, and finding the damping for minimum vibration in M_1 or the structure. The resulting equations will be, to be sure, rather cumbersome.

In an actual case we will rarely know either the natural frequency or rather frequencies, for there are usually many, or the damping characteristics of the structure on which the generator is mounted. It may therefore not be possible in an actual case to determine the optimum damping, but such an optimum nevertheless exists.

In actual practice the writer believes that the difference in the results obtained with various mounting substances, provided that the elasticities of the mountings are the same, may be very small.

From the viewpoint of the suppression of noise directly from the machine frame, the writer believes that the mounting having a relatively high damping might prove superior.

The question of the merits of damping in machine mountings is by no means settled, and it is hoped that it will continue to receive the author's attention.

J. ORMONDROYD.[11] Spring mounting for the elimination of the evil effects of vibration has been a thing very attractive to engineers over a long period of time. Mr. Akimoff discussed spring mounting in general terms in papers published by the A.S.M.E. before the war. The condition of the art at that time was such that spring mounting made the job worse as often as it made it better.

Mr. C. R. Soderberg brought spring mounting under scientific control by work done in 1923 and 1924, published in *The Electric Journal* at that time. In these papers he has defined the transmission factor ϵ and pointed out the fact that, above a certain speed, ratio damping was harmful rather than good. His definitions of transmission factor were the same as those given in this paper. The author's valuable contribution lies in the laboratory test equipment and in the test curves from this equipment.

The drift test is especially valuable in choosing material for flexible mounting, as it is very important that the material should not become progressively stiffer. It would be interesting to find from the author how the drift characteristics, changing thickness, and changing natural frequencies vary with the specific weights of the materials tested. Several years ago the writer had many static tests made on felts, corks, and rubber in which hysteresis loops were found. It was found that the results could be classified best for a given type of material by using the specific weight of the material. Since the specific weight of the material must be intimately connected with the internal structure, it must play an important role in the elastic, plastic, and damping properties of the material.

The author's test measurements were of such a nature that he should be able to give the energy dissipation per cycle for the materials tested. This should be given supplementing the logarithmic decrement figures given by him.

AUTHOR'S CLOSURE

The behavior of rubber is as Mr. Hahn has stated; that is, the stiffness of a pad of given thickness does not increase as the first power of the area, but at some higher power. A very short discussion of this phenomena, including two curves, has been included in a previously presented paper.[12] Discussion of this phase of rubber in elastic supports was purposely omitted as being outside the field of the present paper. The 14 lb. per sq. in. must not be taken as a recommended loading for rubber. It is merely the pressure on 1-in.-thick rubber of a certain grade necessary to give the mass used in the reported experiments a desired frequency. Under other conditions the loading required to give the proper frequency may run to over 100 lb. per sq. in.

Mr. Lewis states that in a system of two degrees of freedom

[10] Consulting Engineer, Webb Institute of Naval Architecture, New York, N. Y. Mem. A.S.M.E.

[11] Experimental Division Engineer, Westinghouse Elec. & Mfg. Co., South Philadelphia Works, Philadelphia, Pa.

[12] "Elastic Supports for Isolating Rotating Machinery," Hull and Stewart, presented before the A.I.E.E., Rochester, N. Y., April 29 to May 2, 1931.

an optimum value of damping may be found. In order to investigate this point a fairly typical arrangement was studied, and calculations were made from the result. An induction motor, M_2 of Fig. 13(*b*), weighing about 600 lb., was elastically mounted with a vertical natural frequency at 15 vibrations per

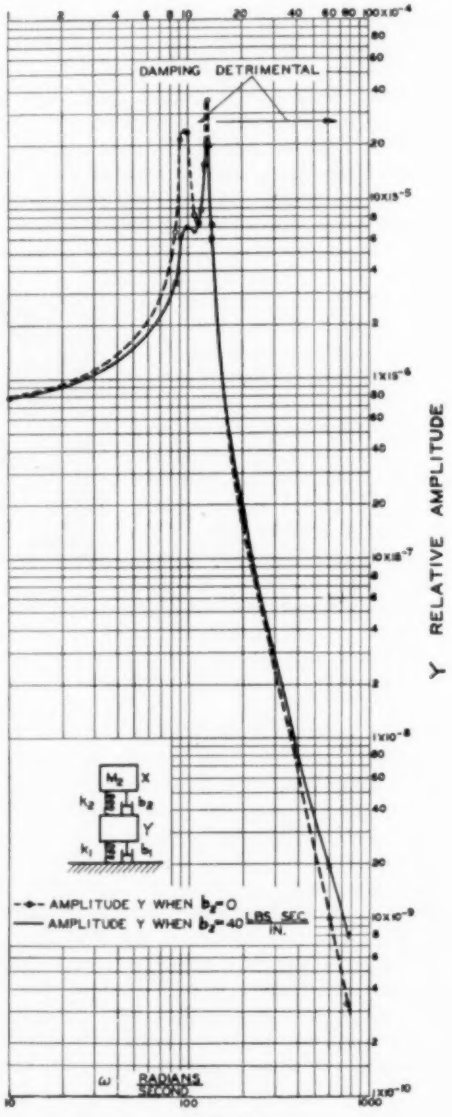


FIG. 18

second on a concrete building floor 9 in. thick having a vertical natural frequency at 20 vibrations per second. The following constants were obtained for the system:

$$M_1 = 80\frac{\text{lb-sec.}^2}{\text{in.}}, \qquad k_1 = 1.3 \times 10^6\frac{\text{lb.}}{\text{in.}}, \qquad b_1 = 250\frac{\text{lb-sec.}}{\text{in.}}$$

$$M_2 = 1.5\frac{\text{lb-sec.}^2}{\text{in.}}, \qquad k_2 = 1.4 \times 10^4\frac{\text{lb.}}{\text{in.}}, \qquad b_2 = 0 \text{ or } 40\frac{\text{lb-sec.}}{\text{in.}}$$

The damping constant for the floor, b_1, was obtained from the value of reinforced concrete, $\delta = 0.075$. That for the support, b_2, is taken for comparison purposes first as zero and then as that due to an elastic foundation constructed of a material having $\delta = 0.8$, such as cork.

Expressions for the motion of the mass M_1, the floor in the foregoing case, were worked out following the method given in the Appendix, and in the last part of the Appendix in particular.

In Fig. 18 the relative amplitude of the floor, M_1, is plotted against impressed frequency for the two values chosen for b_2. These results show that damping is beneficial in two frequency ranges only; from zero up to a point above the first resonant speed, and at the second resonant speed. Damping is detrimental between the two resonant speeds and from a point just above the second resonant speed on up. In this case the base was designed to isolate a vibration of 120 vibrations per second or 753 radians per second from the floor. At this frequency the specified amount of damping in the motor support would cause the floor to vibrate through 2.5 times the amplitude that it would experience with no damping in the motor support.

This effect of damping in a two-degree system is as its action in one-degree systems would indicate; beneficial below the first resonance and at resonance, detrimental at other times. In a closely coupled two-degree system such as the one cited, the frequency of the disturbing force will never be on or below the first resonant frequency of the system, and the second resonant frequency of the system should never be allowed to correspond with the frequency of the disturbing force.

In a loosely coupled system containing, for example, an elastically mounted motor and a wall considerably removed from the motor, the amount of damping in the motor supports will affect the magnitude of the disturbing force transmitted to the structure and hence the amplitude of the wall, but could not affect the motion of the wall, as it would be far removed and small in comparison with other damping forces acting on the wall.

Regarding Mr. Ormondroyd's discussion, the author makes no claim to priority in the theory of elastic supports in systems of one degree of freedom. Mr. Soderberg's work is included in the references.

Referring to the suggestion that similar materials may be classified by density with regard to drift and changing stiffness under load, rubber and the gelatinous material may be taken as an example. The specific weight of this sample of rubber is 0.0381 lb. per cu. in. and that of the gelatinous substance 0.0482 lb. per cu. in., which would indicate that the latter material would drift less under similar conditions, thus leading to an erroneous result.

In some cases the more dense samples of rubber have been found to give greater drift under load than those having a smaller density.

On the sixth page of the paper is a short discussion showing that the solid friction law, in which the internal friction loss per cycle is independent of frequency, does not hold for the type of substance tested. However, the viscous friction law in which the friction loss is proportional to velocity does give a fair idea of the damping effect. In this case the energy dissipation per cycle depends, among other factors, on velocity, so that it cannot be stated as a constant for the material.

APM-53-13

Thermal Stresses in Thin-Walled Cylinders

By CLARENCE H. KENT,[1] FAYETTEVILLE, ARK.

The author, using Föppl's method, investigates the stresses in (1) a cast-iron cylinder 30 in. long, 20 in. mean diameter, and 1 in. in wall thickness, with the temperature 100 deg. fahr. higher inside than outside; (2) the same cylinder and temperature conditions, but with cylinder supported at the ends; and (3) a steel cylinder of same size with flat heads rigidly attached and subjected to 500 lb. per sq. in. internal pressure. Constant and variable temperature gradients are considered, both in axial and radial directions.



THE problem of the stresses in a long, hollow cylinder caused by a temperature gradient in the radial direction was first solved by T. M. C. Duhamel (1)[2] in 1838. The solution was presented in a memoir which laid the foundation for future study of thermal stresses. Later writers have brought the problem to its present-day form (5), (6), (10). The method is exact for an infinitely long cylinder of any thickness and any temperature distribution in the radial direction only. It finds the principal planes of stress in such a cylinder to be in the radial, axial, and tangential directions with normal stresses acting on each, and shows that stresses on a plane perpendicular to the axis are independent of the position of this plane.

In order that the solution may be exact when a cylinder of finite length is chosen, boundary conditions require that external forces equal in value and distribution to the stresses in the axial direction act on the ends. In the usual case such forces will not be present, so that the stresses near the ends will be very different from those calculated by this method. The effect diminishes as we recede from the ends, and practically disappears at a point sufficiently far away. Some writers have assumed that this point was located at a distance from the ends equal to the thickness of the cylinder wall, but its location really depends on the ratio of this thickness to the cylinder diameter. For such examples as the liner of a Diesel-engine cylinder this effect does not disappear until we have moved from the ends a distance equal to about twelve times the wall thickness. Since the tangential stresses at the ends may be as much as 35 per cent higher than those near the middle, it becomes important to investigate them.

For determining the value of these stresses in thin cylinders Dr. August Föppl (21) turned to a method which had been in use for some time in the case of cylinders subjected to internal or external pressure. This was to treat an element cut from the cylinder wall as a beam on an elastic foundation. The method is not exact because it disregards radial stress, but it is sufficiently accurate for cases in which the cylinder diameter is at least ten times the wall thickness.

Applying the method first to a cylindrical shell of finite length in which a temperature gradient exists in the direction of the radius only, it is found that the maximum tangential stress at the ends is 30 per cent higher than the similar stress in an infinite cylinder. In a cast-iron cylinder of 1 in. thickness, 20 in. mean diameter, and 30 in. length, the tangential stress at the ends produced by a temperature difference between inner and outer surfaces of 100 deg. fahr. is 7850 lb. per sq. in. Away from the ends the value of both the tangential and axial stresses is 6000 lb. per sq. in., which is the same as that in a thin plate subjected to the same temperature gradient through its thickness and prevented from bending. This stress is independent of the thickness of the plate, which seems to contradict experience, as it is usually considered that a thicker plate will be subjected to higher thermal stresses. For the same rate of *heat flow* this is true, but in that case the temperature difference will be higher across a thick plate than across a thin one, and thus give rise to higher stress. The cylinder walls are deflected outward 0.0022 in. at the ends, which is the cause of the increase in tangential stress at this point. Fig. 6 shows the deflection of the walls and the distribution of tangential and axial stresses in the exterior surface.

The actual curve assumed by the cylinder walls has the form of a damped wave which is identical with that of a beam on an elastic foundation acted upon by a moment at the ends. In the case of the cylinder a small longitudinal strip of the walls represents the beam, and its elastic support is furnished by the resistance of the wall as a whole to expansion or contraction. Fig. 6 shows clearly how the moment produced by the thermal stresses has bent the end of the wall outward, and how this effect is damped out as one moves away from the ends.

If the deflection at the ends of such a shell is prevented by rigid supports such as those shown in Fig. 8, tangential stresses will be reduced. In the case of the shell just considered this reduction amounts to 43 per cent, as the curves in Fig. 9 indicate. It is noticed that the axial and tangential stresses reach a constant value of 6000 lb. per sq. in. at a point about 12 in. from the ends in both Figs. 6 and 9.

The supports at the ends do not entirely eliminate deflection, but they reduce its maximum value to less than 0.001 in. As the maximum axial stress is only 7 per cent above the normal "thin plate" value, one may conclude that for all practical purposes a cylinder so supported may be treated as of infinite length. The stresses in the wall will then be the same as those in the thin plate. Although the method does not apply to a thick cylinder, we may reason by analogy that the usual stress formulas for an infinite cylinder may be used with an equal degree of accuracy for a finite cylinder provided with similar end supports.

---

[1] Head of Department of Mechanical Engineering, University of Arkansas. Mem. A.S.M.E. Professor Kent was graduated from Purdue University in 1915. His experience comprises three years with the Erie R. R.; two years as 1st Lieut. of Engineers, U. S. Army; three years with the Westinghouse Elec. & Mfg. Co.; eight years as associate professor of Mechanical Engineering, University of Nevada; and, since 1928, as professor and head of department of mechanical engineering at the University of Arkansas. He has received the degrees of M.E. from Purdue (1927) and of M.S. from Iowa State College (1928).

[2] Numbers in parentheses indicate references in the bibliography following Appendix No. 2.

Presented at the National Applied Mechanics Meeting, Purdue University, Lafayette, Ind., June 15 and 16, 1931, of The American Society of Mechanical Engineers. [Selected from a dissertation prepared under the direction of Prof. S. Timoshenko for presentation to the Faculty of the University of Michigan in partial fulfilment of the requirements for the degree of Doctor of Philosophy.]

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

The stress will be reduced, of course, by any support whether or not it is rigid, and its effect can be computed if its elastic properties are known. Obviously the stress at the ends under such conditions will fall somewhere between the 7850 lb. per sq. in. value of Fig. 6 and that of 4500 lb. per sq. in. of Fig. 9, with the values for the middle portion of the cylinder remaining at 6000 lb.

In Example 3 the method is applied to a steel cylinder of the same dimensions and having the same temperature difference as the cast-iron cylinders considered. We now assume that heads of the same material as the walls are rigidly attached; that is, there is no bending in the joint between the head and the walls. An internal fluid pressure of 500 lb. per sq. in. is now introduced, though to avoid unnecessary complications in the calculations no opening is assumed. Such an opening should be placed somewhere within the middle 10 in. of the wall, where the cylindrical form is not interfered with by the application of heat or pressure.

The head applies both a moment and a shear to the wall whose value is calculated from the slope of the head at the joint and its expansion in a radial direction under load. The head tends to be deflected outward by the pressure, but inward by the temperature strain and action of the walls. The tensile stress along any radius in the exterior of the head reaches a maximum of 28,000 lb. per sq. in. at the center for a flat head $1^1/_2$ in. thick.

The curves of Fig. 11 indicate a complex stress distribution in the walls. The internal axial stress which normally is compression is changed to tension at the ends by the moment of the cylinder heads and reaches a maximum value of 26,550 lb. per sq. in. The tangential stresses at the ends are kept low by the action of the heads in limiting deflection.

At the middle of the cylinder the tangential tension reaches a maximum of 20,300 lb. per sq. in. on the exterior, 5000 lb. of which is produced by the internal pressure and the rest is due to the temperature difference. Axial stresses here are lower since the internal pressure has a smaller effect on them. The deflection curve is a damped wave shifted from its normal position by the action of the moment and shear at the origin. At the middle the effect of the internal pressure is to prevent the deflection returning to zero as it did in previous cases.

This example is intended to illustrate the fact that a great variety of end conditions may be successfully handled by this method, as long as there is a means of arriving at an expression for the moment and shear exerted on the cylinder walls at the ends. A flanged connection at the ends would also afford a good example of this.

So far the discussion has been concerned with shells having a temperature gradient in the radial direction only. It happens in many applications that temperature gradients also exist in the axial direction. In an internal-combustion-engine cylinder, for example, heat is flowing radially through the walls, but the temperature of the interior varies at the same time in the axial direction.

A case of very localized heating, such as that which is produced in welding pipe, develops a large gradient in the axial direction. The method used in this paper is quite easily extended to cases where the temperature along the axis varies with integral powers of the axial coordinate. This gives sufficient latitude for handling most practical cases.

Suppose in the cast-iron cylinder of the first example, the interior temperature varies linearly along the axis from zero to 100 deg. fahr., while the exterior is held at zero. Since the cylinder walls are thin, a linear distribution can still be assumed along the radius. Fig. 15 shows the deflection and the stress distribution for this case. The maximum stress at the warmer end is slightly less than that of Fig. 6, and the maximum axial stress is some 1400 lb. per sq. in. less.

Any constant temperature difference may be superposed directly on the curves of Fig. 15 to find the combined effect. If, for example, the internal temperature varies linearly over the length from 100 deg. fahr. to 200 deg. fahr. while the exterior is kept at zero, the stress and deflection curves are obtained by adding the curves of Fig. 6 to those of Fig. 15.

If the temperature varies with the second power of the axial coordinate instead of the first, the problem may be handled in the same way. The stress curves obtained would differ but little from those of Fig. 15 at the end sections. The middle portion would be represented by a parabola instead of a straight line.

A case of this type of temperature distribution occurs in the cylinder liner of a Diesel engine. From experimental measurements of temperatures obtained by Dr. Eichelberg (11) and Dr. Letson (12) the temperature curves of Fig. 16 were drawn. The measurements were made in each case by thermocouples embedded in the liner at a distance of 5 mm. from each surface. The equations of the curves are of the second degree in x, and the stresses produced may be calculated by this method. In Fig. 16 the liner is shown pressed into its casing, which is assumed to exert a sufficient moment and shear on the liner to bring the deflection and slope at this end to zero.

The maximum stress of approximately 6500 lb. per sq. in. occurs at the end. As we have seen before, when the deflection of the end is prevented the cylinder behaves much as if it were infinite in length. The stresses obtained by the usual formulas applying to infinite cylinders are about 150 lb. per sq. in. below the value just given. The actual stress will depend upon the exact amount of the deflection permitted at the end. The assumptions, first of free ends and then of fixed ends, establish the limits between which any particular case might fall. Normally the end conditions will be much nearer to the fixed case.

In the example chosen the sharp peak of stress at the origin could be much reduced by designing the press fit of the liner in its casing to permit an outward slope of about 0.001 in. per inch at the origin and to keep the deflection of the wall beyond its normal position as low as possible. To see the effect of this, imagine that the maximum tangential stress in Fig. 15 is reduced to about that at the origin in Fig. 9. If the moment is removed the axial stresses will be reduced to zero at the end.

Notation

a = linear expansion coefficient
c = mean radius of cylindrical shell
E = modulus of elasticity
e_t = unit tangential deformation
h = thickness of cylindrical shell
I = moment of inertia of strip of unit width and depth h
K = modulus of foundation
l = length of cylinder
M = moment
p = pressure, lb. per sq. in.
r = radius of curvature
σ_r = stress in radial direction
σ_t = stress in circumferential direction
σ_x = stress in axial direction
T = temperature
ΔT = temperature difference between interior and exterior surfaces of cylinder
η = distance from neutral axis
μ = Poisson's ratio
$\phi; \psi; \theta; \alpha$ = functions of x, see Appendix No. 1.

Heat Flow

If the interior surface of a hollow cylinder be maintained at a temperature T_1, and the exterior at a temperature T_2, flow of heat takes place through the walls which after a sufficient time becomes a steady flow. The temperature of any point in the wall is then given by the solution of the differential equation:[3]

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} = 0$$

where T is the temperature of any point whose distance from the axis is r.

The solution is:

$$T = \frac{(T_2 - T_1)\log r + T_1 \log r_2 - T_2 \log r_1}{\log r_2 - \log r_1}$$

r_1 and r_2 being the inner and outer radii.

This discussion is confined to thin-walled cylinders in which the logarithmic distribution above becomes practically linear and is given by:

$$T = \frac{(T_2 - T_1)r + T_1 r_2 - T_2 r_1}{r_2 - r_1}$$

Let $r_2 - r_1 = h$, and $(r_1 + r_2)/2 = c$ = radius of mid-plane of cylinder wall. Let $\eta = r - c$; thus η is measured from the mid-plane and positive outward.

Let $T_2 - T_1 = \Delta T$, $T_1 = -\Delta T/2$, and $T_2 = +\Delta T/2$; thus the temperature is also measured from the mid-plane and positive outward. Since a uniform heating or cooling of the entire cylinder will have no effect on stresses, we may choose the temperature of the mid-plane as we like. For convenience let its temperature be zero. By these substitutions the temperature equation becomes:

$$T = \frac{\eta}{h}\Delta T$$

when the temperature of the external surface is higher than that of the internal. When this condition is reversed and the heat flow is away from the center the temperature of any point is given by:

$$T = -\frac{\eta}{h}\Delta T \qquad [1]$$

This expression is the same as that for a flat plate of thickness h subjected to a constant temperature difference ΔT.

It is to be noted that neither the coefficient of linear expansion a nor the modulus of elasticity E remains constant when the temperature varies. For temperature differences not exceeding 200 deg. fahr., however, this variation will not seriously affect the result. One may express a as a function of the temperature and thus take care of its variation for temperature differences exceeding that mentioned (9). It is questionable whether the introduction of this complexity is warranted in view of the fact that an error of the same order, due to the variation of E, is still present. For purposes of this work it will be assumed that a and E are constant, choosing an average value for the temperature range dealt with.

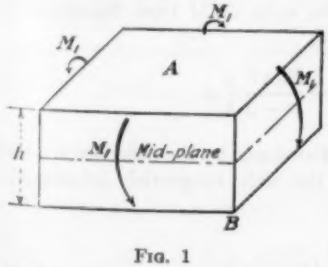


Fig. 1

[3] Carslaw, p. 114.

Constant Temperature Gradient in Radial Direction Only

Consider an element of a flat plate of thickness h and sides unity, shown in Fig. 1. Let a constant temperature difference ΔT be applied to the surfaces A and B so that the temperature of any point is a function of its distance from the mid-plane and is given by Equation [1]. The element will then be bent to a spherical surface of radius:[4]

$$\frac{1}{r} = \frac{a\Delta T}{h} \qquad [2]$$

This curvature will be removed by the application of the moments M_1 in Fig. 1, whose value is:

$$M_1 = -\frac{EI}{1-\mu}\frac{a\Delta T}{h} \qquad [3]$$

The stresses produced by these moments are:

$$\sigma_x = \sigma_y = \frac{Ea\Delta T}{1-\mu}\frac{\eta}{h} \qquad [4]$$

If a similar element is cut from a thin cylindrical shell and heated in the same manner, Equation [2] represents the *change* in curvature which it will experience. The moments required to remove this change and the stresses resulting are given by Equations [3] and [4], except that in the cylinder σ_y is replaced by σ_t.

In order that we may have such a stress distribution in a cylinder of finite length we must assume that at the ends are applied uniformly distributed moments of value M_1 per inch of circumference, as shown in Fig. 2(*a*). In such a cylinder or in one of infinite length the stress distribution due to the application of the temperature difference ΔT is given by Equation [4].

In the usual case the ends of the cylinder are free. To accomplish this we shall have to apply to the cylinder ends a moment equal and opposite to M_1 as shown in Fig. 2(*b*). On the temperature stresses given by Equation [4] we must superpose the stresses produced by the application of the moment $-M_1$, whose value we shall now determine.

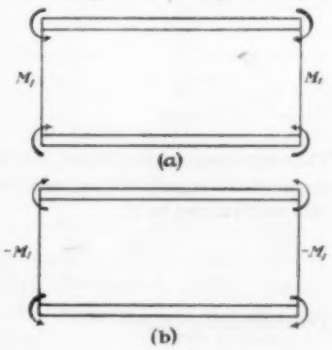


Fig. 2

Consider the cylinder at a uniform temperature which we may take as zero. Let a strip of unit width of the thickness of the cylinder wall be cut from it as in Fig. 3. The moment $-M_1$ acts on the ends of this strip producing a deflection y, which is resisted by the radial component of tangential stresses set up by such movement, as shown in Fig. 4. Since all the strips composing the cylindrical shell are affected alike, there can be no shear between them.

The value of the radial component of these tangential stresses produced by a deflection y, is $(Eh/c^2)y$ per unit length of strip. The moment at any point x in the strip due to these forces can be computed from Fig. 5. If $K = Eh/c^2$, the force acting on the element of length du is $Kydu$ and its lever arm about the point x is $(x - u)$. The summation of moments about the point x gives:

$$M + M_1 + \int_0^x Ky(x-u)du = 0 \qquad [5]$$

[4] See Timoshenko, p. 487.

where M represents the internal moment at the cross-section mn, and is positive in a beam bent concave upward. The relation between moment and deflection in a thin strip is:

$$M = \frac{EI}{1-\mu^2}\frac{d^2y}{dx^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots [6]$$

This modification of the usual bending formula is caused by the prevention of lateral contraction, since no change can take place in the angle θ of Fig. 3. Substituting the values of M and M_1 in [5]:

$$\frac{EI}{1-\mu^2}\frac{d^2y}{dx^2} - \frac{EI}{1-\mu}\frac{a\,\Delta T}{h} + \int_0^x Ky(x-u)du = 0 \ldots [7]$$

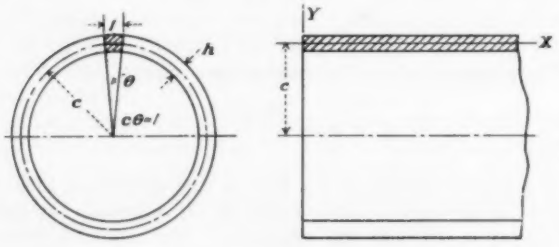


FIG. 3



FIG. 5

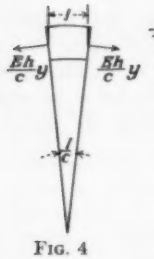


FIG. 4

This equation simply states the law of statics that the sum of moments acting about the point x must be zero. Differentiating this with respect to x:

$$\frac{EI}{1-\mu^2}\frac{d^3y}{dx^3} + \int_0^x Kydx = 0 \quad \ldots\ldots\ldots\ldots [8]$$

Differentiating again:

$$\frac{EI}{1-\mu^2}\frac{d^4y}{dx^4} + Ky = 0 \quad \ldots\ldots\ldots\ldots [9][5]$$

These equations describe a beam on an elastic foundation bent by moments $\frac{EI}{1-\mu}\frac{a\,\Delta T}{h}$ acting on the ends. The solution is:

$$y = e^{\beta x}(C_1 \cos \beta x + C_2 \sin \beta x) + e^{-\beta x}(C_3 \cos \beta x + C_4 \sin \beta x)$$

where $\beta = \sqrt[4]{\frac{K(1-\mu^2)}{4EI}} = \sqrt[4]{\frac{3(1-\mu^2)}{h^2c^2}}$, and C_1, C_2, etc., are constants of integration.

We may dispose of two of these constants at once. As we move away from the ends of the cylinder the deflection of the wall, produced by a moment on the end, should decrease, and when x is large, should be practically zero. Under no circumstances could it show the rapid increase produced by the terms multiplied by $e^{\beta x}$. Therefore we reason: $C_1 = C_2 = 0$. The deflection curve then takes the form:

$$y = e^{-\beta x}(C_3 \cos \beta x + C_4 \sin \beta x) \quad \ldots\ldots\ldots [10]$$

The analysis is valid for a cylinder of any length up to Equation [10]. In placing C_1 and C_2 equal to zero it was assumed that the cylinder was long enough to permit the deflection and slope of the wall to become practically zero at the middle. Specific examples will show what this length must be, but in general a distance equal to $2\pi/\beta$, which is the wave length of the harmonic form assumed by y will be sufficient.

To evaluate the constants C_3 and C_4, we may compute from [10] the values of d^2y/dx^2 and d^3y/dx^3 for $x = 0$ and substitute them in Equations [7] and [8]. By differentiation we find:

$$\frac{d^2y}{dx^2}_{(x=0)} = -2\beta^2 C_4$$

Placing $x = 0$ in [7] and substituting this value for d^2y/dx^2

$$C_4 = -\frac{1+\mu}{2\beta^2}\frac{a\,\Delta T}{h}$$

By differentiation again, we obtain:

$$\frac{d^3y}{dx^3}_{(x=0)} = 2\beta^3(C_3 + C_4)$$

and from Equation [8]:

$$C_3 = \frac{1+\mu}{2\beta^2}\frac{a\,\Delta T}{h}$$

Putting these values in [10],

$$y = \frac{1+\mu}{2\beta^2}\frac{a\,\Delta T}{h}e^{-\beta x}(\cos \beta x - \sin \beta x) = \frac{1+\mu}{2\beta^2}\frac{a\,\Delta T}{h}\psi \quad [11]$$

where $\psi = e^{-\beta x}(\cos \beta x - \sin \beta x)$

and:

$$\frac{d^2y}{dx^2} = (1+\mu)\frac{a\,\Delta T}{h}\phi \quad \ldots\ldots\ldots\ldots [12]$$

where $\phi = e^{-\beta x}(\cos \beta x + \sin \beta x)$.

As noted above, $l/2$ is assumed large enough to permit the effect of the moment on the ends to disappear at the middle of the cylinder. When both ends of the cylinder are free, the deflection, moment, stresses, etc., are symmetrical about the point $x = l/2$.

The axial stress in the shell owing to the deflection y is:

$$\sigma_x' = -\frac{M\eta}{I}$$

We may substitute in this the value of M from Equation [6], and of d^2y/dx^2 from [12], giving

$$\sigma_x' = -\frac{Ea\,\Delta T}{1-\mu}\frac{\eta}{h}\phi$$

To find the tangential stresses produced by the deflection we note that $e_t = y/c$, where e_t is the unit tangential deformation. From Hooke's law,

$$e_t = \frac{1}{E}(\sigma_t' - \mu\,\sigma_x')$$

[5] An alternate derivation of this formula is given in Appendix No. 2.

Substituting the values of y and σ_x':

$$\sigma_t' = \frac{Ea\,\Delta T}{h(1-\mu)}\left[\frac{1-\mu^2}{2\beta^2 c}\,\psi - \eta\mu\phi\right]$$

These stresses are due to the deflection y alone. We may now refer to Equation [4] and superpose the stresses existing in the shell owing to the temperature gradient to obtain the complete expressions for the stresses in a cylindrical shell, heated in the manner assumed and having ends free from external forces:

$$\sigma_x = \frac{Ea\,\Delta T}{1-\mu}\,\frac{\eta}{h}\,(1-\phi) \qquad [13]$$

$$\sigma_t = \frac{Ea\,\Delta T}{1-\mu}\,\frac{\eta}{h}\left[1 - \mu\phi + \frac{1-\mu^2}{2\beta^2 c\eta}\,\psi\right] \qquad [14]$$

Since $\phi = 1$ for $x = 0$, this expression for σ_x satisfies the condition of free ends. When $\eta = h/2$

$$\sigma_x = \frac{Ea\,\Delta T}{2(1-\mu)}\,(1-\phi)$$

$$\sigma_t = \frac{Ea\,\Delta T}{2(1-\mu)}\left[1 - \mu\phi + \frac{1-\mu^2}{\beta^2 hc}\,\psi\right]$$

when $x = 0$, max. $\sigma_t = \dfrac{Ea\,\Delta T}{2}\left(1 + \dfrac{1+\mu}{\sqrt{3(1-\mu^2)}}\right)$ [15]

when x is large, $\phi = \psi = 0$, and

$$\sigma_x = \sigma_t = \frac{Ea\,\Delta T}{2(1-\mu)} \qquad [4]$$

It may be shown in the usual way that the maximum value of σ_t occurs when $x = 0$ and $\eta = h/2$. The absolute value is less for $\eta = -h/2$.

If we assume $\mu = 0.25$, we may compare the value of σ_t at the ends with that for the region near the middle. Equation [4] then gives for the exterior surface, when x is large:

$$\sigma_t = \frac{Ea\,\Delta T}{2}\,(1.333)$$

For a similar point at the ends, Equation [15] gives:

$$\text{max. } \sigma_t = \frac{Ea\,\Delta T}{2}\,(1.744)$$

The peak stress at the ends is thus 30 per cent higher than the value near the middle.

The maximum value of σ_x occurs at $x = \pi/\beta$. Since the minimum value of $\phi = -0.043$, by referring to Fig. 17, the maximum axial stress will be only 4 per cent above the value given by Equation [4].

Example 1. As an illustration of the preceding section, assume a cast-iron cylinder of 20 in. mean diameter, 1 in. thickness, and 30 in. length. Let the inside temperature be 100 deg. fahr. above the outside. Other constants are:

$E = 15 \times 10^6$ lb. per sq. in. $\quad a = 6 \times 10^{-6}$ in. per in. per deg. fahr.

$\mu = 0.25$ $\quad \beta = \sqrt[4]{\dfrac{3(1-\mu^2)}{h^2c^2}} = 0.41$ in.$^{-1}$

The deflection at the ends is:

$$\text{max. } y = \frac{1+\mu}{2\beta^2}\,\frac{a\,\Delta T}{h} = 2.23 \times 10^{-3} \text{ in.}$$

The tangential and axial stresses away from the ends are given by Equation [4] and are: $\sigma_x = \sigma_t = \dfrac{Ea\,\Delta T}{2(1-\mu)} = 6000$ lb. per sq. in. tension and compression on the outer and inner surfaces, respectively. A formula[6] taking into account the true logarithmic temperature distribution in the cylinder walls and applying only to the portion distant from the ends gives, at the outer surface, $\sigma_t = \sigma_x = 5990$ lb. per sq. in.; and at the inner surface, $\sigma_t = \sigma_x = -6100$ lb. per sq. in. Thus for a ratio of $h/c = 0.1$, Equation [4] is very accurate.

The maximum tangential stress is:

$$\text{max. } \sigma_t = \frac{Ea\,\Delta T}{2}\left(1 + \frac{1+\mu}{\sqrt{3(1-\mu^2)}}\right) = 7850 \text{ lb. per sq. in.}$$

tension on the outer surface at the ends. The maximum axial stress occurs at $\pi/\beta = 7.67$ in. from each end, and its value is:

$$\text{max. } \sigma_x = 6270 \text{ lb. per sq. in.}$$

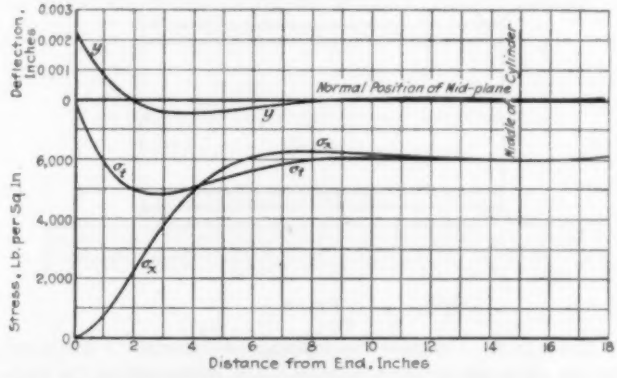


FIG. 6 TENSION IN OUTER FIBERS OF HOLLOW CAST-IRON CYLINDER (Length, 30 in.; diameter, 20 in.; thickness, 1 in. Temperature of internal surface, 100 deg. fahr.; of external surface, 0 deg. fahr.)

Fig. 6 shows the deflection of the cylinder wall from its normal position, and the variation in the tangential and axial stresses in the outer surface over one-half the length of the cylinder.

EFFECT OF CYLINDER HEADS OR OTHER END CONNECTIONS

An advantage of the method given is that it lends itself particularly well to the solution of the problem of the cylinder with end connections or cylinder heads exerting moments and loads on the ends and subjected at the same time to temperature gradients.

Let us again assume the cylindrical shell at a uniform temperature and with the moments and loads M_0 and q_0 per unit of circumference applied at the ends. A unit strip of this shell

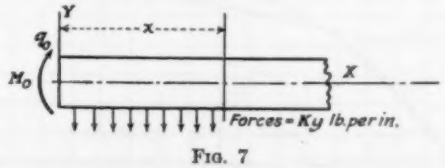


FIG. 7

will then be loaded as in Fig. 7. Equation [10] gives the deflection curve. Evaluating the constants for the end conditions in this case, we have:

$$C_3 = \frac{1-\mu^2}{2\beta^2 EI}\left(M_0 + \frac{q_0}{\beta}\right); \qquad C_4 = -\frac{1-\mu^2}{EI}\,\frac{M_0}{2\beta^2}$$

[6] Timoshenko, p. 554.

and:

$$y = \frac{1-\mu^2}{EI}\frac{1}{2\beta^2}\left(M_0\psi + \frac{q_0}{\beta}\theta\right) \quad \ldots\ldots [16]$$

where, $\theta = e^{-\beta x}\cos\beta x$

From [16] we find:

$$\sigma_x = -\frac{12\eta}{h^3}\left(M_0\phi + \frac{q_0}{\beta}\alpha\right) \quad \ldots\ldots [17]$$

where $\alpha = e^{-\beta x}\sin\beta x$, and

$$\sigma_t = \frac{Ey}{C} + \mu\sigma_x \quad \ldots\ldots [18]$$

On these results will be superposed those in Equations [11], [13], and [14] to obtain the effects of a moment and load M_0 and q_0 combined with temperature difference.

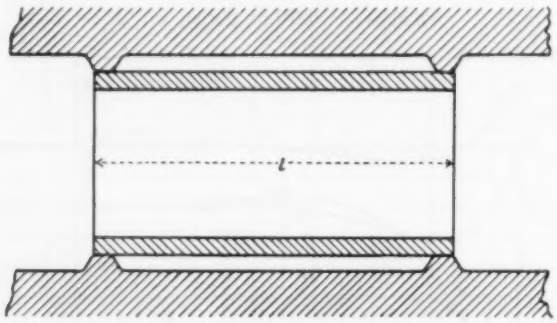


FIG. 8

Example 2. Consider the cylinder of Example 1 subjected to the same temperature difference but supported at the ends as in Fig. 8. Let these supports be sufficiently rigid that we may consider the deflection zero at the ends. Since there is no external moment at the ends, the load q_0 is determined from the fact that the sum of the deflections at the point $x = 0$ given by Equations [11] and [16] is zero:

$$0 = \frac{1+\mu}{2\beta^2}\frac{a\Delta T}{h} + \frac{1-\mu^2}{EI}\frac{q_0}{2\beta^3}$$

and

$$q_0 = -\frac{EI}{1-\mu}\beta\frac{a\Delta T}{h} = -410 \text{ lb. per in. of circumference}$$

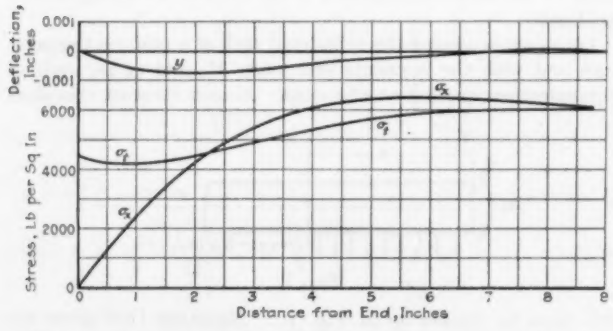


FIG. 9 TENSION IN OUTER FIBERS OF CYLINDER OF FIG. 6 SUPPORTED AS IN FIG. 8

To obtain the complete results for this case we may superpose on the curves of Fig. 6 the results of Equations [16], [17], and [18], in which $M_0 = 0$, and $q_0 = -410$ lb. per in. Fig. 9 shows the deflection and stresses produced in the cylinder walls.

Far from the ends the stresses are the same as those in Fig. 6. At the ends, however, the tangential stress has been reduced by the supports from a maximum of 7850 lb. per sq. in. in Fig. 6 to 4500 lb. per sq. in. in Fig. 9. This reduction depends only on the value of μ, and amounts to 42.7 per cent for $\mu = 0.25$.

Example 3. Assume a steel cylinder of the same dimensions as the cast iron one of Example 1 and heated in the same manner. In addition let it be subjected to an internal pressure of 500 lb. per sq. in. and fitted with flat heads rigidly attached to the walls. The heads are of the same material but h_1 in. thick. The temperature gradient is assumed to exist in the heads as well as the walls. The constants are:

$E = 30 \times 10^6$ lb. per sq. in. $\quad h = 1$ in.
$a = 6.7 \times 10^{-6}$ in. per in. per deg. fahr. $\quad h_1 = 1.5$ in.
$\mu = 0.3$
$\beta = 0.407$ in.$^{-1}$

The deflection and stresses in the cylinder walls owing to the moment and shear at the ends produced by the heads are given in Equations [16], [17], and [18]. To these must be added the effects of the internal pressure, p lb. per sq. in., which produces a constant deflection pc^2/Eh. It increases the axial stress by an amount $pc/2h$, and the tangential stress by double this value. The deflection and stresses due to the temperature gradient may be taken from Equations [11], [13], and [14].

M_0 and q_0 are determined from a consideration of the cylinder head for which the axes are taken as in Fig. 10. The moment

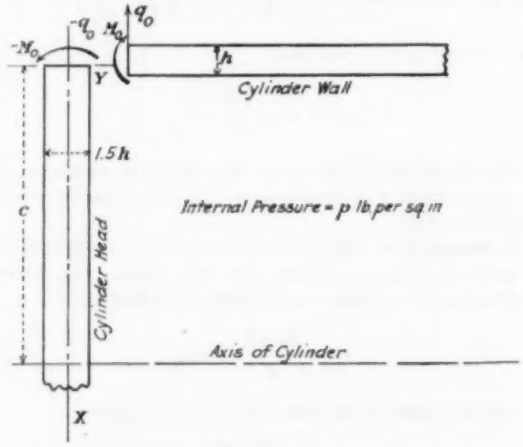


FIG. 10

and load at the edges are equal and opposite to those acting on the wall. The formulas applying to the head, which is treated as a thin plate, may be taken directly from textbooks covering this subject. The extension of the radius of the plate produced by the load $-q$ is $-\frac{q_0 c}{Eh_1}(1-\mu)$, which must be equal to the deflection of the walls at the end when the two are brought together. The wall deflection is given by Equations [11] and [16] plus pc^2/Eh. Equating these two expressions:

$$\frac{1+\mu}{2\beta^2}\frac{a\Delta T}{h} + \frac{pc^2}{Eh} + \frac{1-\mu^2}{EI}\frac{1}{2\beta^3}\left(M_0 + \frac{q_0}{\beta}\right) = -\frac{q_0 c}{Eh_1}(1-\mu) \quad \ldots\ldots [a]$$

The slope of the cylinder head will be influenced by three factors: the internal pressure; the temperature difference; and the moment $-M_0$ at the edges. The slope at the edges owing to the internal pressure is:

$$-\frac{3pc^3(1-\mu)}{2Eh_1^3}$$

That due to the moment is

$$\frac{12\,M_0 c}{Eh_1^3}(1-\mu)$$

That due to the temperature gradient may be determined from the radius of curvature of a thin plate as given in Equation [2] and is: $\frac{a\Delta T}{h_1}c$. The slope of the wall at the ends is determined from the slope of Equations [11] and [16], when $x = 0$:

$$\frac{dy}{dx_{(x=0)}} = -\frac{1-\mu^2}{EI}\frac{1}{2\beta^2}(2\beta M_0 + q_0) - \frac{1+\mu}{\beta}\frac{a\Delta T}{h}$$

When joined, the slope of the wall and the head will be the same:

$$-\frac{1-\mu^2}{EI}\frac{1}{2\beta^2}(2\beta M_0 + q_0) - \frac{1+\mu}{\beta}\frac{a\Delta T}{h} = -\frac{3pc^2(1-\mu)}{2Eh_1^2} + \frac{12M_0 c}{Eh_1^3}(1-\mu) + \frac{a\Delta T}{h_1}c \quad \ldots\ldots [b]$$

Equations [a] and [b] determine M_0 and q_0 as follows:

M_0 = 180 lb-in. per inch of circumference.
q_0 = —1590 lb. per inch of circumference.

These values substituted in Equations [17] and [18] give the stresses in the wall resulting from the moment and shear produced by the head.

On these results we must superpose the deflection and stresses produced by the internal pressure as well as those resulting from the temperature gradient as given by Equations [11], [13], and [14]. The totals are:

$$y = \frac{1-\mu^2}{EI}\frac{1}{2\beta^3}(180\psi - 3880\theta) + \frac{1+\mu}{2\beta^3}\frac{a\Delta T}{h}\psi + \frac{pc^2}{Eh}$$

$$\sigma_x = \frac{Ea\Delta T}{1-\mu}\frac{\eta}{h}(1-\phi) - \frac{12\eta}{h^3}(180\theta + 3880\psi) + \frac{pc}{2h}$$

$$\sigma_t = Ea\Delta T\frac{\eta}{h} + E\frac{y}{c} + \mu\left(\sigma_x - \frac{pc}{2h}\right)$$

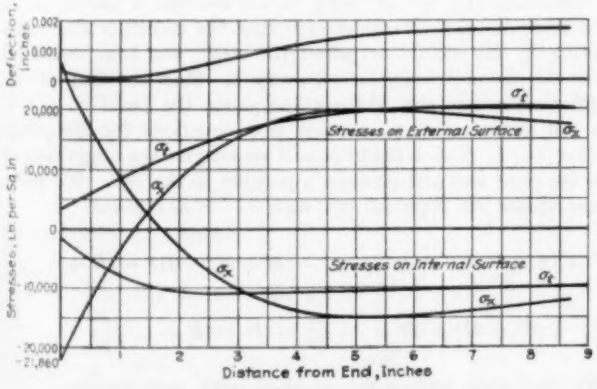


FIG. 11 CYLINDER WALL OF EXAMPLE 3

Fig. 11 shows the deflection of the cylinder wall and the stresses on the exterior and interior surfaces. For the plate, the effect of the internal pressure must be added to that of the moment and shear at the edges. The tensile stress along any radius of the outer surface is given by:

$$\sigma_r = {}^3/_4\frac{p}{h_1^2}(3+\mu)(2cx - x^2) - \frac{6M_0}{h_1^2} - \frac{q_0}{h_1}$$

Fig. 12 shows the deflection and tensile stress along a radius of the outer surface of the cylinder head.

VARIABLE TEMPERATURE GRADIENT IN RADIAL DIRECTION

Assume now that a hollow cylinder of the form we have been considering is heated in such a manner that its temperature gradient in the radial direction varies along its length. As the cylinder walls are assumed thin, this gradient still remains linear. In order to prevent deflection of the mid-plane of the cylinder its temperature must remain at a constant value which

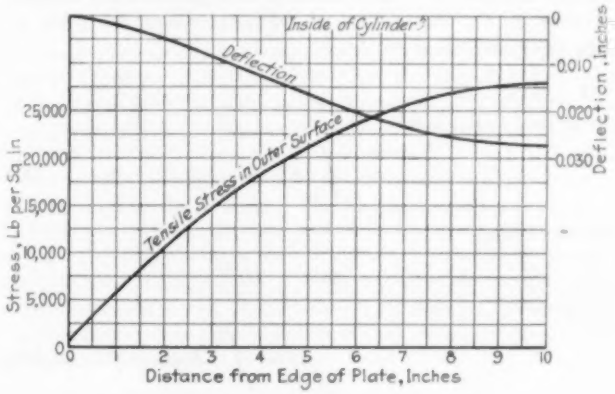


FIG. 12 CYLINDER HEAD, EXAMPLE 3

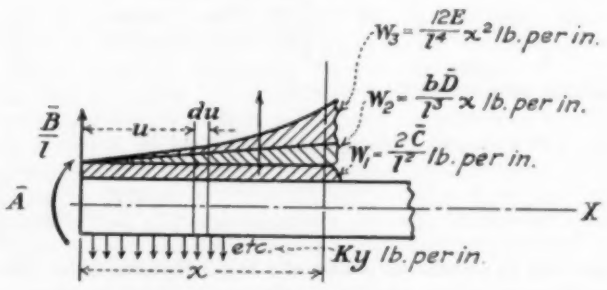


FIG. 13

we may take as zero. Let the temperature of the interior be given by the power series:

$$T_i = \frac{1}{2}\left[A + B\frac{x}{l} + C\frac{x^2}{l^2} + D\frac{x^3}{l^3} + E\frac{x^4}{l^4} + \ldots\right] \quad \ldots [19]$$

where A, B, etc., are constants.

Then the temperature of the exterior at any value of x is equal to $-T_i$. The temperature difference between the two surfaces is $2T_i$. This difference produces the curvature of the element shown in Fig. 1, and the radius of curvature is found, as in Equation [2], to be:

$$\frac{1}{r} = \frac{2aT_i}{h}$$

Obviously this curvature is not now a constant in the X-direction. If the unit strip of Fig. 3 were bent by moments and loads to the form described by this equation, its internal moment at any point x would be:

$$M_x = \frac{EI}{1-\mu}\frac{1}{r} = \frac{EI}{1-\mu}\frac{a}{h}\left[A + B\frac{x}{l} + C\frac{x^2}{l^2} + \ldots\right]$$

allowing no change in the angle θ, Fig. 3. To remove the curvature in the X-direction will require application of moments

and loads equal and opposite to those which produced the internal moment M_x. The moment at the ends is $M_{x(x=0)} = \overline{A}$, where $\overline{A}$ is written for $-\frac{EI}{1-\mu}\frac{a}{h}A$; the load at the end is $\frac{dM_x}{dx_{(x=0)}} = \frac{\overline{B}}{l}$; the distributed load is $\frac{d^3M_x}{dx^3} = 2\frac{\overline{C}}{l^2} + \frac{6\overline{D}}{l^3}x + \ldots$ pounds per unit length. The sum of these separate moments and loads represents the total forces acting on the strip. Fig. 13 shows these forces, remembering that all the loads have a minus sign, and that the number of distributed loads depends on the number of terms in T_i. Summation of moments about the point x, as in Equation [7], gives:

$$\frac{EI}{1-\mu^2}\frac{d^2y}{dx^2} - \frac{EI}{1-\mu}\frac{a}{h}\left(A + B\frac{x}{l} + C\frac{x^2}{l^2} + \ldots\right) + \int_0^x Ky\,(x-u)\,du = 0 \quad \ldots\ldots [20]$$

Differentiating,

$$\frac{EI}{1-\mu^2}\frac{d^3y}{dx^3} - \frac{EI}{1-\mu}\frac{a}{h}\left(\frac{B}{l} + \frac{2Cx}{l^2} + \frac{3Dx^2}{l^3} + \ldots\right) + \int_0^x Ky\,dx = 0 \quad \ldots\ldots [21]$$

Differentiating again and transposing,

$$\frac{EI}{1-\mu^2}\frac{d^4y}{dx^4} + Ky = \frac{EI}{1-\mu}\frac{a}{h}\left[\frac{2C}{l^2} + \frac{6D}{l^3}x + \frac{12E}{l^4}x^2 + \ldots\right]$$

Solving, after setting $C_1 = C_2 = 0$,

$$y = e^{-\beta x}(C_3\cos\beta x + C_4\sin\beta x) + \frac{ahc^2}{12(1-\mu)}\left[\frac{2C}{l^2} + \frac{6D}{l^3}x + \frac{12E}{l^4}x^2 + \ldots\right]$$

provided T_i contains no powers of x higher than the fifth. The modification of this solution to take care of higher powers of x is easily made. Let us assume that T_i is limited to the first four terms of Equation [19] and obtain expressions for the stresses produced. Then:

$$\frac{d^2y}{dx^2_{(x=0)}} = -2\beta^2C_4$$

From [20], when $x = 0$,

$$-\frac{EI}{1-\mu^2}2\beta^2C_4 - \frac{EI}{1-\mu}\frac{a}{h}A = 0$$

$$C_4 = -\frac{1+\mu}{2\beta^2}\frac{a}{h}A$$

$$\frac{d^3y}{dx^3_{(x=0)}} = 2\beta^3(C_3 + C_4)$$

and from [21]:

$$C_3 = \frac{1+\mu}{2\beta^2}\frac{a}{h}\left(A + \frac{B}{\beta l}\right)$$

Then

$$y = \frac{1+\mu}{2\beta^2}\frac{a}{h}\left[\left(A + \frac{B}{\beta l}\right)\theta - A\alpha\right] + \frac{ahc^2}{12(1-\mu)}\left[\frac{2C}{l^2} + \frac{6D}{l^3}\right] \quad \ldots\ldots [22]$$

$$\frac{d^2y}{dx^2} = (1+\mu)\frac{a}{h}\left[\left(A + \frac{B}{\beta l}\right)\alpha + A\theta\right]$$

$$\sigma_x' = -\frac{Ea}{1-\mu}\frac{\eta}{h}\left[\left(A + \frac{B}{\beta l}\right)\alpha + A\theta\right] \quad \ldots\ldots [23]$$

$$\sigma_t' = \frac{Ey}{C} + \mu\sigma_x \quad \ldots\ldots [24]$$

To these stresses produced by the deflection y must be added those due to the temperature difference which are given by substituting $2T_i$ for ΔT in Equation [4]:

$$\frac{Ea}{1-\mu}\frac{\eta}{h}\left(A + B\frac{x}{l} + C\frac{x^2}{l^2} + D\frac{x^3}{l^3}\right) \quad \ldots\ldots [25]$$

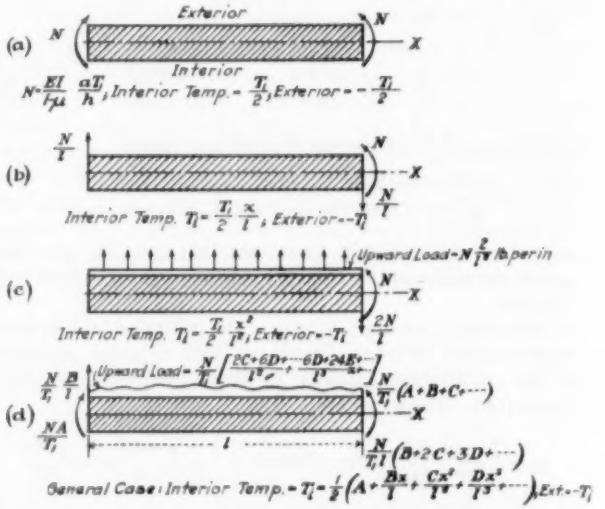


FIG. 14 MOMENTS AND LOADS ON SECTION OF CYLINDER WALL, 1 UNIT WIDE, PRODUCED BY THERMAL STRESSES. TEMPERATURE OF MID-PLANE = 0 DEG.

(In all cases the elastic foundation damps out deflection and stresses produced by moments and loads on ends before middle of cylinder is reached.)

Equations [19] to [24] apply to the left end of the cylinder. We have assumed l large enough that the moments and loads acting on one end do not affect the other, and have indicated that this distance should be at least $2\pi/\beta$. Since the temperature difference is not now symmetrical about the point $x = l/2$, it will be necessary to develop new equations for the range $l/2 < x < l$. This is easily done if we imagine the origin moved to the right end and measure x positive to the left. We must now replace x in Equation [19] with $(l - x)$ as follows:

$$T = [A + B + C + D + \ldots] - [B + 2C + 3D + 4E + \ldots]\frac{x}{l} + [C' + 3D + 6E + \ldots]\frac{x^2}{l^2} - [D + 4E + \ldots]\frac{x^3}{l^3} + \ldots$$

We may then replace the brackets with new constants, A', B', etc., as:

$$T = [A'] - [B']\frac{x}{l} + [C']\frac{x^2}{l^2} - [D']\frac{x^3}{l^3} + \ldots$$

Equations [20] to [24] might then be rewritten by placing a prime on each of the constants, A, B, etc., and thus apply to the right half of the same cylinder, remembering that x is measured positive toward the left.

A temperature distribution such as has been assumed in this

section would be rarely met with. Its usefulness lies in the fact that it produces no distortion of the mid-plane of the cylinder. We may then superpose on this case the stresses caused by a temperature gradient in the axial direction only, obtaining any desired temperature distribution on the two surfaces of the shell, assuming always that it has circular symmetry. Fig. 14 shows the moments and loads applied to secure the condition of free ends for different cases of temperature distribution. These moments and loads produce the deflection y in this section.

Temperature Gradient in Axial Direction Only

Assume again a thin cylindrical shell of elastic material, of mean radius c and of length l. Let the temperature rise be a constant over any cross-section, but vary with the axial coordinate in a manner that may be expressed by the power series:

$$T = \frac{1}{2}\left[A + B\frac{x}{l} + C\frac{x^2}{l^2} + D\frac{x^3}{l^3} + \dots\right] \dots\dots [26]$$

Consider a unit strip cut from this shell as in Fig. 3. Let s_t be the *average* tensile stress in the tangential direction acting on the sides of the strip and y its displacement in the direction of the radius. The unit strain in the circumferential direction is:

$$e_t = \frac{y}{c} = \frac{s_t}{E} + aT$$

Then

$$s_t = \frac{Ey}{c} - EaT$$

The force per unit length of each side of the strip is

$$s_t h = h\left(E\frac{y}{c} - EaT\right)$$

The component of this force in the radial direction per unit length of strip is

$$\frac{s_t h}{c} = \frac{Eh}{c}aT - \frac{Eh}{c^2}y$$

The change of sign indicates that for an outward deflection the force is inward. Referring to Fig. 5, there is now acting in addition to Ky the force $(Eh/c)aT$ per unit length of strip, which is equivalent to a distributed load of that intensity acting on a beam supported by an elastic foundation. The differential equation becomes:

$$\frac{EI}{1-\mu^2}\frac{d^4y}{dx^4} + Ky = \frac{Eh}{c}aT$$

The solution is, as before,

$$y = e^{-\beta x}(C_3\cos\beta x + C_4\sin\beta x) + \frac{ac}{2}\left[\left(A - \frac{6E}{\beta^4 l^4} - \dots\right) + \left(B - \frac{30F}{\beta^4 l^4} - \dots\right)\frac{x}{l} + \left(C - \frac{90G}{\beta^4 l^4} - \dots\right)\frac{x^2}{l^2} + \dots\right.$$

Assume T to be given by the first four terms of the series in Equation [26]. From the condition that the ends are free, we find:

$$\frac{d^2y}{dx^2}_{(x=0)} = 0 = -2\beta^2 C_4 + \frac{ac}{2}\frac{2C}{l^2}. \qquad C_4 = \frac{ac}{2}\frac{C}{\beta^2 l^2}$$

$$\frac{d^3y}{dx^3}_{(x=0)} = 0 = 2\beta^3(C_3 + C_4) + \frac{ac}{2}\frac{6D}{l^3}. \quad C_3 = -\frac{ac}{2\beta^3 l^3}(\beta lC - 3D)$$

$$y = \frac{ac}{2}\left[A + B\frac{x}{l} + C\frac{x^2}{l^2} + D\frac{x^3}{l^3} + \frac{3D - \beta lC}{\beta^2 l^2}\theta + \frac{C}{\beta^2 l^2}\alpha\right]. \quad [27]$$

$$\frac{d^2y}{dx^2} = ac\left[\frac{C}{l^2}(1-\phi) + \frac{3D}{\beta l^3}(\beta x + \alpha)\right]$$

$$\sigma_x = \frac{Eac}{1-\mu^2}\eta\left[\frac{C}{l^2}(\phi - 1) - \frac{3D}{\beta l^3}(\beta x + \alpha)\right] \dots\dots [28]$$

$$\sigma_t = \frac{Ea}{2}\left[\frac{3D - \beta lC}{\beta^2 l^2}\theta + \frac{C}{\beta^2 l^2}\alpha\right] + \mu\sigma_x \dots\dots [29]$$

No stresses result from temperature gradients in this direction unless Equation [26] includes at least the second power of x. If no terms above the third power of x are present, the deflection curve away from the ends is the normal expansion curve of the cylinder wall for the temperature rise assumed.

As in the preceding section, these equations apply only to the left end of the cylinder. For the right end we must again use the constants A', B', etc., and measure x positive to the left.

By combining this case with the preceding one we may arrive at the desired distribution. There may also be superposed on these the effect of end connections or of pressure. Stresses due to gradients in the axial direction will be small, except in cases of localized heating as in welding.

Interior Temperature a Linear Function of x

Let us now consider cases in which a variable temperature gradient exists in the radial direction accompanied by a gradient along the mid-plane of the cylinder in the X-direction. These will be solved by superposing the solutions already obtained in these two cases. Assume the external temperature of the cylindrical shell to be kept at zero and the interior to be held at a value: $T_i = T_1 x/l$. In Equation [19], $B = T_1$; all other constants are zero. In Equation [26], the temperature along the midplane is: $T = (T_1/2)x/l$, so that $B = T_1$ in this equation also, and other constants are zero. Substitute these values of the constants in Equations [22] and [27], which may then be added to give:

$$y = \frac{1+\mu}{2\beta^2}\frac{aT_1}{hl}\theta + \frac{acT_1}{2}\frac{x}{l}$$

In the same way, by adding Equations [23], [25], and [28],

$$\sigma_x = \frac{EaT_1}{1-\mu}\frac{\eta}{h\beta l}(\beta x - \alpha)$$

And by adding [24], [25], and [29],

$$\sigma_t = \frac{EaT_1}{1-\mu}\frac{\eta}{h\beta l}\left(\beta x - \mu\alpha + \frac{1-\mu^2}{2\beta^2 c\eta}\theta\right)$$

These equations apply to the left end of the cylinder. For the right end we must substitute A', B', etc., to obtain:

$$y = \frac{1+\mu}{2\beta^2}\frac{aT_1}{h}\left(\psi - \frac{\theta}{\beta l}\right) + \frac{aT_1}{2}\frac{c}{l}(l - x)$$

$$\sigma_x = \frac{EaT_1}{1-\mu}\frac{\eta}{h}\left(1 - \phi - \frac{\beta x - \alpha}{\beta l}\right)$$

$$\sigma_t = \frac{EaT_1}{1-\mu}\frac{\eta}{h}\left[1 - \frac{x}{l} - \mu\left(\phi - \frac{\alpha}{\beta l}\right) + \frac{1-\mu^2}{2\beta^2 c\eta}\left(\psi - \frac{\theta}{\beta l}\right)\right]$$

When $\eta = h/2$ and $x = l/2$,

$$\sigma_x = \sigma_t = \frac{EaT_1}{4(1-\mu)}$$

This value is necessarily the same as that obtained from the equations derived from the other end.

$$\text{When } x = 0,\ \sigma_x = 0, \text{ and } \sigma_t = \frac{EaT_1}{2}\left[1 + \frac{1+\mu}{\beta^3 hc}\left(1 - \frac{1}{\beta l}\right)\right]$$

We should expect the values of the deflection and tangential stress at the end where $T_i = 0$ to be quite low, as may be shown by an example.

Example 4. In order to compare the result with Example 1 we shall again assume a cylinder of the same material and dimensions, but the internal temperature is to be given by:

$$T_i = T_1 \frac{x}{l}$$

where $T_1 = 100$ deg. fahr., and the external temperature is zero.

The stresses at the middle are, of course, one-half those in

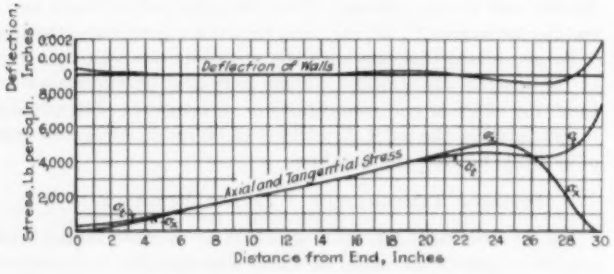


FIG. 15 CYLINDER OF EXAMPLE 4

the first example. The other results will differ at each end. At the end $T_i = 0$,

$$y = \frac{1+\mu}{2\beta^2}\frac{aT_1}{hl} = 0.182 \times 10^{-3} \text{ in.}$$

When $\eta = h/2$,

$$\sigma_t = \frac{EaT_1}{2}\frac{1+\mu}{\beta^3 hcl} = 272 \text{ lb. per sq. in.}$$

At the end $T_i = T_1$,

$$y = \frac{1+\mu}{2\beta^2}\frac{aT_1}{h}\left(1 - \frac{1}{\beta l}\right) = 2.05 \times 10^{-3} \text{ in.}$$

When $\eta = h/2$,

$$\sigma_t = \frac{EaT_1}{2}\left[1 + \frac{1+\mu}{3^3 hc}\left(1 - \frac{1}{\beta l}\right)\right] = 7575 \text{ lb. per sq. in.}$$

Fig. 15 shows graphically the stresses and deflection in this case.

INTERIOR TEMPERATURE A FUNCTION OF x^2

If in Equation [19] we place $C = T_1$ and all other constants equal to zero, we come to the case of parabolic distribution of the internal temperature in the direction of the axis. In Equation [26], if the temperature along the mid-plane is $T = T_1 x^2/l^2$, which makes $C = T_1$, and all other constants zero, we bring the exterior temperature to zero and the interior becomes $2T_i$.

After substituting these values for the constants we add Equations [22] and [27]:

$$y = \frac{aT_1}{1-\mu}\frac{hc^2}{6l^2} + \frac{aT_1 c}{2\beta^2 l^2}(\beta^2 x^2 - \psi)$$

Adding [23], [25], and [28],

$$\sigma_s = \frac{EaT_1}{1-\mu}\frac{\eta}{h}\frac{1}{l^2}\left(x^2 + \frac{hc}{1-\mu}(1-\phi)\right)$$

And from [24], [25], and [29],

$$\sigma_t = \frac{EaT_1}{1-\mu}\frac{\eta}{h}\frac{1}{l^2}\left[x^2 + \frac{h^2 c}{6\eta} - \frac{h\psi}{2\beta^2\eta}(1-\mu) + \mu hc(\phi - 1)\right]$$

By substitution of the values of the constants applying to the right end of the cylinder we obtain the equations for stress and deflection in this part. In general, the curves of deflection and stresses will be similar to those in Fig. 15, except that the stress curve connecting the two end portions now becomes a parabola instead of a straight line.

CASES INVOLVING VARIATIONS OF PREVIOUS METHODS

Cases of sudden changes in wall thickness may be investigated in the same way as this problem is handled in the ordinary beam. The strip is cut at the point of change of section and a moment and shear assumed at each of the cut ends. The magnitude of these is then determined by the conditions of continuity at the junction. For a slightly tapering wall thickness an average value may be assumed, since the wall thickness plays only a small part in determining the stress.

A sudden change in the form of the function describing the interior temperature T_i might be handled in the same way.[7] Usually such changes in the interior temperature will not occur suddenly, but merge into each other gradually, which requires no special treatment of the junction unless one portion be shorter than the half-wave-length, π/β.

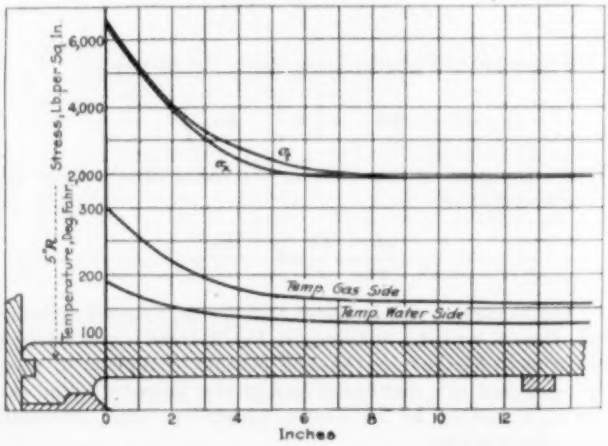


FIG. 16 TEMPERATURE AND STRESSES IN CYLINDER LINER—EXAMPLE 5

An example of this kind occurs in the cylinder liner of a Diesel engine. Near the combustion end the temperature of both the gas and water side are represented by equations of the second degree for distances of one-third to one-half the stroke. From this point on the curves become linear with a slight downward slope. Dr. Eichelberg (11) calculated these curves from a consideration of heat flow within the cylinder and later verified them by means of thermocouples embedded in the walls. Dr. Letson (12) also made experimental determinations of the temperature distribution in a liner, which agree with those of Dr. Eichelberg. Of course the maximum temperatures reached are far from being the same since these depend on the type of engine investigated, but the shape of the curves on both the gas and water sides were very similar.

Example 5. Fig. 16 shows a portion of a cylinder liner with temperature curves for the two surfaces as determined in the

[7] Timoshenko and Lessells, "Applied Elasticity," p. 146.

experiments referred to. At the upper end the liner is pressed into its casing which is assumed to exert a moment and shear sufficient to bring the slope and deflection of the liner to zero at this point. At other points the liner is simply supported, no moment being exerted. The material is cast iron and the dimensions are: mean diameter, 10 in.; thickness, 1 in.; complete length, possibly 25 to 30 in. The temperature of the liner on the gas side is given by

$$T_g = 300 - 50x + 5x^2$$

and on the water side by

$$T_w = 190 - 20x + 1.8x^2$$

The moment equation becomes, for the region between $0 < x < 7$:

$$\frac{EI}{1-\mu^2}\frac{d^2y}{dx^2} + \int_0^x Ky(x-\mu)d\mu + M_0 + q_0 x = \frac{EI}{1-\mu}\frac{a}{h}(T_g - T_w)$$

The solution of this equation yields the following results:

M_0 = 1004 lb. in. per in. of circumference.
q_0 = —294 lb. in. per in. of circumference.
y = $0.1066(1-\phi)10^{-3}$ in.

For $\eta = h/2$,

$$\sigma_x = 60(110 - 30x + 3.2x^2) - 191.2\psi$$
$$\sigma_t = 60(110 - 30x + 3.2x^2) + 319.8(1-\phi) - 47.8\psi$$

The temperature of the mid-plane in this case will be, $T = 55 - 15x + 1.6x^2$. Stresses resulting from the distortion of this plane will be

$$\sigma_x = 2\frac{Eac}{1-\mu^2}\eta\left[\frac{1.6}{l^2}(\phi - 1)\right]$$

$$\sigma_t = Ea\frac{1.6}{\beta^2 l^2}\alpha + \mu\sigma_x$$

The maximum value of σ_x in the above equations is —75 lb. per sq. in. in this example, and for σ_t is 36 lb. per sq. in. The effect of the distortion of the mid-plane on stresses in this case is therefore small.

Beyond the point $x = 7$ the temperature curve becomes a straight line, and its downward slope is so small that it may be neglected. The stress in this portion is then given by $Ea(T_g - T_w)/2(1-\mu)$, which corresponds with the value of σ_x taken from the equation above for the point $x = 6$. In the stress curves of Fig. 16 it is noted that σ_t is slightly higher for this value of x, which is owing to the effect of the constant deflection produced by the parabolic temperature distribution.

Because of the effect of the moment and shear at the end in limiting the deflection, the maximum stress in this case is only 150 lb. per sq. in. above that calculated by the usual formulas applying to long cylinders with temperature varying only in the radial direction. Example 2 indicated similar results. The general effect of such moments and shears is, of course, to approximate the conditions of a long cylinder, but whatever the value of external forces or moments applied, the method developed in this paper enables the resulting stresses to be easily calculated.

Appendix No. 1

FROM the curves of Figs. 17 and 18 may be taken the values of the expressions represented by ϕ, ψ, θ, and α, which are required for the calculation of stresses and deflection of the cylinder wall. The derivatives of these functions are required in many calculations and are given below.

$$\frac{d\theta}{dx} = -\beta\phi; \quad \frac{d\alpha}{dx} = \beta\psi; \quad \frac{d\phi}{dx} = -2\beta\alpha; \quad \frac{d\psi}{dx} = -2\beta\theta$$

$$\frac{d^2\theta}{dx^2} = 2\beta^2\alpha; \quad \frac{d^2\alpha}{dx^2} = -2\beta^2\theta; \quad \frac{d^2\phi}{dx^2} = -2\beta^2\psi; \quad \frac{d^2\psi}{dx^2} = 2\beta^2\phi$$

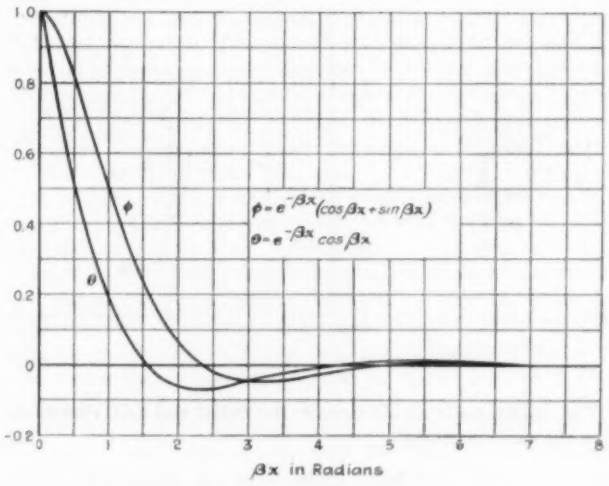


FIG. 17 VALUES OF THE FUNCTIONS ϕ AND θ

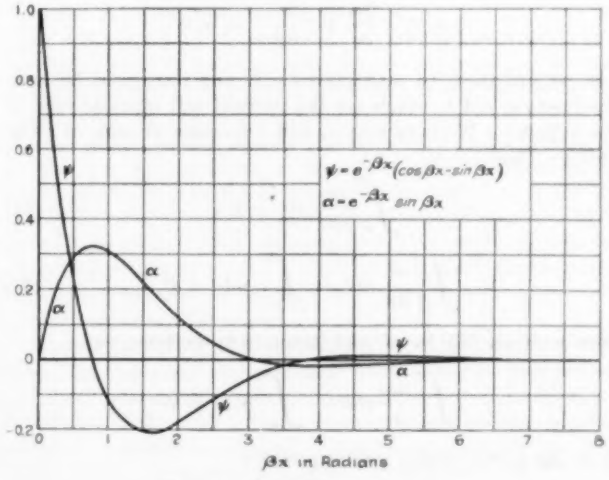


FIG. 18 VALUES OF ψ AND α

Appendix No. 2

ALTERNATE DERIVATION OF FORMULA [9]

Fig. 2(*a*) shows a portion of a thin cylinder which is assumed to be infinitely long. Its internal surface is held at the constant temperature T_1 while the outside is kept at zero. The resulting axial and tangential stresses are the same as those in a thin plate, similarly heated and not allowed to bend, that is, $\frac{EaT_1}{1-\mu}\frac{\eta}{h}$. We note that at the ends of the portion assumed, the axial stresses are such as to produce a moment whose value is $\frac{EI}{1-\mu}\frac{aT_1}{h}$. If we wish to investigate a cylinder of finite length we can produce a condition of free ends by imagining a moment of opposite direction applied to them as in Fig. 2(*b*). The stresses pro-

duced by this moment may then be added to those already existing.

To investigate the stresses produced by a moment acting on the ends of such a cylinder, we assume an infinitesimal parallelopiped cut from its wall as in Fig. 19. In this figure the expression $\sigma_r + \ldots$ is an abbreviation for $\sigma_r + \frac{\partial \sigma_r}{\partial r} dr$, and similarly for the stresses σ_x and τ.

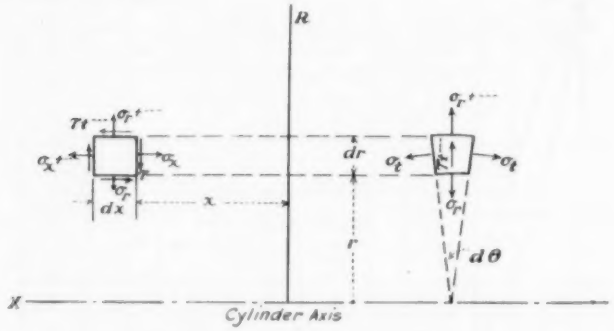


FIG. 19

The conditions of equilibrium in the radial and axial directions give the two equations:

$$r \frac{\partial \sigma_x}{\partial x} + r \frac{\partial \tau}{\partial r} + \tau = 0 \ldots\ldots\ldots\ldots [30]$$

$$\sigma_r + r \frac{\partial \sigma_r}{\partial r} + r \frac{\partial \tau}{\partial x} - \sigma_t = 0$$

Let each of these be multiplied by dr and integrated between the limits a and b, which are the internal and external radii of the cylinder. We note that σ_r and τ are zero at each of these limits. Then:

$$\int_a^b \frac{\partial \sigma_x}{\partial x} r dr = 0 \ldots\ldots\ldots\ldots [31]$$

$$\int_a^b \frac{\partial \tau}{\partial x} r dr - \int_a^b \sigma_t dr = 0 \ldots\ldots\ldots [32]$$

Now multiply [30] by rdr and integrate in the same way:

$$\int_a^b \frac{\partial \sigma_x}{\partial x} r^2 dr - \int_a^b \tau r dr = 0$$

Take the partial of this with respect to x:

$$\int_a^b \frac{\partial^2 \sigma_x}{\partial x^2} r^2 dr - \int_a^b \frac{\partial \tau}{\partial x} r dr = 0$$

and from [32] we have

$$\int_a^b \frac{\partial^2 \sigma_x}{\partial x^2} r^2 dr = \int_a^b \sigma_t dr \ldots\ldots\ldots [33]$$

Now consider a strip cut from this cylinder as in Fig. 3, whose width is $cd\theta$. The condition that the axial stress over any cross-section of the strip shall form a couple, is:

$$\int_a^b \sigma_x r d\theta dr = 0$$

The agreement of this equation with [31] indicates that this is a possible stress condition. The moment of this couple about any point is:

$$M = - \int_a^b \sigma_x r^2 d\theta dr$$

by choosing a point on the cylinder axis. Then from Equation [33]:

$$\frac{d^2M}{dx^2} = - d\theta \int_a^b \sigma_t dr \ldots\ldots\ldots\ldots [34]$$

Now by geometry $e_t = y/c$, and from the equations expressing Hooke's law for the two-dimensional stress condition,

$$e_t = \frac{\sigma_t}{E'} - \mu \frac{\sigma_x}{E}$$

where e_t is the unit tangential deformation. Then[8]

$$\sigma_t = E \frac{y}{c} + \mu \sigma_x$$

and since

$$\int_a^b \sigma_x dr = 0, \qquad d\theta \int_a^b \sigma_t dr = E \frac{y}{c} d\theta \int_a^b dr = E \frac{y}{c} h d\theta$$

Equation [34] becomes:

$$\frac{d^2M}{dx^2} = - E \frac{y}{c} h d\theta \ldots\ldots\ldots\ldots [35]$$

For the bending of a thin strip, we have[8]

$$M = \frac{EI}{1 - \mu^2} \frac{d^2y}{dx^2}$$

where $I = cd\theta h^3/12$. Then from [35]:

$$\frac{d^4y}{dx^4} + \frac{12(1 - \mu^2)}{h^2c^2} y = 0 \ldots\ldots\ldots\ldots [9]$$

and we have as before:

$$y = e^{-\beta x}(C_3 \cos \beta x + C_4 \sin \beta x)$$

At the ends the shear is zero and the moment is $M_0 = \frac{EI}{1 - \mu} \frac{aT_1}{h}$

By evaluating $\frac{d^2y}{dx^2_{(x=0)}}$ and $\frac{d^3y}{dx^3_{(x=0)}}$, we find $C_3 = - C_4 = \frac{aT_1}{h} \frac{1 + \mu}{2\beta^2}$.

The axial and tangential stresses produced by the application of the moment alone are then found to be

$$\sigma_x' = - \frac{EaT_1}{1 - \mu} \frac{\eta}{h}$$

$$\sigma_t' = \frac{EaT_1}{1 - \mu} \frac{\eta}{h} \left(\frac{1 - \mu^2}{2\beta^2 c \eta} - \mu \right)$$

The total stress in the cylinder walls will be found by adding to these the stress already existing owing to the temperature gradient. This, of course, reduces σ_x to zero at the ends and gives for σ_t, as before,[8]

$$\sigma_t = \frac{EaT_1}{h} \left(\frac{1 + \mu}{2\beta^2 c} + \eta \right)$$

BIBLIOGRAPHY OF THERMAL STRESSES

PERIODICALS

1. T. M. C. Duhamel, "Memorie sur le Calcul des Actions Moleculaires etc.," Académie des Sciences de L'institute de France, vol. 5 (1838), p. 440.

[8] Contrast with Föppl, "Vorlesungen über Technische Mechanik," vol. 5, p. 248.

2. Franz Neumann, "Die Gesetze der Doppelbrechung des Lichts," Abhandlungen Akademie der Wissenschaften zu Berlin, 1841, pp. 3–247.
3. C. W. Borchardt, "Untersuchungen über die Elasticität fester isotroper Körper unter Berucksichtigung die Wärme," *Monatsberichte*, Akademie der Wissenschaften zu Berlin, 1873, pp. 9–56.
4. J. Hopkinson, "On the Stresses Caused in an Elastic Solid by Inequalities of Temperature," *The Messenger of Mathematics*, vol. 8 (1879), p. 168.
5. Alfons Leon, "Spannungen und Formänderungen eines Holzylinders und einer Hohlkugel, etc.," *Zeitschrift für Mathematik und Physik*, vol. 52 (1905), pp. 174–190.
6. Rudolph Lorenz, "Temperaturspannung in Holzylinder," *Zeit. V.D.I.*, vol. 51 (1907), pp. 743–747.
7. Adams and Williamson, "Temperature Distribution in Solids During Heating or Cooling," *Physical Review*, vol. 14 (1919), pp. 99–114.
8. Williamson, "Strains Due to Temperature Gradients, etc.," *Journal Washington Academy of Sciences*, vol. 9 (1919), pp. 209–217.
9. Adams and Williamson, "The Annealing of Glass," *Journal Franklin Institute*, vol. 190 (1920), pp. 597–632 and 835–870.
10. C. H. Lees, "The Thermal Stresses in Solid and Hollow Circular Cylinders Concentrically Heated," Proc. Royal Soc. of London, vol. 101 (1922), p. 411.
11. G. Eichelberg, "Temperaturverlauf und Wärmespannungen in Verbrennungsmotoren," *Forschungsarbeiten V.D.I.*, Heft 263 (1923).
12. H. F. G. Letson, "Temperature Distribution in Diesel-Engine Liners," Proc. Inst. M. E., London (1925), pp. 19–52.
13. G. Grünberg, Über—durch ungleichförmige Erwärmung erregten Spannungzustand," *Zeitschrift für Physik*, vol. 35 (1925), pp. 548–555.
14. W. Nusselt, "Der Wärmeübergang in den Diesel-maschine," *Forschungsarbeiten V.D.I.*, Heft 14, Band 70 (1926), p. 468.
15. R. Sulzer, "Temperature Variations and Heat Stresses in Diesel Engines," *Engineering*, vol. 121 (1926), p. 447.
16. L. H. Barker, "Stresses in Tubes," *Engineering*, vol. 124 (1927), p. 443.
17. K. Requa, "Beitrag zur Beurteilung von Temperaturefeld, etc.," *Forschungsarbeiten V.D.I.*, Heft 301 (1928).
18. A. Nägel, "Transfer of Heat in Reciprocating Engines," *Engineering*, vol. 127 (1929), pp. 59, 179, 279, 466, 626.

BOOKS

20. Franz Neumann, "Vorlesungen über die Theorie der Elasticität," (1885), pp. 107–120.
21. Föppl, A., "Vorlesungen über Technische Mechanik," vol. V, 4th Ed., p. 238.
22. Case, "Strength of Materials," (1925), p. 456.
23. Timoshenko and Lessells, "Applied Elasticity," (1925), pp. 102, 146, 272.
24. Love, "Theory of Elasticity," 4th Ed., (1927), p. 108.
25. Stodola, "Steam and Gas Turbines," (1927), pp. 388, 1155.
26. Timoshenko, "Strength of Materials," (1930), pp. 487, 550.
27. Carslaw, "The Conduction of Heat," (1921).
28. Ingersoll and Zobel, "The Mathematical Theory of Heat Conduction," (1913).
29. Byerly, "Fourier Series and Spherical Harmonics," (1893).

Discussion

J. E. Boyd.[9] Determination of thermal stresses in hollow cylinders is of great importance for the design of modern high-pressure boilers and for internal-combustion engines. Some phases of the problem have been presented before the Society. At the Cleveland meeting, in May, 1924, O. G. C. Dahl gave a paper entitled, "Temperature and Stress Distribution in Hollow Cylinders," Trans., 1924, Vol. 46, pp. 161–208. This paper gives complete derivation of the equations for radial and tangential deformations for constant temperature differences and for temperature which undergoes a sinusoidal variation with time. The approximate theory is given on the assumption of a straight-line temperature gradient, together with the exact theory based on the logarithmic gradient.

[9] Professor of Mechanics, Ohio State University, Columbus, Ohio. Life Mem. A.S.M.E.

Dahl's Equations [71] and [72], on page 179, give the maximum tangential stresses at the inner and outer surfaces, calculated from the logarithmic temperature gradient. These stresses were computed on the assumption of zero axial stress. With axial deformation equal to the tangential deformation, as is found to be the case at some distance from the ends, these equations must be divided by $(1 - \mu)$ to get the true combined stress in either direction. (μ = Poisson's ratio.)

Dahl's paper and the appendix together develop the subject from the beginning. The reader with limited library facilities will find all that he needs for the study of the radial and tangential deformation caused by constant temperature difference between the inner and outer walls. The problem of sinusoidal temperature difference demands considerable training in advanced mathematics.

In addition to the tangential and radial deformations in a tube caused by a difference in temperature between the inner and outer surfaces, there is an axial deformation. If the ends are free to move, the entire tube elongates to a length which corresponds to the temperature approximately midway between the inner and outer surfaces. This temperature of a cylinder of filaments under zero axial stress may be represented by $\bar{t}$. Filaments of temperature $\bar{t}$ have their expansion reduced by shear transmitted from filaments of lower temperature. Unit deformation of a length Δx is calculated by subtracting the actual elongation from the elongation which corresponds with the temperature difference $t - \bar{t}$, and dividing the remainder by the original length Δx. The unit deformation is the change in unit length caused by forces from portions external to the element. Near the ends of an open tube the axial unit deformations are small, as the only external force is the shear over a small area. A plane section near the end, which is normal to the axis under constant temperature, becomes a conical surface when a temperature difference is established between the inner and outer surfaces of the tube. On the other hand, a length Δx at a considerable distance from the free end is subjected to a compressive or tensile force which is equal to the entire shear acting on the element from the end of Δx to the free end of the tube. At a relatively small distance from the end this direct force may produce a deformation in Δx nearly equal to the expansion caused by $t - \bar{t}$. A plane section then remains practically plane, and the unit deformation is given by $a(t - \bar{t})$, in which a is the coefficient of expansion.

For any part of the tube at sufficient distance from the ends to keep plane sections approximately plane when a temperature difference is established between the inner and outer surfaces, the calculation of the axial unit deformation at any position is easily made by elementary calculus. Using the logarithmic temperature gradient, the exact solution gives for the outer surface

$$\delta = at_1\left(\frac{1}{2\log k} - \frac{1}{k^2 - 1}\right) \quad [36]$$

and for the inner surface

$$\delta = at_1\left(\frac{1}{2\log k} - \frac{k^2}{k^2 - 1}\right) \quad [37]$$

in which t_1 is the difference of temperature between the inner and outer surfaces, a is the temperature coefficient, and k is the ratio of the outer radius to the inner radius. Since these expressions for axial unit deformation are exactly the same as the expressions for tangential unit deformation, the resultant unit stress in either direction is found by multiplying Equations [36] and [37] by $\frac{E}{1 - \mu}$ in which μ is Poisson's ratio.

At the outer surface

$$S_t = S_x = \frac{aEt_1}{1 - \mu}\left(\frac{1}{2 \log k} - \frac{1}{k^2 - 1}\right) \ldots\ldots [38]$$

At the inner surface

$$S_t = S_x = \frac{aEt_1}{1 - \mu}\left(\frac{1}{2 \log k} - \frac{k^2}{k^2 - 1}\right) \ldots\ldots [39]$$

in which S_t is the tangential unit stress and S_x is the axial unit stress.

The derivation of these formulas may be found in some advanced textbooks. For instance, the foregoing equations are written in slightly different form as Equations [159] and [160] on page 554 of Timoshenko's "Strength of Materials." These equations are given for the tangential stress, but it is shown below when b is substituted for r in Equation [161] for axial stress, the result is Equation [160], and when a is substituted for r, the result is Equation [159]. These equations then apply for both tangential and normal stress at the surfaces away from the ends.

Incidentally, the writer is not able to check the figures at the top of the second column on the fifth page. He gets 5810 lb. per sq. in. at the outer surface and —6190 lb. per sq. in. at the inner surface, instead of 5990 and —6100 lb. per sq. in. given by the author. In any case the deviation of the actual stress from the approximate value of 6000 lb. per sq. in. obtained from the straight line temperature gradient is little greater than 3 per cent.

Since the equations for the stresses under uniform conditions away from the ends have been well established, the author has attacked the more difficult problem of the stresses at or near the ends. To solve this problem he imagines a longitudinal strip to be separated from the tube by two planes which intersect on the axis and make an angle with each other. When this strip is heated at the inner surface, it bends upward at each end as it would be bent by a positive moment. The solution of the problem consists in finding the negative moment which will bend the strip considered as a beam back to its original straight form.

The author uses the straight-line temperature gradient which has been shown to cause relatively small error for ratio of external radius to internal radius commonly found in practice. The problem is sufficiently difficult with the straight-line gradient; it probably would be impossible if the logarithmic gradient were used. The author has shown ingenuity and skill in setting up the physical equations and in their mathematical solution. The writer was struck by the method for disposing of the constants C_1 and C_2.

As was to be expected from the fact that axial deformation is free at the end, the author finds the axial tension in the outer fibers to begin at zero at the end and gradually increase (Fig. 6). On the other hand, the maximum tangential stress is at the end and is 30 per cent greater than the stress near the middle. The curves of Fig. 6 alone represent a large and important addition to knowledge.

The author has also solved several problems of cylinder heads and other end connections and has investigated problems in which the temperature varies as the first and second powers of x.

Since the writer has lacked time to study these solutions, he must leave their discussion to others.

This paper forms a valuable contribution to the theory of an important practical problem.

Author's Closure

Professor Boyd has shown how it is possible to take the stress equations given by Mr. Dahl applying to a thin disk and adapt them to that portion of a long cylinder not near the ends. In either form the solutions are well known, but in making use of them one must know the limits of their applicability.

The numerical difference which Professor Boyd notes comes from the fact that the author used Equations [154] and [155] on page 553 of Timoshenko's "Strength of Materials," developed particularly for thin-walled cylinders. The results which Professor Boyd obtains take into account both the logarithmic temperature distribution and the radial stress. The error of the method used in the paper is seen to be larger than that given by Equations [154] and [155], but still small enough for practical purposes.

APM-53-14

Jets From Manifold Tubes

By JAKOB KUNZ,[1] URBANA, ILL.

After a brief reference to previous work on the subject, the author develops an expression for the discharge coefficient. He then determines the flow of a sheet of water through a slot of infinitesimal width in the wall of a pipe by means of a variable and a constant discharge coefficient. The slot is then conceived of as a row of separate holes and the flow problem is solved by means of difference equations for a variable and a constant discharge coefficient. A special case is also considered in which the pipe has a hole through its further (closed) end.

AN ANALYSIS of this problem was given by Prof. M. L. Enger and M. I. Levy in the Journal of the American Water Works Association, Vol. 21, No. 5, May, 1929. In what follows the author proposes to carry Professor Enger's investigation a little further, giving a higher approximation, considering special cases, and using difference equations as well as differential equations.

In a tube (Fig. 1) of cross-section A a stream of water enters with an initial velocity v_0 and escapes through a narrow slot of width b and length l. The tube is closed at the further end. It is found that the height of the jet h_0 at the beginning of the slot is much lower than at the end where it is h_n. This agrees qualitatively with Bernoulli's principle, where the velocity in the tube is high and the pressure low and vice versa; but we cannot simply apply Bernoulli's principle here because the volume of water which passes through any cross-section of the tube per unit time depends on the distance from the beginning of the slot to the cross-section. At a given point distant x from the beginning of the slot, let us assume the velocity in the tube to be v, the corresponding height of the jet, h, and the velocity of the issuing jet, w. Then conservation of energy gives Equation [1]:

$$-d(\tfrac{1}{2}mv^2) = d(mgh) \qquad [1]$$

or

$$dh = -\frac{1}{g}vdv$$

and

$$\int_h^{h_n} dh = h_n - h = -\frac{1}{g}\int_v^0 vdv = \frac{1}{g}\frac{v^2}{2}$$

whence

$$v = \sqrt{2g(h_n - h)} \qquad [2]$$

The volume of water which flows per unit time through the slot bdx is equal to

$$-Adv = wbdxc \qquad [3]$$

where c is the coefficient of discharge. By dimensional reasoning we see that this coefficient is a pure number, which depends at the same time on the height of the jet. The volume of water ΔV which flows with velocity w through an area a is equal to

$$\Delta V = wac$$

The dimensional equation is

$$\frac{L^3}{T} = \frac{L}{T}L^2c$$

hence c is a pure number. Consider now a volume dV which passes through a slot of length dx per unit time, so that

$$dV = wdxc$$

Then the dimensional equation gives for c a length, say, h, whence

$$c = kh, \quad c_n = kh_n, \quad \text{and} \quad \frac{c}{c_n} = \frac{h}{h_n}$$

or

$$c = c_n\frac{h}{h_n} \qquad [4]$$

or, from Equation [2],

$$c = c_n\frac{h_n - \frac{1}{2g}v^2}{h_n}$$

This formula has been verified by Enger; the deduction here given involves at the same time the principle of similitude, as two slits of length dx and dx_1 and of the same width are compared.

A more rigorous deduction would be expected from the theory of streamlines. In this case the height of the jet in the position x is given by

$$\tfrac{1}{2}mw^2 = mgh$$

or by

$$w = \sqrt{2gh} \qquad [5]$$

From [2], [3], [4], and [5] we obtain

$$dv = -c_n\frac{h}{h_n}b\frac{dx}{A}\sqrt{2gh} = -g\frac{dh}{\sqrt{2g(h_n - h)}}$$

$$\frac{2bc_n}{h_nA}\int_0^x dx = \int_{h_0}^{h}\frac{dh}{\sqrt{(h_n - h)h}} = \frac{-2\sqrt{h_nh - h^2}}{h_nh}\Bigg|_{h_0}^{h}$$

or

$$x = \frac{A}{bc_n}\left[\frac{\sqrt{h_nh_0 - h_0^2}}{h_0} - \frac{\sqrt{h_nh - h^2}}{h}\right] \qquad [6]$$

This formula gives x as a function of h; but as we wish the inverse function, we write

$$x = C - C_1\frac{\sqrt{h_nh - h^2}}{h} \qquad [6a]$$

where

$$C_1 = \frac{A}{bc_n} \quad \text{and} \quad C = \frac{A}{bc_n}\frac{\sqrt{h_nh_0 - h_0^2}}{h_0} \qquad [7]$$

From [6a],

$$\frac{(C - x)^2 + C_1^2}{C_1^2} = \frac{h_n}{h}$$

or

$$h = \frac{h_nC_1^2}{(C - x)^2 + C_1^2} \qquad [8]$$

[1] Professor of Physics, University of Illinois.

Presented at the National Applied Mechanics Meeting, Purdue University, Lafayette, Ind., June 15 and 16, 1931, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

which gives the curve of Fig. 1. Introducing the boundary conditions, for $x = 0$, $h = h_0$, hence

$$h_0 = \frac{h_n C_1^2}{C^2 + C_1^2} \quad [9]$$

and for $x = l$, $h = h_n$, hence

$$h_n = \frac{h_n C_1^2}{(C - l)^2 + C_1^2} \quad [10]$$

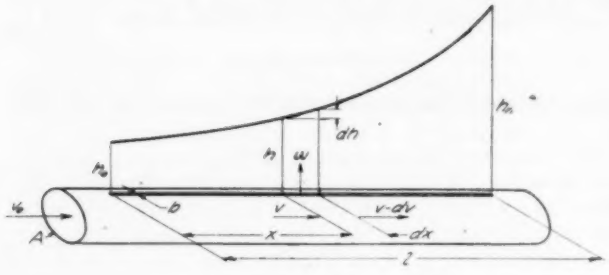


FIG. 1

From this it follows that

$$C = l = C_1 \frac{\sqrt{h_n h_0 - h_0^2}}{h_0} \quad [11]$$

or

$$C^2 = C_1^2 \left(\frac{h_n}{h_0}\right) - C_1^2$$

or

$$\frac{C^2 + C_1^2}{C_1^2} = \frac{h_n}{h_0}$$

If the tube and slot dimensions and the initial velocity of the flow of water are given, then h_n and h_0 are determined as follows. From [2] we obtain

$$V_0^2 = 2g(h_n - h_0)$$

or

$$h_n = \frac{V_0^2 + 2gh_0}{2g} \quad [12]$$

and from [11]

$$l = \frac{A}{bc_n} \frac{\sqrt{h_n h_0 - h_0^2}}{h_0}$$

or

$$\left(\frac{lbc_n}{A}\right)^2 = \frac{h_n}{h_0} - 1$$

or by [12],

$$\left(\frac{lbc_n}{A}\right)^2 + 1 = \frac{V_0^2 + 2gh_0}{2gh_0} = \frac{V_0^2}{2gh_0} + 1$$

whence

$$h_0 = \frac{V_0^2 A^2}{2g(lbc_n)^2} \quad [13]$$

From [12],

$$h_n = \frac{V_0^2}{2g}\left[1 + \frac{A^2}{(lbc_n)^2}\right] \quad [13a]$$

h_n and h_0 are now determined. Let us next assume that the pipe has at the end $x = l$ a hole of cross-section A_1 through which the water flows with the velocity v_0 so that $A_1 v_0 = A v_1$; then we obtain from

$$dh = -\frac{1}{g} v dv$$

$$\int_h^{h'_n} dh = -\frac{1}{g}\int_{v_1}^{v_0} v dv = \frac{1}{2g}(v_0^2 - v_1^2) = h_n' - h$$

h_n' is the height of the new jet at $x = l$, and h_0' the initial height at $x = 0$, or

$$v_0^2 = 2g(h_n' - h) + v_1^2$$

hence

$$v_0^2 = 2g(h_n' - h_0') + v_1^2$$

or

$$\frac{v_0^2 - v_1^2}{2g} = h_n' - h_0'$$

But from [2],

$$\frac{v_0^2}{2g} = h_n - h_0$$

hence

$$\frac{h_n' - h_0'}{h_n - h_0} = 1 - \frac{v_1^2}{v_0^2} \quad [14]$$

We can now obtain the curve by the equation:

$$x = \frac{A}{bc_n}\left[\frac{\sqrt{h_n' h_0' - h_0'^2}}{h_0'} - \frac{\sqrt{h_n' h - h^2}}{h}\right]$$

or

$$-x = -C' + \frac{C_1\sqrt{h_n' h - h^2}}{h}$$

where

$$C_1 = \frac{A}{bc_n} \quad \text{as before}$$

and

$$C' = \frac{A}{bc_n}\frac{\sqrt{h_n' h_0' - h_0^2}}{h_0'}$$

hence

$$h = \frac{h_n' C_1^2}{(C' - x)^2 + C_1^2}$$

and the whole curve of the liquid sheet appears lowered with respect to the original sheet. It is easy to calculate the limiting heights of the sheet as before, the result being given by the equations

$$h_0' = \frac{(v_0^2 - v_1^2)A^2}{2g(lbc_n)^2}$$

$$h_n' = \frac{v_0^2 - v_1^2}{2g}\left[1 + \frac{A^2}{(lbc_n)^2}\right]$$

If we replace the slot by a row of holes each of area a, so that

$$lb = Na \quad \text{and} \quad xb = N_x a$$

where $N = n + 1$ is the total number of holes and N_x is the number of holes over the distance x, then the sheet will be broken up into isolated jets of variable heights h, so that we obtain from

$$h = \frac{h_n C_1^2}{(l - x)^2 + C_1^2}$$

the equation

$$h = \frac{h_n C_1^2}{\left(\frac{a}{b}\right)^2 (N - N_1)^2 + C_1^2}$$

where now

$$C_1 = \frac{A}{bc_n} = \frac{Al}{N_a c_n}$$

hence

$$h = \frac{h_n}{1 + \frac{a^2 c_n^2}{A^2}(N - N_1)^2}$$

If h_f is the loss of height per unit distance through friction, then

$$h = \frac{h_n}{1 + \frac{a^2 c_n^2}{A^2}(N - N_1)^2} - \sum h_f \quad \ldots\ldots\ldots [15]$$

Next put $c = c_n$ = constant. Then, from [3],

$$dv = -\frac{wbdxc_n}{A}$$

or

$$dv = -\sqrt{2gh}\,\frac{b}{A}\,c_n dx$$

From [2]

$$dv = -\frac{gdh}{\sqrt{2g(h_n - h)}}$$

hence

$$\frac{2bc_n}{A}\int_0^x dx = \int_{h_0}^{h} \frac{dh}{\sqrt{h(h_n - h)}} = -\arcsin\left(1 - \frac{2h}{h_n}\right)_{h_0}^{h}$$

whence

$$x = \frac{A}{2bc_n}\left[\arcsin\frac{h_n - 2h_0}{h_n} - \arcsin\frac{h_n - 2h}{h_n}\right] \quad [16]$$

or

$$x = C - \frac{A}{2bc_n}\arcsin\frac{h_n - 2h}{h_n}$$

or

$$\sin(C - x)\frac{2bc_n}{A} = 1 - \frac{2h}{h_n}$$

or

$$h = \frac{h_n}{2}\left[1 - \sin\left\{(C - x)\frac{2bc_n}{A}\right\}\right] \quad \ldots\ldots\ldots [17]$$

This equation is given by Enger. Let us now proceed to apply the method of difference equations to the given problem, assuming that the tube contains at equal distances Δl holes of cross-section a, and that the water escapes in jets of heights h_0, h_1, h_2, h_k, h_{k+1} . . . . h_{n-1}, h_n. Then $h_{k+1} = h_k + \Delta h$, and the velocity of the water through the hole k is w_k and through the following hole $k + 1$ is w_{k+1}. The principle of conservation of energy leads to the equation

$$-\Delta(\tfrac{1}{2}mv^2) = \Delta(mgh)$$

$$\Delta(v^2) = -2g\,\Delta h$$

or

$$(v - \Delta v)^2 - v^2 = -2g\,\Delta h$$

or

$$\Delta h = \frac{1}{2g}[2v\,\Delta v - (\Delta v)^2] \quad \ldots\ldots\ldots [18]$$

Moreover

$$\tfrac{1}{2}mw_k^2 = mgh_k$$

or

$$w_k = \sqrt{2gh_k}$$

The velocity of the water in the pipe arriving at the hole k is v (or v_k), and leaving the hole it is v_{k+1} or $v - \Delta v$; the principle of conservation of mass gives the equation

$$-A\,\Delta v = ac_k w_k$$

$$= a\sqrt{2gh_k}\,C_n\frac{h_k}{h_n}$$

Instead of a we may write $a = b\,\Delta l$; hence

$$\Delta v = -\frac{b\,\Delta l}{A}\sqrt{2gh_k}\,\frac{h_k}{h_n}c_n$$

and

$$\tfrac{1}{2}mv^2 = mg(h_n - h_k)$$

or

$$v = \sqrt{2g(h_n - h_k)}$$

and

$$\Delta v = \sqrt{2g}\,[\sqrt{h_n - (h_k + \Delta h)} - \sqrt{h_n - h_k}] \quad \ldots [19]$$

From the last two equations we obtain

$$\frac{b\,\Delta l c_n}{A h_n} = \frac{\sqrt{h_n - h_k} - \sqrt{h_n - (h_k + \Delta h)}}{h_k^{3/2}} \quad \ldots\ldots [20]$$

The left-hand member of Equation [20] is a constant C, and if instead of h_n we write a, the equation then assumes the form

$$\frac{\sqrt{a - h_k} - \sqrt{a - h_{k+1}}}{h_k^{3/2}} = C \quad \ldots\ldots\ldots [21]$$

In order to put this expression into a more standard form of difference equation, let us write

$$g_k^2 = a - h_k, \quad g_{k+1}^2 = a - h_{k+1}$$

whence

$$g_k - g_{k+1} = C(a - g_k^2)^{3/2} \quad \ldots\ldots\ldots [22]$$

or

$$g_{k+1} = g_k - C(a - g_k^2)^{3/2} \quad \ldots\ldots\ldots [23]$$

If h_0 and $g_0^2 = a - h_0$ are given, then we can calculate from [23] the values g_1 and h_1 from $h_1 = a - g_1^2$. Having determined g_1, we can use [23] again to calculate g_2, and so on. In this way the problem is essentially solved. Equation [22] is a difference equation of the first order, but non-linear. For such a difference equation we have no general theory and no theorems as yet. If we consider g_{k+1} as given, then it is more difficult to calculate g_k or h_k. In order to take the first step let us assume h_n as given and calculate h_{n-1} by means of [20]; $k = n - 1$, and $h_k + \Delta h = h_n$, hence

$$\frac{b^2\,\Delta l^2 c_n^2}{A^2 h_n^2} = \frac{\Delta h}{h_{n-1}^3} = \frac{h_n - h_{n-1}}{h_{n-1}^3} = \frac{h_n}{h_{n-1}^3} - \frac{1}{h_{n-1}^2}$$

Let $\alpha = h_n$, $\beta = \dfrac{b^2\,\Delta l^2 C_n^2}{A^2 h_n^2}$, and $x = \dfrac{1}{h_{n-1}}$; then we obtain the cubic equation

$$\alpha x^3 - x^2 - \beta = 0 \quad \ldots\ldots\ldots [24]$$

We can obtain approximate values of x and h_{n-1} by the following method, assuming $C = C_n$ = constant. Instead of [18] we obtain:

$$\Delta v = -\frac{b\,\Delta l c_n}{A}\sqrt{2gh_k}$$

and by means of [19]

$$\frac{b\,\Delta l c_n}{A} = \sqrt{\frac{h_n - h_k}{h_k}} - \sqrt{\frac{h_n - (h_k + \Delta h)}{h_k}}$$

or using the same abbreviations as before,

$$\sqrt{a - h_k} - \sqrt{a - h_{k+1}} = C\sqrt{h_k} \quad \ldots\ldots [25]$$

Let

$$g_k^2 = a - h_k \quad \text{and} \quad g_{k+1}^2 = a - h_{k+1}$$

then

$$g_k - g_{k+1} = C\sqrt{a - g_k^2} \quad \ldots\ldots [26]$$

or

$$g_k^2 - 2g_k g_{k+1} + g_{k+1}^2 - C^2(a - g_k^2) = 0$$

or

$$g_k^2 - \frac{2g_k g_{k+1}}{1 + C^2} + \frac{g_{k+1}^2 - C^2 a}{1 + C^2} = 0$$

or

$$g_k = \frac{g_{k+1}}{1 + C^2} \pm \sqrt{\left(\frac{g_{k+1}}{1 + C^2}\right)^2 - \frac{(g_{k+1}^2 - C^2 a)}{1 + C^2}} \quad \ldots [27]$$

For every value g_{k+1} we can calculate g_k and $h_k = a - g_k^2$. This value we can use as a first approximation and calculate a second approximation by means of Equation [20].

Discussion

W. E. Howland.[2] The writer has been interested in this problem in its relation to the design of filter laterals through which water is forced into the bottom of a filter for the purpose of washing the sand. It is important to maintain an approximately equal distribution of water in the base of the filter as expressed in rate of water applied per unit area of horizontal cross-section of sand. This means that for ordinary arrangements of piping the rate of water emitted per unit distance along the pipe should be approximately a constant. If the pressure distribution along the pipe were known, as well as the hydraulic characteristics of the orifice in the pipe under the conditions of the problem, then one could effect the desired uniformity of distribution either by varying the diameter of the holes or their spacing.

The importance of a correct prediction of the pressure distribution curve is of fundamental importance to this practical problem.

Such studies as this of Professor Kunz and the earlier one of Professor Enger throw considerable light upon the problem. The effect of the velocity in the main pipe upon the coefficient of the orifice shown by Professor Enger's studies is especially valuable. Fortunately the practical problem which the writer has named is much simpler than the one considered by Professor Kunz. It involves a differential equation of an ordinary type, and the very interesting method so ably presented by Professor Kunz does not appear to be required.

The fundamental hydraulics of the two problems are, however, the same. The assumptions made at the beginning of the analysis required further study, it is believed.

[2] Assistant Professor of Civil Engineering, School of Civil Engineering, Purdue University, Lafayette, Ind.

Effect of Velocity Upon Coefficient of Discharge

Professor Enger's formula for coefficient of discharge fits the values determined from his experiment very well, but obviously it could not be correct for all cases even though dimensionally consistent as implied by Professor Kunz. Consider the case where the pressure in the pipe was equal to the velocity head. Then the numerator in the equation $c = \dfrac{y - \dfrac{v^2}{zy}}{y} c_e$ is zero. The formula would indicate that no water would flow from the pipe, and yet so long as there is static pressure at the hole, water obviously would flow from the pipe. This is not a criticism of the formula, but rather of the method used for rationalizing it given by the author.

It is to be noted that the author interprets the formula in a slightly different way from that given by Professor Enger. According to Professor Kunz, the formula should be $c = \dfrac{h_n - \dfrac{v^2}{zg}}{h_n}$, where h_n is the pressure at the closed end of the pipe. Considering the case where there is very considerable friction in the pipe—i.e., where the holes are widely spaced—then it is possible that at some orifice the pressure will be the same at the closed end of the pipe. If at this point the pressure were equal to the velocity head, as before, the formula would indicate no discharge, and yet surely there would be discharge from the orifice.

Energy Distribution at the Orifices

Both Professors Enger and Kunz have assumed that the velocity energy of the water in the pipe is largely dissipated. Consider the energy of that portion of the stream within the pipe which is about to leave an orifice. It is equal to $qw\left(h + \dfrac{v^2}{zy}\right)$, where q is the rate of flow from the orifice, w the weight of a cubic foot, h the pressure, and v the velocity. The author assumes that the water leaves the pipe at a rate proportional to $h^{1/2}$, as if the energy of the water leaving were pressure energy only. This would imply that the remaining part of this energy $qw\dfrac{v^2}{zy}$ is lost or is given to the stream of water in the pipe and remaining within the pipe. The author assumes that it is lost, for he says that the differential change in pressure along the pipe is merely $\left(\dfrac{vdv}{g}\right)$. Here he is considering an orifice of differential area. This effect is entirely the result of the change in kinetic energy of that portion of the water which remains within the pipe, as may be shown in the following way:

Let Q_1 be rate of flow of that portion of the water which remains within the pipe. The amount of energy possessed by this water will be considered a constant; i.e., the differential change in the energy of this water is zero. The energy of this water is $Q_1 w\left(hp + \dfrac{v^2}{zy}\right)$

$$Q_1 w\left(dhp + \frac{2vdv}{zy}\right) = 0$$

$$dhp = -\left(\frac{vdv}{g}\right)$$

The writer believes that this assumption may sometimes be wrong; that in many cases there is a significant transfer of energy from the side stream to the main stream which predominates over the loss of energy in the main stream in the vicinity

of the orifice, thus producing a significant increase in pressure in the main stream which the simple Bernoulli equation used by Professors Kunz and Enger cannot explain. The writer is aware that this statement may not seem plausible, and so it is repeated for emphasis. The simple Bernoulli equation in the form of head is wrong as applied to manifold pipes. Actually there is a greater increase in pressure along the line of flow than can be accounted for by means of this equation. The total energy equation must be used to explain the phenomenon.

In support of this assertion the writer quotes the following experiments, with references appended:

(1) The Malishewsky experiments can be explained by assuming a recovery of approximately $^3/_4$ of the initial velocity energy of the water of each side stream. The Enger-Kunz theory does not explain the results of the experiment.

(2) The experiments of Professor Goodenough on the model of ventilation ducts of the Holland tunnel and subsequently upon the tunnel itself indicate that about 40 per cent of the initial velocity energy of the side streams is recovered.

(3) The Enger experiments are well explained by the theory to which the writer is objecting, but when he applies the assumption that 30 per cent of the initial velocity energy of the side stream is recovered he obtains results which are in close agreement with the results of the experiment, and the disagreement is less than the experimental error. In other words, the effect which the writer is mentioning in this case is too small to be significant.

(4) The experiments conducted at the laboratory at Munich under the direction of Thoma in divided flow on a single tee fitting show a noticeable gain in total head in the main stream, which the authors explain in the following way: "One can assume that through the side pipe a greater part of the slow-moving edge layers is scooped off."

This is a slightly different way of looking at the phenomenon from that of the writer and is believed to be of considerable merit, but in effect it is the same, for it assumes that the water leaving the orifice takes with it less than its proportion of the mean velocity energy and also that the water remaining possesses more than its proportional amount of energy. Thus it helps to explain the mechanism of transfer of energy from the side stream to the main stream.

This viewpoint may also help to compose the differences between the Enger and the Malishewsky experiments. In the Malishewsky experiments only about $^1/_{40}$th of the total flow was taken through a single orifice, for there were 40 orifices; in the Enger experiments $^1/_{19}$th. Thus in the Malishewsky experiments a larger portion of the "scooped off" water was obtained from the edge layer.

It is true, as shown by the experiments of Professor Enger, that cases do arise in which the effect which the writer is mentioning is of no practical consequence. But in other cases, particularly that of unequally spaced orifices designed to accommodate the large flows now demanded in water works practice, this effect of an additional velocity-head recovery will probably be very significant.

Further experimental studies of the problem are urgently needed.

References

(1) "Hydraulics of Filter Underdrains," by N. Malishewsky. *Jl. Am. W. W. Assn.*, June, 1927; also in *Water Works*, vol. 66, no. 10, October, 1927.

(2a) "New York's Automobile Tunnel," by Jorgen Aabye. *Ingenioren-Dansk Ingeniorforening*, no. 24, 1928, p. 293.

(2b) "Ventilation of Vehicular Tunnels," by Ole Singstad, paper no. 339 of the World Engineering Congress, Tokyo, 1929.

(3) "Pressures in Manifold Pipe," by M. L. Enger and M. I. Levy, *Jl. Am. W. W. Assn.*, vol. 21, no. 5, May, 1929.

(4) "Untersuchungen uber den Verlust in recht winkligen Rohrverzweigungen," by G. Vogel (in heft I) and "Der Verlust in schiefwinkligen Rohrverzweigungen," by Franz Peterman (in heft II), both from Mitteilungen des Hydraulischen Instituts der Technischen Hochschule, München.

M. L. Enger.[3] The discharge of jets from manifold tubes must be determined in many cases in engineering practice. When high velocities are used in manifold pipes, unexpected results will sometimes occur. The wash-water system of water filters should supply water uniformly over the entire filter area if the filter is to be washed effectively; and as it is often economical to use high velocities in the manifold pipes, the problem has

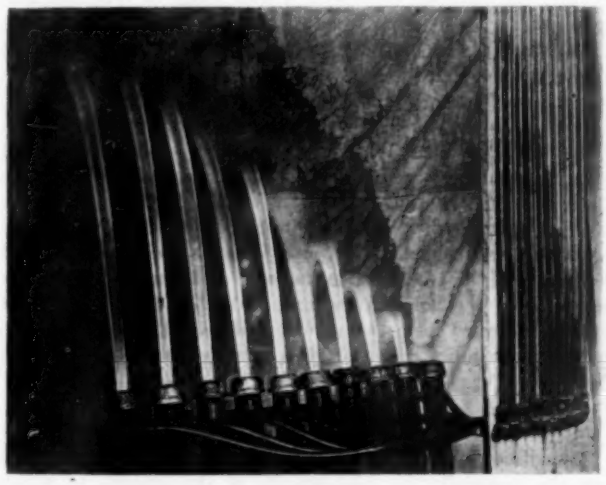


Fig. 2

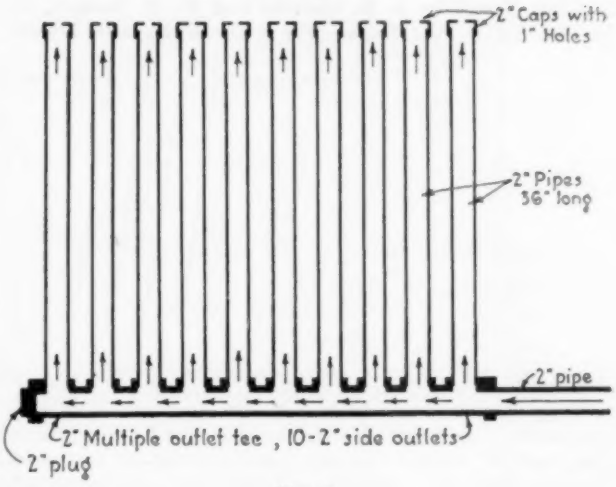


Fig. 3

been given much study in this connection. The problem of delivering the water uniformly beneath the filter bed has been solved in three different ways: (*a*) By the use of large manifold pipes, thus making the velocities small; (*b*) by the use of manifold pipes of a variable diameter, thus keeping the velocities in the manifold nearly constant; and (*c*) by the use of variable spacing of the discharge openings from the manifold pipes. The first two methods are too expensive except in the case of large, costly filter plants.

Another application occurs in hot-water heating when a num-

[3] Professor of Mechanics and Hydraulics, University of Illinois, Urbana, Ill.

ber of radiators are connected to one header. In this case the loss of head at the entrance is also a function of the velocity in the manifold as well as of the velocity of the diverted water.

The converse problem occurs in the case of the wash-water collecting pipe for pressure filters. The method used by the author can be applied to this case.

The writer is particularly interested in the derivation of the equation for the coefficient of discharge from openings in the manifold pipe in terms of the pressure and the velocity in the pipe. The equation is the same as the one derived by the writer from experiments and which was assumed to be purely empirical. Further experiments will soon be undertaken to determine whether the equation holds over a wide range of velocities and pressures.

An extreme case of the variation of pressure and discharge from a manifold pipe is shown in Fig. 2, which is from a photograph taken in the Hydraulics Laboratory of the University of Illinois. A diagrammatic sketch of the arrangement used is shown in Fig. 3. Ten 2-in. pipes, each 36 in. long, are connected to a manifold pipe, which was lined with cement to make its inside diameter exactly 2 in. On the ends of the 2-in. laterals are caps in which 1-in. holes have been drilled. The end of the manifold was plugged. The water entered from the right and flowed out of the 1-in. holes in the caps. The heights of the jets issuing from the holes in the caps indicated the quantities of water discharged from the various orifices. The heads on the orifices were indicated in 10 glass tubes which can be seen in Fig. 2.

Author's Closure

The discussion by W. E. Howland contains some valuable remarks, especially the bibliography of the subject, to which there should be added: "Friction Heads in One-Inch Standard Cast-Iron Tees," by F. E. Giesecke and W. H. Badgett, *Jl. Am. Soc. Hig. & Vent. Engrs.* The results of this latter investigation have been recalculated and show that the loss of head due to water flowing out of the side of a tee can be expressed over a large interval by $h = 0.91\ v^2/2g$ in accordance with our theory. In the paper the author has neglected the friction of the flowing liquid except in formula [15]. The analysis is therefore only a first approximation. Against the criticism of Mr. Howland, the author has to raise only few objections.

(1) The discharge coefficient c is equal to:

$$c = \frac{h_n - \frac{v^2}{2g}}{h_n} c_n = c_n \frac{h}{h_n};$$

c would only vanish if h were equal to zero. This cannot happen in the phenomenon studied; but if it did happen, then no liquid would flow through the given hole.

(2) Bernoulli's equation applies only to a given streamline, and not to dividing streamlines; it can therefore not be used in the present problem and Enger and the author have not used it.

(3) The author cannot understand the statement by Mr. Howland that in many cases there is a significant transfer of energy from the side stream to the main stream, which predominates over the loss of energy in the main stream in the vicinity of the orifice. How is this compatible with the principle of conservation of energy?

(4) In the theory of N. Malishewski there are found the formulas

$$dh = \frac{K(Q + qx)dx}{d^5}, \qquad h = \frac{K}{d^5}(Q_1 + 0.55Q_2)^2$$

which the author does not understand. But the phenomenon, which he observed, agrees qualitatively at least with the author's theory.

A comprehensive critical review of the present literature will be presented in a later paper, the author hopes.

APM-53-15

The Yield Point of Mild Steel

By C. W. MacGREGOR,[1] EAST PITTSBURGH, PA.

This paper is a review of some of the more important investigations made concerning the yield point, together with the effect of certain conditions on the yielding of mild steel. Recent investigations concerning the yield point are also described. The pronounced effect on the yield stress of grooves inclined at 45 deg. to the axis of the test piece is discussed. In addition, tests are included to show the effect of notches in the test piece on the yield stress.



IT IS customary to base the design of structures and machine parts made from ductile metals and subjected to static loading on the limit of plasticity. The limit of plasticity is, as its name implies, that stress at which plastic flow of the material begins. At this stress the deformations begin to become permanent, and increase at practically constant stress. It has been customary to call this limit the yield stress of the material. It is then obvious that one of the reasons for choosing the yield stress as a criterion in design is that such permanent deformations as those which occur at this stress are prohibitive in engineering practice. For structural-steel shapes submitted to a compressive stress such as columns and beams, buckling is likely to occur if they are stressed above the yield point. This is true since on the exceeding of the yield stress the modulus of elasticity E can no longer be used in Euler's simple column formula and the critical load under which buckling occurs is greatly reduced. It is therefore apparent that a more exact knowledge of what is taking place at the yield point and of the various factors affecting its value may be of considerable aid to the designer.

The limit of plasticity, or yield point, has been a subject of much controversy, both in this country and in Europe. At different times many definitions have been proposed, and then subsequently revised. Moser[2] and Körber[3] have indicated the difference in significance of the definitions of a yield point based on an arbitrary amount of deformation and that based on the drop of the beam of the testing machine. They have pointed out that the yield stress of mild steel in tension based on the drop of beam is a natural limit or property of the material, while the other is only arbitrarily based on a conventional rule. In the following discussion, when the yield point is mentioned, reference is made only to the limit of plasticity and not to that corresponding to an arbitrary amount of deformation.

However, it is not the purpose of this paper to analyze the conventional methods of defining the yield point or to present new definitions in this connection. It is more the intention to review a few of the important investigations made by other experimenters concerning the physical aspect of the yield point and to contribute to this the results of a few tests that were made on the same question. These tests will show the effects of certain mechanical conditions, such as grooves and notches, on the yield stress.

Previous Work on the Physical Side of the Yield Stress by Other Investigators

Since the beginning of the twentieth century, much attention has been paid to the yield point. Among the first to investigate this question in Germany were Bach and Martens. Later such

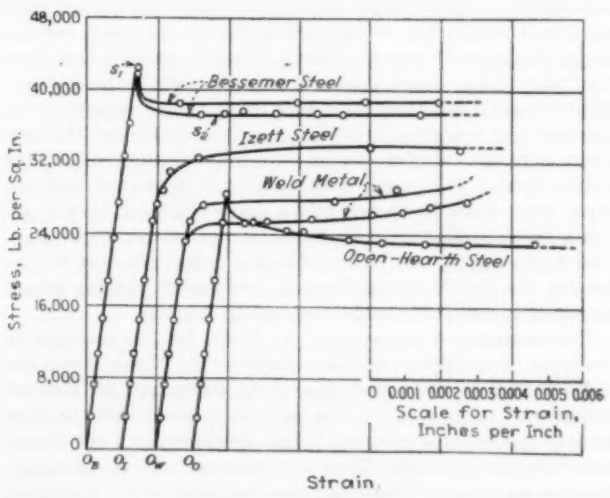


Fig. 1 Portions of Stress-Strain Curves for Different Materials Tested in Tension

men as Moser, Nádai, Körber, Ludwik, and Sachs, with many others, took up this question. Recently Kühnel, Kuntze, and Schwinning have been added to the list. The field of work covered by these many investigators is too broad to go into in much detail, and so only a few of the important results will be described.

(*a*) *Mechanism of Yielding.* C. von Bach[4] seems to have been the first to notice the remarkable phenomenon of the upper and lower yield stresses present in mild steel. He defined the upper yield limit as the stress which, if exceeded, would start the flow process, and the lower yield limit, the lowest value of stress at which the extension proceeds still further. Thus, as represented in Fig. 1, the stress s_1 is the upper yield stress and s_2 the lower yield stress.

In order to explain more clearly what is meant by the flow proc-

---

[1] Research Engineer, Westinghouse Elec. & Mfg. Co. Jun. A.S.M.E. The author was graduated from the University of Michigan in 1929 with B.S. degrees in Engineering and Mathematics. He studied under Dr. S. Timoshenko in applied mechanics during this time. After graduation he became associated with the Westinghouse Research Laboratories under the supervision of Dr. A. Nádai of the Mechanics Division, who is also directing his graduate work at the University of Pittsburgh.

[2] M. Moser, "Grundsatzliches zur Streckgrenze," *Forschungsarbeit, V.D.I.*, 1927, p. 75.

[3] F. Körber, "Das Problem der Streckgrenze," Internationaler Kongress für die Materialprüfungen der Technik, Amsterdam, 1927. (Both authors use the term "Dehngrenze" to signify an arbitrary value of stress corresponding to 0.2 per cent plastic strain and the term "Streckgrenze" to signify the yield limit, or the common yield point observed in mild steels by the drop-of-beam method.)

Presented at the National Applied Mechanics Meeting, Purdue University, Lafayette, Ind., June 15 and 16, 1931, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[4] C. von Bach, *Z.V.D.I.*, 48 (1904), p. 1040.

ess mentioned by Bach, the formation of Lueders lines, or slip layers, in mild steel will be briefly discussed. Many investigators have studied these slip layers which begin to form at the upper yield stress s_1 and continue to form through the lower yield point s_2. Among the early investigators of this question are included Hartmann,[5] Lueders,[6] and Martens and Heyn,[7] and others.

It has been found that these Lueders lines consist of wedge-shaped slip layers which extend through the material and are not just a surface phenomena. They initially form at a point along the test bar where there is a slight concentration of stress, and gradually proceed along the bar until the entire length has become work-hardened. When this has been effected, the load can start to increase to its maximum value, since no more single layers are forming. These layers nearly coincide with the planes of maximum shear stress which are theoretically at 45 deg. to the bar axis, but which have been found by various investigators to form at 47 deg. for tension and 43 deg. for compression.

There is considerable variation in the type of yielding even among mild steels. Some of these metals exhibit a sharply defined yield point with considerable drop in load, while others have a more gradual transition from the elastic to the plastic stage. The reasons for the difference in shapes of the various types of stress-strain curves near the yield point have not been definitely settled. An explanation for the difference in many cases has been suggested by A. Nádai[8] on the basis of the difference in strength of the grain-boundary substance. It has been observed that materials which exhibit a sharply defined yield point in a tensile test with considerable drop in load usually have a fine grain structure, while those with a gradual transition from the elastic to the plastic states are often coarse-grained. Since the grain-boundary substance forms a rigid cement about the grains in a fine-grained mild steel, the ability for the flow layers to form is thus prevented until the skeleton-like structure breaks down. This means that yielding is prevented until the upper yield point is reached and a sharply defined limit is obtained. In coarse-grained materials, or metals having less grain-boundary cement, this latter substance is less effective in preventing the flow layers in forming, and earlier yielding occurs without a sudden break in the stress-strain diagram.

This suggested explanation of A. Nádai for the variation in the types of transition from the elastic to the plastic states has found further corroboration through G. Sachs and W. Kuntze[9] and others. Kuntze and Sachs have emphasized that the yield point in mild steel is governed by the breaking down of a brittle constituent in the steel. They have observed that in the majority of cases yielding first begins near the head of the test piece, where more or less stress concentration[10] may be present, depending on the actual shape of the head of the bar. One would expect that when the yield point is reached in a ductile material, the stresses would become more evened out and not affect so much the yielding of the entire bar. However, they have pointed out that this phenomenon of first yielding at a point of stress concentration is analogous to that which happens in the failure of a brittle material, in that brittle materials are much more subject to the effects of stress concentration than ordinary ductile metals. Where in a brittle material cracks form due to stress concentration, slip layers form due to the same cause in a ductile metal.

At the same time that these slip layers are forming, the cross-section of the test bar is decreasing. Kuntze and Sachs[11] have shown by very illustrative profile diagrams how the area of cross-section gradually decreases along the test bar with progressive yielding.

(b) Effect of Certain Mechanical Conditions on the Yield Point. It has been shown that mild steel exhibits the phenomenon of an upper and a lower yield point. The question of whether to use the upper or lower value of stress as a limit is, in many countries, not definitely settled. In Germany, Kühnel and others advocate using the upper value, while Moser, Körber, and others believe that the upper yield stress is unstable and can be affected by testing conditions. The latter therefore advocate using the lower yield stress.

Many investigations have shown the effects of various conditions on the yield point. Some of the more important factors which have been found by other investigators to affect the value of the yield stress will be mentioned here.

Velocity. This is a large question, and only a few phases of it will be discussed. Among a large number of tests may be mentioned those of E. H. Schulz and H. Buchholtz,[12] together with those of F. Körber.[13] It was found that both the lower and the upper yield stresses increased with greater testing speed. However, the upper yield point showed itself much more unstable and affected by increasing velocity than the lower stress.

In this connection the tests of Kühnel[14] may also be quoted. He arrives at different results from those of the former tests mentioned. His test results seemed to indicate that the lower and upper yield stresses increased in a parallel manner with increasing speed. From this fact he concludes that the upper yield stress could hardly be called unstable and the lower stable. However, by consulting his diagrams for St-48 steel tested under various mechanical conditions, it appears that the lower yield stress is much less affected by variations in testing conditions than the upper, indicating that the lower yield stress is the more stable of the two.

Type of Head on Test Bar. It has been shown by F. Körber[15] and M. Moser[16] that the sharper the transition from the head of the test piece to the cylindrical portion, the lower will be the upper yield stress. The change of section from the head of the test piece to the cylindrical length does not affect the lower yield stress to any great extent. The effect of no head on the test piece is similar to that with a sharp transition. This follows since there is a sharp discontinuity at the point of the bar where it joins the grip of the testing machine, and consequently high stress concentrations similar to a sharp fillet exist.

Notches in Test Bars. Among a large number of tests made on the effects of notches in test pieces the more recent tests of A. Nádai,[17] P. Ludwik and R. Scheu,[18] and W. Schwinning[19] may

[5] L. Hartmann, "Distribution des deformations dans les metaux soumis a des efforts," Paris; Berger-Levrault, 1896.

[6] Lueders, *Dingler's Polytechnisches Journal*, 1854.

[7] Martens-Heyn, Materialienkunde, 1, Aufl.

[8] A. Nádai, "Der Bildsame Zustand der Werkstoffe," J. Springer, Berlin, 1927, p. 63.

[9] W. Kuntze and G. Sachs, "Zur Kenntnis der Streckgrenze von Stahl," *Z.V.D.I.*, 1928, p. 1014.

[10] By stress concentration here is meant the raising of the stress by means of irregularities or sudden changes in the cross-section, above the average uniform value when the load is uniformly distributed.

[11] Loc. cit.

[12] E. H. Schulz and H. Buchholtz, "Mitteilungen aus den Versuchsanstalten der Vereinigten Stahlwerke," Dortmunder Union-Hoerder Verein, 2 (1926), p. 1.

[13] F. Körber, "Zwanglose Mitteilungen des D.V.M.," No. 8 (1926), p. 91.

[14] Kühnel, "Die Benutzung der Streckgrenze bei Berechnung und Abnahme," *Z.V.D.I.*, 1928, p. 1226.

[15] Loc. cit.

[16] Ibid.

[17] A. Nádai, "Untersuchungen der Festigkeitslehre mit Hilfe des thermoelektrischen Temperaturmessverfahrens," Dr. Ing-Diss. Techn. Hochsch., Berlin, 1911.

[18] P. Ludwik and R. Scheu, "Über Kerbwirkungen bei Flusseisen," *Stahl und Eisen*, Aug. 2, 1923, p. 126.

[19] W. Schwinning, "Beurteilung von Werkstoffen nach Kerbversuchen," *Z.V.D.I.*, March 9, 1929, p. 321.

be mentioned. The effect of the raising of the ultimate strength by short notches in bars of mild steel is quite widely known. As this discussion is confined to the yield point, only those tests will be described which show an effect on it. Both A. Nádai and W. Schwinning have shown that the lower and upper yield stresses are increased by short notches, while the difference between the two is constantly decreasing. For very short notches they practically coincide. This fact may be noticed by reference to Fig. 16, which shows the load-deformation diagrams for bars containing grooves of different lengths.

Cross-Section of Test Bars. In some of his early tests Bach[20] investigated the effect of the cross-section of the bar on the upper yield stress. His tests have shown that the highest upper yield stress was obtained with round bars, the next with rectangular bars, and the lowest with a double T-shaped cross-section. This is easily explained since the round bar has the most uniform distribution of stresses over the cross-section, and hence a higher value of yield stress may be reached due to a prevention of local yielding. In all cases the lower yield stress was not appreciably affected by a change of cross-section.

Aging of Test Pieces.[21] Another factor influencing the yield point is the aging of the steel. Much work has been done on this question in the last few years by various men. Among them may be mentioned P. Ludwik and R. Scheu,[22] who have illustrated how the yield point of various metals may be greatly increased due to aging. Steel bars were stretched to a few per cent elongation, unloaded, heated to various low temperatures for a short time, and then reloaded. On reloading, the yield point was considerably raised. Many other important investigations have been carried on in this field which will not be mentioned here.

Recent Tests on the Yield Point

In the following, tests will be described showing the changes which take place on polished surfaces of various steels at the yield point, and in addition the effect of surface conditions and notches in the test piece on the yield stress.

(*a*) *Sharpness of Transition From Elastic to Plastic States in the Stress-Strain Diagram in Relation to the Formation of Lueders Lines.* The fact that the upper yield stress is associated with the formation of Lueders lines has been mentioned before. It has been noticed, however, that certain mild steels do not show these strain lines as well as others. In order to show that the most pronounced Lueders lines appear in steels which have a sharp transition from elastic to plastic states, tensile tests were made on four materials—namely, a bessemer steel, an open-hearth steel, an Izett steel,[23] and an arc-deposited weld metal.[24] All of these metals were of low-carbon content. The stress-strain curves for each material are shown in Fig. 1. From these curves it is to be noticed that the bessemer steel shows the most sharply defined yield point. The materials may be listed in the order of sharpness of transition from the elastic to the plastic states in the following manner, those with the sharpest transition being placed first: bessemer, open-hearth, weld metal, and Izett steel.

In order to correlate these results with the ability of each steel to show Lueders lines, or slip layers, tests were made on cylinders of each material stressed in two different manners. In the first case, cylinders of Izett and bessemer steel with highly polished faces were compressed laterally between hard plates. The cylinders after they were stressed are shown in Figs. 2 and 3. A comparison of these photographs with the stress-strain curves of Fig. 1 indicates that the steel with the most sharply defined yield point shows many more Lueders lines than the other material. The Izett steel indicates a boundary of the elastic and plastic regions, but no individual slip layers. The other manner of stressing the test pieces was to press a small, hard cylinder axially on the polished surface of each. Fig. 5 shows the slip lines produced by this type of loading in the four different materials in the form of logarithmical spirals. The test pieces in each of these photographs originally came from the same material from which were taken the tensile test pieces whose curves are shown in Fig. 1. By comparing these last results with the curves of Fig. 1, it follows immediately that the material with the sharpest transition from the elastic to plastic states exhibits the most pronounced Lueders lines. It is more difficult to distinguish between the weld metal and the Izett steel, but for this material it appears that the weld metal has a sharper transition in the stress-strain curve and consequently shows more slip lines than the Izett steel. Somewhat better flow layers were exhibited by a test piece from a different bar of Izett material than that shown in Figs. 2 and 5. This example is shown in Fig. 4.

Along this line of investigation tests are now in progress concerning the effect of nitrogen on the ability to etch these flow figures. This work is being done with the cooperation of Dr. Hensel, of the Metallurgical Division of the Research Laboratories, and will be reported at a later time.

(*b*) *Effect of Surface Conditions of Test Bar on the Upper Yield Stress.* It has been generally known for some time that small scratches will greatly reduce the strength of a brittle material. As mentioned in the discussion of the mechanism of yielding, the yield point of mild steel seems to be governed by the breaking down of a brittle constituent in the steel. It is naturally not to be expected that for such a ductile metal as mild steel, small scratches would have much effect. However, it is still conceivable that if it is true that the yield point is governed by a brittle constituent, small scratches might still affect the upper yield point.

It has been shown in many tests by A. Nádai that a slight stress concentration even in a ductile metal such as mild steel produces flow layers and local yielding. Fig. 6 shows a flat bar with grooves along the sides which has been tested in tension and from which it can be readily seen how the flow layers have originated at the corners of the grooves. This early local yielding means that the upper yield stress may be thus reduced by such stress concentrations.

In order to try to determine the effect of small scratches and grooves on the upper yield stress, tests were made on several bars of the types shown in Fig. 7. The round bars *a* and *b* were highly polished and designed so as to have a very small amount of stress concentration. This was done by providing a very gradual change of cross-section at the heads. Test piece *b* is identical with *a* except that it contains four small grooves around the circumference of the bar in two places. These grooves are perpendicular to the axis of the bar. The flat bar *d* is provided with a large number of sharp grooves inclined to the axis of the bar on the two faces at an angle of 45 deg. and perpendicular to the bar axis on the two sides. The results of the tests on these bars are recorded in Table 2.

The highly polished bar of type *a* showed a large difference between lower and upper yield stresses. The bar containing small grooves perpendicular to the axis *b* showed about the same yield-

[20] C. v. Bach, *Z.V.D.I.*, 49 (1905), I, 615; Mitt. Forshungsergebnisse Heft, 29 (1905), I, 61.

[21] There are several other factors which have been found to affect the yield point which will only be referred to here. Such items as the type of machine used, the length of test piece, etc., are known to have some influence.

[22] "Werkstoffausschussbericht des Vereins deutscher Eisenhüttenleute," No. 70, Düsseldorf, 1925.

[23] Furnished by F. Krupp, Essen, Germany.

[24] Formed by electrically depositing weld metal in the shape of a pad from which test pieces were cut.

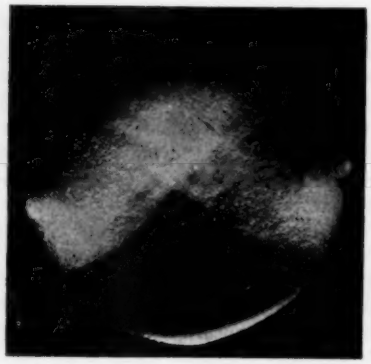


FIG. 2 CYLINDER OF IZETT STEEL COMPRESSED LATERALLY
(Showing elastic and plastic regions with no definite slip layers.)

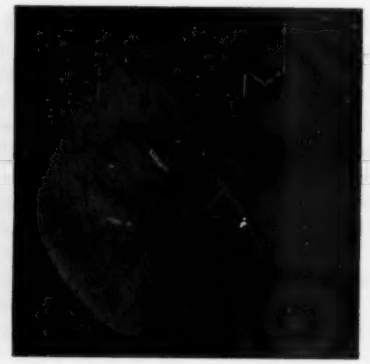


FIG. 3 CYLINDER OF BESSEMER STEEL COMPRESSED LATERALLY
(Showing many slip layers.)

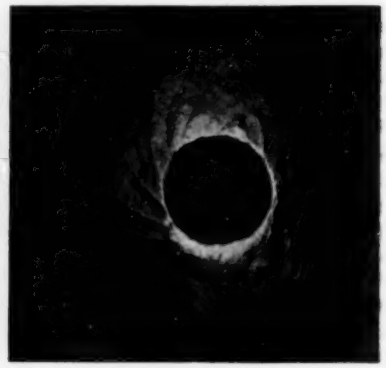


FIG. 4 INDENTATION OF HARD-STEEL CYLINDER ON POLISHED SURFACE OF IZETT STEEL
(This material not the same as that shown in Figs. 2 and 5.)

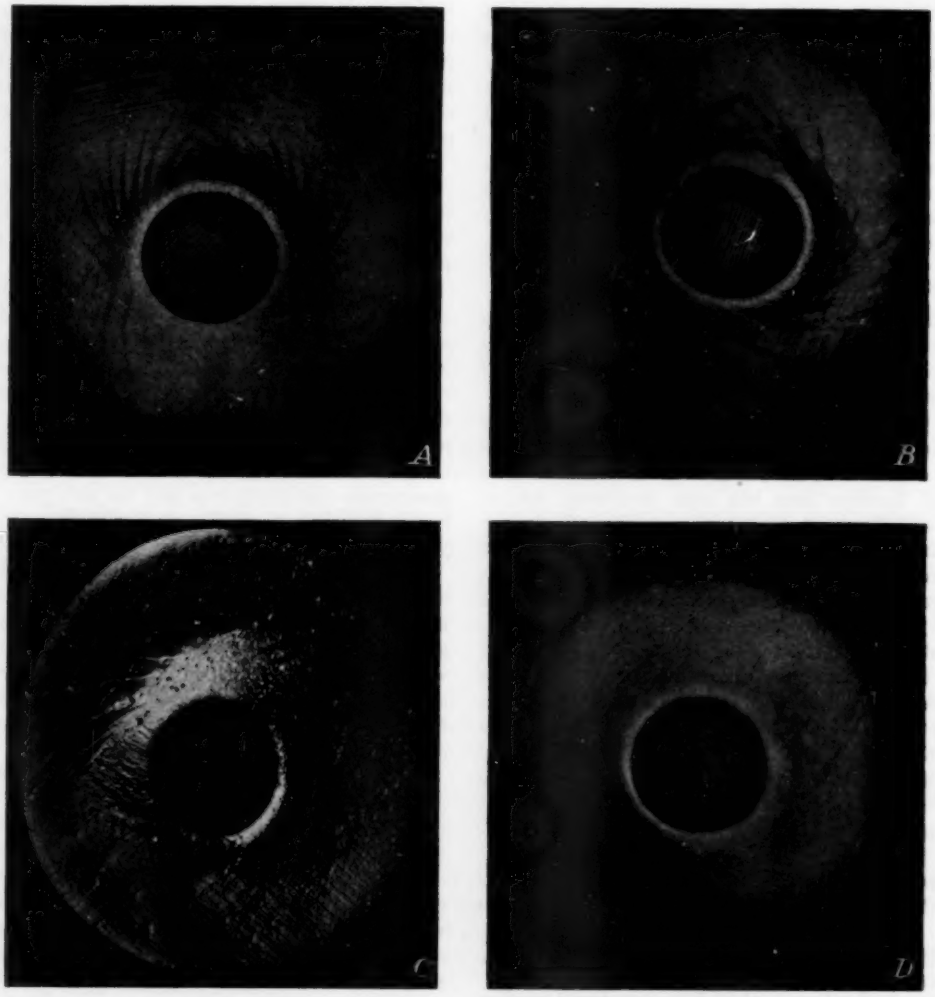


FIG. 5 INDENTATIONS ON POLISHED SURFACES OF VARIOUS STEELS BY A HARD-STEEL CYLINDER
(Showing slip layers in the form of logarithmical spirals. *A*, Bessemer steel; *B*, open-hearth steel; *C*, weld metal; *D*, Izett steel.)

stress difference as type *a*. Hence, single small grooves perpendicular to the bar axis do not seem to decrease the upper yield stress appreciably. However, on close examination of bar *b* after unloading, some local yielding could be distinguished near the grooves. This means that although the grooves had some effect in causing local yielding, it was not great enough to cause a lowering of the upper yield stress. It may be remembered in this connection that small holes through a test piece perpendicular to the axis of the bar have been shown in tests by A. Nádai to produce definite flow layers. The action of these grooves is similar to the case of the holes in producing yielding.

A further test concerning the effect of small scratches on the upper yield point was made with bars of type *c*. These tests were included to show the effect, if any, of scratches due to the knife edges of the Martens extensometer equipment. Some of the bars were tested with the extensometer knife edges inside the part of the bar with reduced section, and others had the knife edges resting on the parts of the bar of larger diameter, the comparison strips spanning

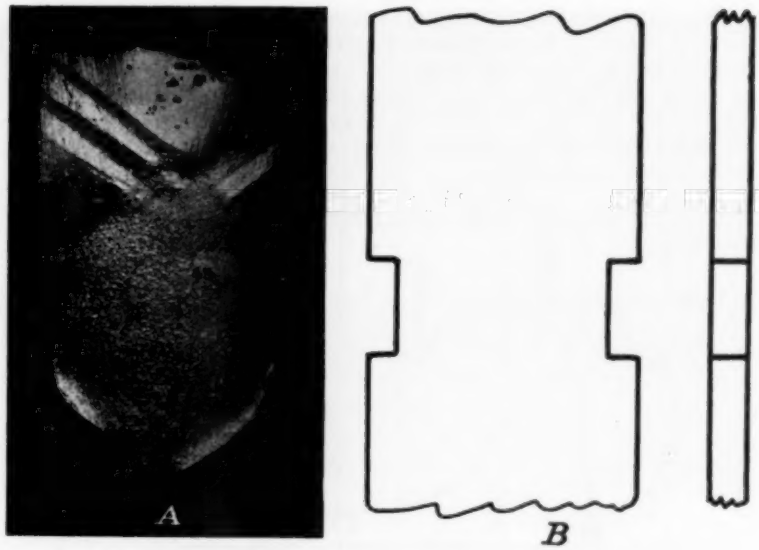


FIG. 6 FLAT BAR CONTAINING TWO GROOVES ON SIDES
(Showing how the flow layers originate from the corners of the grooves. *A*, after deformation; *B*, before deformation.)

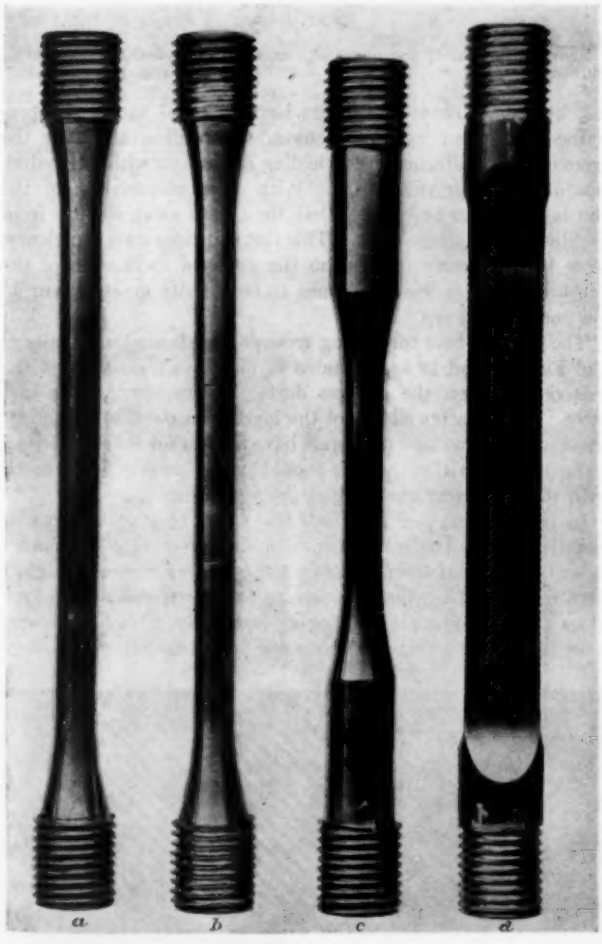


FIG. 7 TYPES OF TENSILE-TEST BARS USED TO SHOW EFFECT OF SURFACE CONDITIONS, SUCH AS SCRATCHES AND GROOVES, ON THE UPPER YIELD POINT

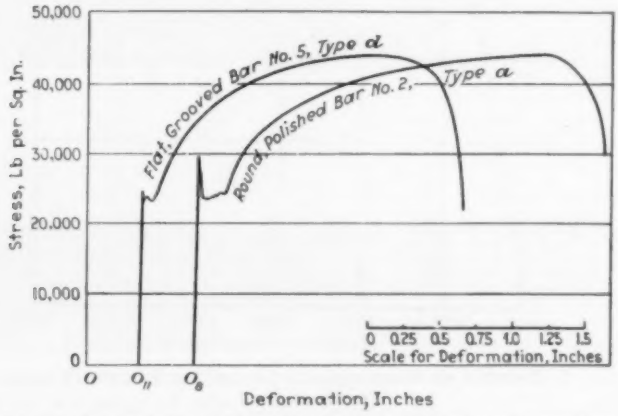


FIG. 8 AUTOGRAPHIC STRESS-DEFORMATION DIAGRAMS FOR TWO TYPES OF TENSILE-TEST BARS

the reduced section. The results were similar to those found for the bar with small perpendicular grooves, in that no appreciable effect on the upper yield point could be noticed.

The results of tests on the flat bars of type *d*, however, were quite different from those obtained in the previous cases. In all of the flat, grooved bars tested, the upper yield stress showed a very marked decrease from that obtained for bars of type *a*. The lower yield stress for both types *a* and *d*, however, was practically the same. Fig. 8 gives a comparison of the stress-deformation diagrams, obtained for these two types *a* and *d*, and shows

TABLE 1 TEST RESULTS OF CYLINDERS WITH HELICAL GROOVES STRESSED IN COMPRESSION

Test piece No.[a]	Final load applied, lb.	Total vertical displacement, per cent	Maximum vertical displacement of vertical lines, in.	Maximum lateral displacement of vertical lines, in.
1	37,450	7.5	0.035	0.025
2	37,450	7.6	0.065	0.025
3	50,000	11.2	0.055	0.04
4	50,000	11.3	0.085	0.04
5	39,000	6.5	0.045	0.025
6	39,000	7.1	0.055	0.025

[a] Nos. 1 and 2 were tested together, and 3 with 4, and 5 with 6. Nos. 1, 2, 3, and 4 are short cylinders with helical grooves; Nos. 5 and 6 are long cylinders with helical grooves.

how the upper yield stress for the flat, grooved bar has approached the lower in value. Fig. 9 shows the type of stress-strain curve obtained for an unannealed round bar of type *a* indicating for comparison the position of the upper yield point of an unannealed flat bar of type *d*. It is thus seen that in every case the upper yield stress for a flat, grooved bar is much less than that obtained for a round bar.

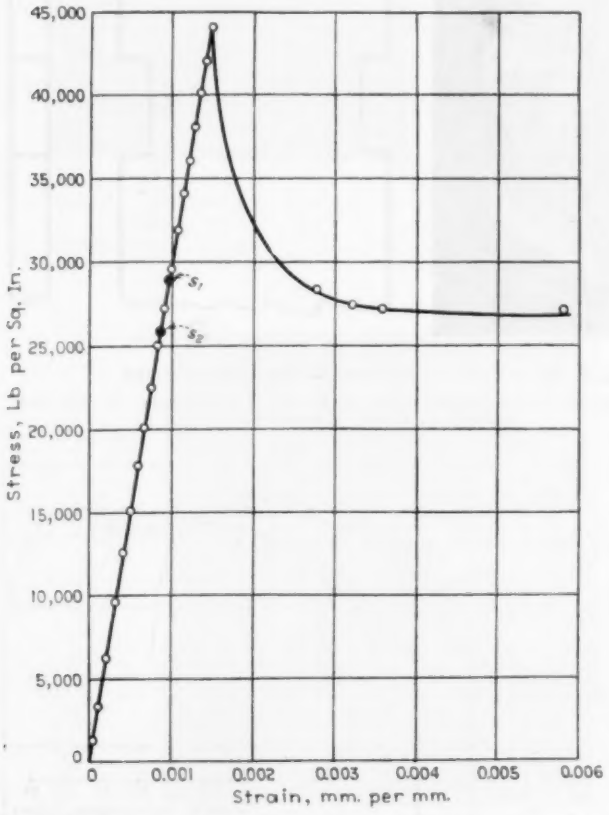


FIG. 9 PORTION OF STRESS-STRAIN CURVE FOR ROUND TENSILE-TEST BAR NO. 9

(s_1, upper yield stress for flat, grooved bar No. 10, s_2, lower yield stress for same.)

The grooves in the flat bars have contributed to a marked degree in reducing the upper yield stress. Bach[25] has shown that the upper yield stress was reduced in changing from a round bar to one with a rectangular cross-section. This reduction, however, was not nearly so marked a decrease as has been obtained with the grooved rectangular bars. A comparison of the differences between the upper and lower yield stresses for a change from round to rectangular cross-section obtained by Bach with those given here shows this clearly. Using data obtained from his tests, the ratio of the difference between the upper and lower yield stresses for rectangular bars to that of round bars is 0.59, while the same ratio for tests shown here is about 0.20. By referring to Fig. 10, the traces of the slip layers in the central non-grooved portion of the bar

[25] Loc. cit.

TABLE 2 TEST RESULTS SHOWING EFFECT OF SURFACE CONDITIONS ON YIELD POINT FOR VARIOUS TYPES OF TEST BARS

Material	Type of bar	Bar No.	Upper yield point, lb. per sq. in.	Lower yield point, lb. per sq. in.	Diff. between upper and lower yield points as per cent of upper	Remarks
2	a	1	32,100	23,800	25.8	Round bar, highly polished
	a	2	29,900	23,400	21.7	Round bar, highly polished
	b	3	31,000	23,200	25.1	Round bar, highly polished; eight grooves perpendicular to bar axis
	b	4	29,000	22,500	22.4	Same as for round bar No. 3
	d	5	24,600	22,900	6.9	Rectangular cross-section, grooved at 45° to axis of bar
3	d	6	24,600	22,800	7.3	Same as for rectangular bar No. 5
	c	7	30,100	26,900	10.6	Round bar, highly polished; knife edges on reduced section
	c	8	29,000	25,900	10.7	Round bar, highly polished; knife edges not on reduced section
1	a	9	44,200	27,200	38.5	Round bar, highly polished
	d	10	29,000	25,700	11.3	Rectangular cross-section; grooved at 45° to axis
	d	11	31,800	28,200	11.3	Same as for rectangular bar No. 10

NOTE: Material 1, not annealed; material 2, annealed; material 3, annealed.

can be seen. These slip layers have followed the course prescribed for them by the grooves, which illustrates how the grooves have influenced the yielding of the bar with the reduction of the upper yield point. With close examination of the slip layers it can be noticed that they bend away slightly from the direction of the grooves. This fact coincides with experience, since in most cases for tension the angle of inclination of the slip layers to the bar axis seems to be slightly greater than 45 deg., or about 47 deg.

The flat test bars containing grooves were loaded until failure, and Figs. 11 and 12 are included to illustrate the sliding of the material between the grooves during the necking of the test piece. The relative sliding of the layers can be seen in Fig. 11, and Fig. 12 shows how the layers have slipped until finally cracks formed between them. This should not, however, be confused with the slip layers occurring at the yield point.

In order to further illustrate the effect of grooves along a direction of the bar coinciding with the direction of maximum shear stress, compression tests on the cylinders shown in Fig. 13 were made. The cylinders with helical grooves allow the load to be applied on one-half of the cylinder only. Each test piece consists of two separate cylinders one above the other. In each

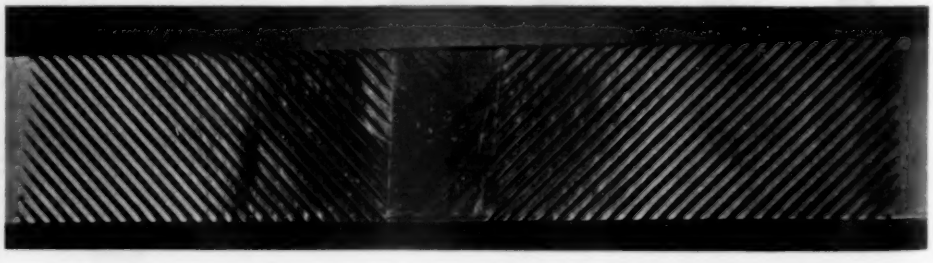


FIG. 10 GROOVED TEST BAR SHOWING HOW THE SLIP LAYERS FOLLOW THE INCLINATION OF THE GROOVES

(The slip layers may be seen in the central rectangular non-grooved part of the bar.)

case the upper and lower test pieces were grooved in opposite directions at 45 deg. to the axis of the bar. When the load was applied, the material between the grooves displaced itself in the same direction in both cylinders, thus minimizing the effect of friction at the common surface of each. Both test pieces have an axial hole bored along the entire length in order to separate the flow layers on opposite sides of the cylinder. Lines were drawn axially along the cylinder. Indentations were provided on the compression faces of each cylinder in order to allow the material between the grooves to slide.

When the load was applied, the material between one pair of grooves displaced itself with respect to that between the adjacent pair. Fig. 14 shows how the test pieces deformed. The amounts of lateral displacement of the vertical lines are given in Table 1. This shows that the material displaced itself along the directions prescribed by the grooves. By so designing the test piece, it is possible to thus produce a rotation of the material about the axis of the cylinder by the application of a pure compressive force. The screw action of the material on opposite sides of the grooves is similar to that between a bolt and nut.

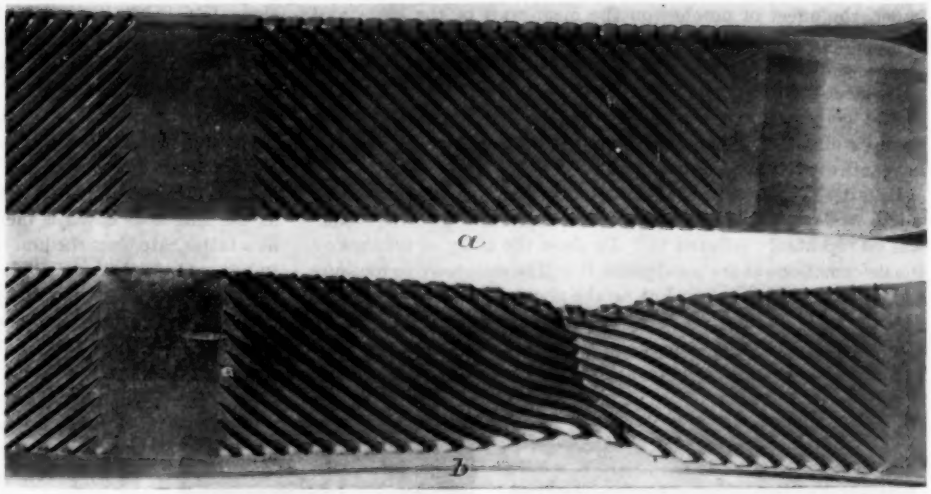


FIG. 11 FRONT VIEW OF GROOVED BAR SHOWING HOW THE LAYERS SLIPPED AT FRACTURE
(*a*, before deformation; *b*, after deformation.)

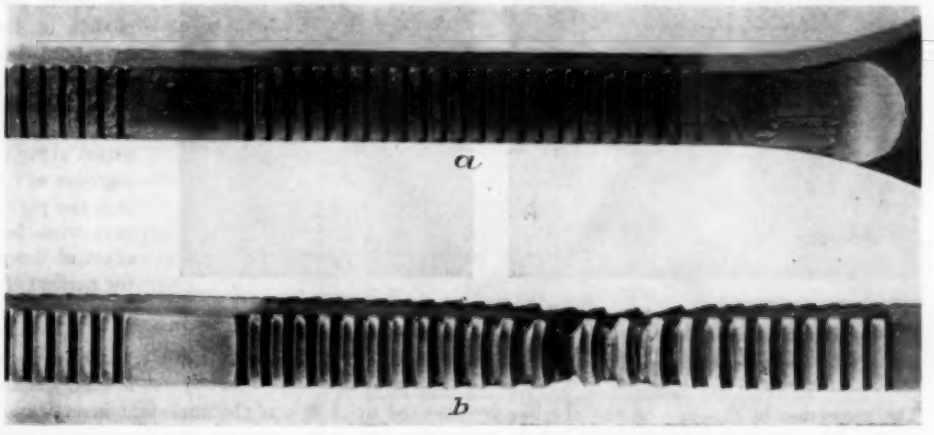


FIG. 12 SIDE VIEW OF GROOVED BAR SHOWING HOW THE LAYERS SLIPPED AT FRACTURE
(*a*, before deformation; *b*, after deformation.)





FIG. 13 COMPRESSION-TEST CYLINDERS WITH HELICAL GROOVES

The helical groove on each test piece is a curve whose osculating plane at every point coincides with a slip layer. In other words, the helical groove prescribes a course for the formation of an infinite number of slip layers. The surface of slip then in this case is a helicoid. On the basis of what has gone before, one would then naturally expect that these grooves should have a marked effect on the upper yield point.

In order to show this, tests were also made on plain cylinders for comparison. Curves shown in Fig. 15 illustrate the difference in the type of yielding in the two cases. Although a well-defined yield point with considerable drop in load was obtained for the plain cylinder, the helically grooved cylinders showed a rather gradual transition from elastic to plastic stages. Local yielding in the formation of slip layers has caused the upper yield point to entirely disappear.

From the tests on surface conditions so far completed it can then be concluded that small scratches do not appreciably reduce the upper yield stress of mild steel. The same is true of small grooves perpendicular to the axis of the bar. However, when grooves of sufficient depth are placed at 45 deg. to the axis of the bar, the upper yield point practically disappears.

(*c*) *The Effect of Notches in Test Piece.* In the first part of the

paper, the effect of notches on the magnitude of the lower and upper yield stresses was discussed. From the autographic curves shown in Fig. 16, it can be readily seen that the notches do not only affect the magnitude of the stresses, but also the magnitude of the deformation. It is therefore the intention for the remainder of the paper to describe in more detail how the notches have affected the deformations.

In order to facilitate the discussion of this question, it will be worth while to define in a short manner a certain quantity which will be repeatedly referred to. To show the effect of notches on the deformations at the yield point, it will be necessary to measure the projection of a certain line on the abscissa of the stress-strain diagram. This line consists in that portion of the stress-strain curve extending from the upper yield point to that point on the curve at which the load starts to rise to its maximum value.

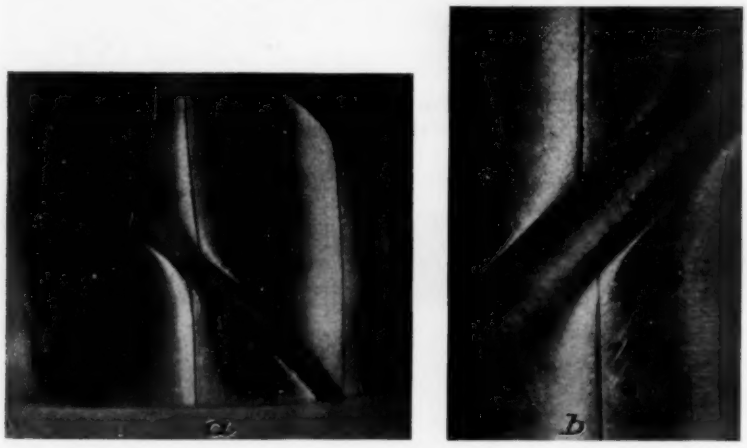


FIG. 14 CYLINDERS WITH HELICAL GROOVES AFTER DEFORMATION SHOWING RELATIVE SLIPPING OF MATERIAL ON EACH SIDE OF GROOVE

(*a*, view of entire test piece; *b*, close-up view of material near the groove.)

The projection of this line on the abscissa is indicated by *A-B* in Fig. 16. Hereafter, this length will be referred to as *A-B*.

For the investigation of the per cent elongation of the length *A-B* with various widths of notches, tensile tests were made on seven bars. Five of these bars contained notches as shown in Fig. 18; the other two were of a uniform cross-section such as type *a*, Fig. 7. The load-deformation diagrams for these bars have already been shown in Fig. 16. The elongation of *A-B* was measured with the Martens extensometer, the knife edges resting on the section of the bar of larger diameter than that in the notch, with the comparison strips spanning the notch. The value of the elongation of *A-B* contributed by the cylindrical portion of the notch in each bar was then estimated. It was possible to obtain this value of elongation for *A-B* without knowing how much of the total value measured between the knife edges was taken up in the fillets, etc. This becomes apparent from consideration of Fig. 18, bar No. 15. The total elongation between *X* and *Y* on bar No. 15 to the upper yield point *A* of Fig. 16 consists approximately of an elastic elongation in each of the regions *L*, *M*, and *N*. After the upper yield point has been passed and the bar has stretched to point *B*, the regions *L* and *N* are still sensibly elastic, and their elongations have not changed appreciably, since the load is still about the same. Region *M*, however, has both an elastic elongation, and in addition the plastic portion consisting of *A-B*, Fig. 16. If now the total elongation measured between *X* and *Y* to the upper yield point *A* and after the upper yield point is passed to point *B* are subtracted from each other, the result is the average plastic elongation in the notch, or *A-B*. This elongation so found for each test piece was then divided by the cylindrical length of its respective notch, and the average per cent elongation plotted as a function of the length of notch in Fig. 17.

In referring to this latter figure, it is noticed that the per cent elongation of *A-B* decreases as the length of notch increases until a certain length of notch has been reached. For short notches the lateral contraction of the cross-section is prevented. Thus, as the length of notch is increased, the gage length is increased at a faster rate than the amount of elongation, causing a decrease in per cent elongation. This effect continues until such a length of notch is reached that the prevention of lateral contraction of the test piece in the notch is no longer very effective over a larger portion of it. From this point on, as the length of notch is further increased, material is being added which is doing the same amount of extending in per cent as for the bar of uniform cross-section. Hence for bars with long notches, the per cent elongation of the quantity *A-B* approaches the case of the bar with a uniform cross-section, as shown in Fig. 17.

In order to get a clearer picture of how the per cent elongation changes in the neighborhood of the notch, tests were made on some flat rubber models of the bars shown in Fig. 18. After stretching these rubber models a certain amount, the per cent elongations were measured, both along the outside edges and along the middle. Fig. 19 shows the variation of the per cent elongation along the notch for a rubber test piece with a groove of 1 in. length, and it can be readily seen that the per cent elongations in the notched portion were far from uniform. This shows the effect of the prevention of lateral contraction in the portion of the bar of smaller diameter.

SUMMARY

A brief discussion has been given of the yielding of mild steel, together with a brief review of a few of the important investigations which have been carried on relative to the upper and lower yield points.

The effects on the yield point of velocity, type of head on the test piece, different kinds of cross-sections, etc. were briefly described.

To this discussion were added tests both in tension and compression, to show the effect of scratches and grooves on the upper yield point. It was found that small grooves and scratches perpendicular to the bar axis did not appreciably affect the upper yield stress.

However, if grooves were placed at 45 deg. to the axis of the bar, there resulted a marked change in the shape of the stress-strain curve in the neighborhood of the yield point. The upper yield stress was also greatly reduced.

Tests were also included to illustrate how the sharpness of transition from the elastic to the plastic state affected the resulting Lueders lines. It was found that materials having a sharply defined yield point showed the best slip layers.

Finally, tests on notched bars were described showing the effect of notches on the per cent elongation of a portion of the stress-strain diagram after the upper yield point has been passed.

ACKNOWLEDGMENT

The author wishes to acknowledge the valuable aid given him by Dr. A. Nádai, who has supervised the investigation. Acknowledgment is also due Mr. J. M. Lessells for encouragement of the work and to Mr. P. G. McVetty and Mr. K. R. Waugh for assistance in the various tests.

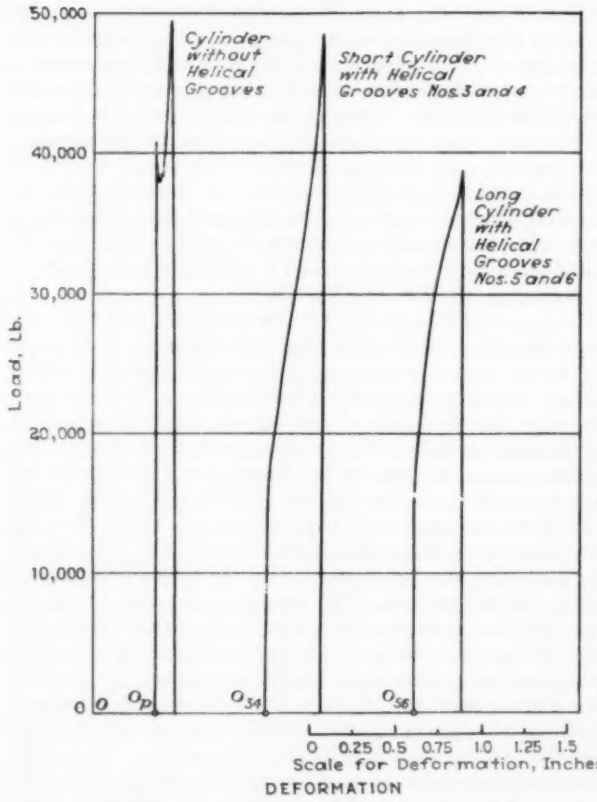


FIG. 15 PORTIONS OF LOAD-DEFORMATION DIAGRAMS FOR DIFFERENT TYPES OF CYLINDERS TESTED IN COMPRESSION

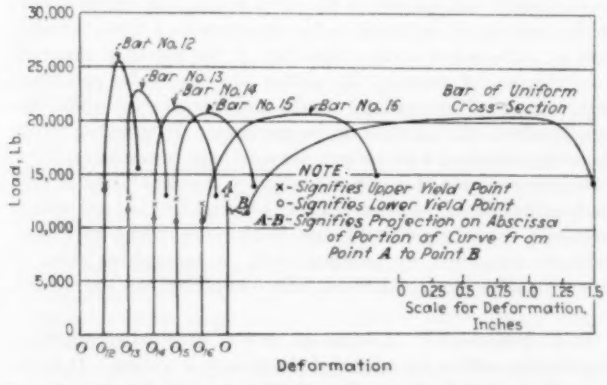


FIG. 16 TYPES OF LOAD-DEFORMATION DIAGRAMS FOR THE NOTCHED BARS OF FIG. 18 TESTED IN TENSION

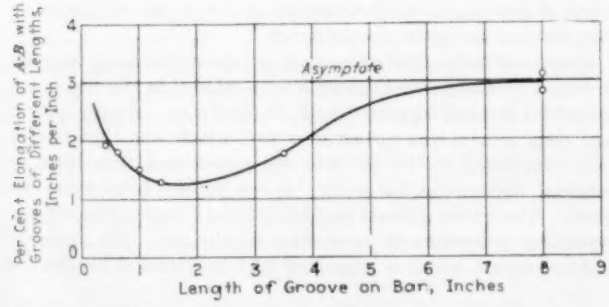


FIG. 17 VARIATION OF PER CENT ELONGATION OF LINE *A-B* OF FIG. 16 WITH GROOVES OF DIFFERENT LENGTHS

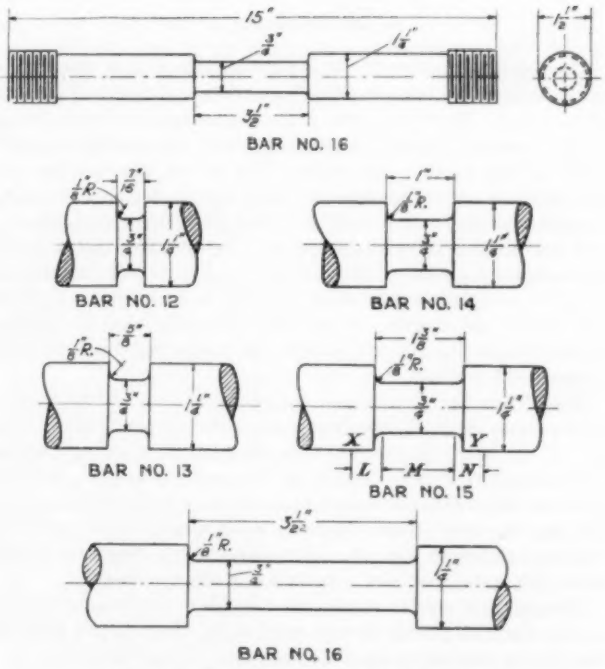


FIG. 18 VARIOUS NOTCHED BARS TESTED

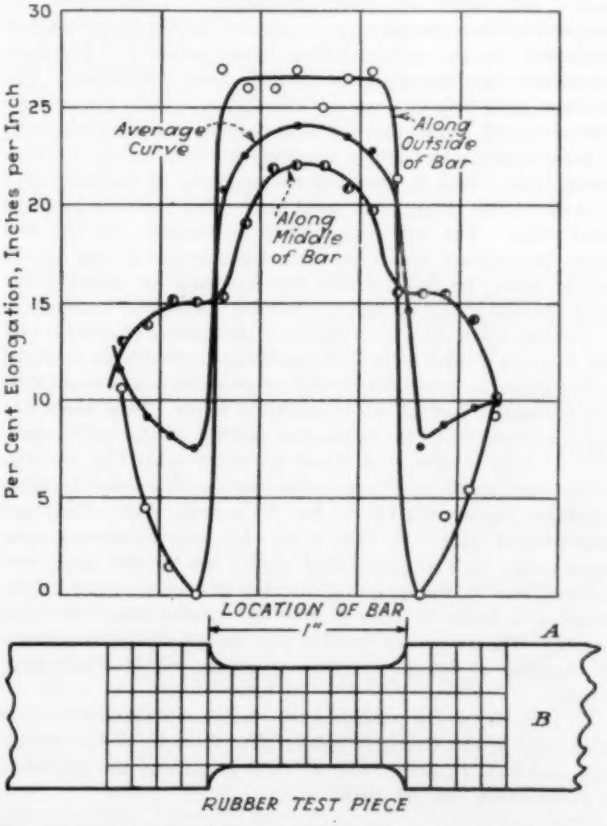


FIG. 19 CURVES FROM RUBBER MODEL SHOWING THE VARIATION OF PER CENT ELONGATION ALONG THE LENGTH OF A GROOVED BAR STRESSED IN TENSION

(Mode 1 stretched 20 per cent on 2-in. gage length.)

Discussion

GEORGE C. PRIESTER.[26] It is pleasing to note that the author has demonstrated the fact that there exists an upper and a lower yield point. Most American engineers assume that the stress-strain diagram is always a horizontal line representing the unit stress at the upper yield point. This is not the case for our ordinary low-carbon bessemer or open-hearth steel. The writer has tested many specimens of hot-rolled steel bars, round, square, and rectangular, in cross-section, and has obtained autographic stress-strain diagrams similar to type *a* in Fig. 8. The specimens were not polished or machined, but still retained the mill scale over their entire length. From this it is evident that the polishing of the specimen is not a factor in producing this type of a stress-strain diagram.

The writer has also carried out tests on bars of rectangular cross-section in which holes had been drilled perpendicular to the axis of the bar. The stress-strain diagram for these specimens were similar to type *d*, Fig. 8, of the author's paper. In test pieces in which there was placed a series of holes along the axis of the bar, the stress-strain diagrams were similar to the author's diagrams shown in Fig. 16. The shape of the diagrams varied when different-sized holes were drilled in the specimen.

The author should be commended for his valuable contribution to the study of plastic flow in mild steel. It is hoped that he may find it possible to continue his studies along this line.

J. A. VAN DEN BROEK.[27] The paper raises a number of queries. The first one is on terminology. In the title there occurs the term "yield point." In the fourth line of the first page the author introduces the term "plasticity." Furthermore, by implication, he seems to suggest substituting the term "limit of plasticity" for the well-tried term "yield point" and the word "plasticity" for the good old English word "ductility." The writer cannot help but wonder whether a certain German influence is not responsible for this change in our terminology. It may be that the German language has but one word, "plastizität," by which it describes the property of modeling clay as well as the property of mild steel when stressed past the yield point. The writer's objection is twofold. In the first place, the author thus blurs a useful distinction, and in the second place, he impoverishes our language by rejecting the term "ductility" and by making the word "plasticity" less explicit.

Further, when the author speaks of the limit of plasticity, why not mention a third limit? It would appear that what is meant in the paper by plasticity is what is graphically represented by the horizontal part of the stress-strain curve. How about the right extremity of this horizontal portion of the stress-strain curve? And further, how about the point S_1 on Fig. 1? The writer understands that there are two terms to describe the stress condition represented by S_1, Fig. 1—namely, elastic limit and proportional limit. He fails to see any gain in offering some more terms, such as upper yield stress. On the first page, first column, line 16, the author states that in "columns and beams, buckling is likely to occur if they are stressed above the yield point." The writer had the idea that the essential characteristic of buckling of columns is that it occurs within elastic-limit stresses.

In 1918 the writer published the results of some 250 tests on mild steel in the Carnegie Scholarship Memoirs of the British Iron and Steel Institute, Vol. 9. This paper was also published in *Engineering*, July 26, 1918.

[26] Professor of Materials of Engineering, University of Minnesota, Minneapolis, Minn.

[27] Professor of Engineering Mechanics, University of Michigan, Ann Arbor, Mich.

The writer was very much interested in the amorphous-cement theory as an explanation of the effects of cold-working on the elastic properties of steel. In 1918 he found Bauschinger's classic experiments of 1886 uniformly misinterpreted in American and English literature. Some of this misinterpretation persists till this day. See Rosenhain, in his "Physical Metallurgy," 1914, page 247. See also page 15 of the writer's paper referred to and also page 18, from which is quoted, "If the contentions set forth in this paper are correct, it would appear that the strongest evidence Rosenhain mentions against the validity of his favorite amorphous-cement theory is invalid." Pertinent matter may be found on page 27, "Tests on Steel Balls."

The paper mentions the fact that the specimens used were very highly polished. On the third page, second column, is discussed the effect of surface conditions, yet the author neglects to attribute any weight to the highly polished surface. It would seem that, after the thousands of stress-strain curves of mild steel recorded during the last 60 years, little need exists for recording still more unless new features are brought out and attempts are made to explain such new features. The writer regards Fig. 9, which shows an upper yield stress 66 per cent higher than the lower yield stress, as uncommon to say the least. He does not deny that they may be obtained. On the second page, second column, the author says, "The question of whether to use the upper or lower values of stress as a limit is in many countries not definitely settled." So far as the bridge-engineering and building-engineering professions in America are concerned, the writer offers without much fear of contradiction by any one in responsible charge of designing of either bridges or skyscrapers the statement that the question is quite definitely settled. It is recognized that by special test a condition may arise similar to one in a supercooled liquid, or possibly similar to the difference manifested by static as against dynamic friction. The writer does not believe that the American bridge engineer, one of the greatest users of mild steel, will accept the curve in Fig. 9 as representative of the behavior of material in his structure as a train thunders over it. The writer further feels that, in his efforts to eliminate concentration of stresses, the author has coated his specimens with a layer of exceedingly tough, amorphous steel which materially affects the behavior of the specimen. Witness the results the writer obtained with his tests on steel balls previously quoted. The tests recorded then are not so much representative of mild steel as the title of the paper leads one to suppose, but are tests of steel bars consisting of mild steel in the core, which core is effectively supported by polished steel, or amorphous steel, or intercrystalline steel, whichever term one prefers, on the surface.

R. E. PETERSON.[28] A series of tests of grooved specimens is described in which the width of the groove is varied. It is not quite clear what happens when the notch width becomes zero—in other words, a circumferential crack. From Fig. 17 it is not apparent what the percentage elongation of *A-B* becomes at zero width of groove. If total elongation of *A-B* is plotted, the origin may be used as a point on the curve.

A series of fatigue tests was made by the writer some time ago in which circumferential cracks were produced in the test specimens by a method suggested by P. G. McVetty. A very narrow and deep groove was turned in a bar, which was then heated and compressed axially to close the groove and thus form an artificial circumferential crack. A few tensile tests were also made. These tests indicate that failure is of a triple-tension type, producing a fracture of crystalline appearance. No apparent yielding occurs, and it is suggested that the stress at breaking is

[28] Research Laboratories, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Assoc-Mem. A.S.M.E.

the cohesion strength[29,30] of the material and further that all ordinary strength properties (elastic limit, upper and lower yield points, true ultimate strength) approach the cohesion strength as a limiting value as the width of groove approaches zero. These are, of course, suggestions, and it is hoped that further tests will be made along this line.

A. NÁDAI.[31] The various interesting phenomena accompanying the yield point of mild steel as observed or determined in the standard tensile testing methods and the various factors which seem to have an influence on the magnitude of the stress under which the first permanent deformations of considerable magnitude in steel are produced, deserve perhaps more attention than they have found until now. To the Verein deutscher Eisenhüttenleute in Düsseldorf is due the credit of having stimulated a number of investigations several years ago among the engineers with the aim to determine the various mechanical and thermal factors which have an influence on the magnitude of the yield stress of mild steel. The main results of these investigations are contained in a number of valuable reports mentioned in the paper. Among these, particular attention may be called to the valuable contributions contained in the investigations of P. Ludwik,[32] M. Moser,[2] and F. Körber.[3] In these recent investigations and in tests made by other experimenters not quoted here, many points have been brought out concerning the conditions that have an influence on such an important mechanical material constant as the stress under which the permanent deformations start to develop in mild steel. It has been shown that besides the carbon content, which has its primary influence on the shape of the stress-strain curve at normal temperatures, a number of other factors must be taken into account for a more exact determination of the yield stress of steel.

The influence of notches and of local stress concentrations in a tensile-test specimen on the upper yield point has been particularly studied by the author. It is a noteworthy result of his tests that he has carefully tried to correlate the occurrence of the drop of the load so characteristic for mild steel at the yield point to the appearance of the flow or strain figures.

The yield point or yield stress as a mechanical characteristic and as an expression of one of the important qualities of steel will mean more for the designer and the engineer, who must base their calculations on certain standard quantities, if they will keep in mind, together with the limiting stress values, the more important mechanical behavior of the steel chosen for a given purpose. The engineer should perhaps not be satisfied to know that his working stresses are well below one or the other of the material constants on which, according to our present knowledge, he must base the dimensions in his constructions. He should consider rather the whole behavior of the material, as read from the stress-strain diagram, and should take into account all those factors described in the paper and in the references mentioned above. These various factors have a definite influence on the magnitude of the yield stress and on the shape of the transition curve from the purely elastic state to the plastic state of steel. A more complete study of the various modes of yielding will help finally to establish a more satisfactory knowledge of the factors limiting dimensions of machine parts in design than a more or less rigid adherence to a limiting stress or a similar material constant.

Apart from the points reported in the paper, there are a number of other questions which may still be investigated in connection with the yield stress. One or two of these points may briefly be touched here.

It has been shown by P. Ludwik that age hardening has a great influence on the upper yield stress. He has shown that a comparatively small change of the temperature after a first tensile test may raise considerably the upper yield stress in a subsequent test. In this connection some further results obtained by W. Koester,[33] which point to the nitrogen content in steel as a determining factor of the magnetic age hardening, should be mentioned here. Small quantities of nitrogen (0.005 to 0.01 per cent) which can be dissolved in steel or precipitated from it under various conditions seem not only to produce these variations (for example, magnetic aging) but also to affect the developing of the dark lines by Fry's etching solution. On this point further work is in progress at the Research Laboratories of the Westinghouse Company.

As a last point, which might perhaps also need further investigation, should be mentioned the study of the conditions of stability in various testing methods. As was observed long ago by Considère and a number of other investigators, the static equilibrium in a test bar subjected to pure tensile stresses becomes unstable. Under certain conditions when the load-extension curve reaches an analytical maximum (a test bar at the upper yield point is in such a condition), an unstable state of equilibrium results. It would be interesting to discuss the mechanical conditions of the equilibrium, or rather the dynamic conditions while the equilibrium is disturbed, after the upper yield point has been reached and during the period in which the stress decreases from the upper to the lower yield stress.

From the metallurgical side some points deserve further investigation. For example, the strength of the boundary substance lying between the grains in the structure of mild steel influences the yield point. There are indications that to the excessive strength of the boundary substance enclosing the soft ferrite grains in mild steel may be attributed the cause for the sharp break in the stress-strain diagram at the yield point of mild steel.

AUTHOR'S CLOSURE

The author wishes to thank Professor Priester for further emphasizing several important points brought out in the paper. It is interesting to hear that he has made tests which corroborate and check the results reported in the paper. The tests which he made show very plainly that the polish on the specimen has nothing to do with producing the peak in the stress-strain curve. In regard to the tests mentioned on flat bars with holes drilled perpendicular to the axis, local yielding in the formation of slip layers prevented the attainment of a high peak, and in this respect should produce diagrams similar to the notched-bar tests results plotted in Fig. 16. It is interesting to note that Professor Priester's tests check this observation.

With regard to Professor Van den Broek's remarks concerning the terminology used in the paper, the author wishes to say that the term "yield point" was employed throughout in the majority of cases, and was not discarded for another term. The expression "limit of plasticity" was used in certain cases in order to differentiate between the actual physical phenomenon taking place and that arbitrarily defined quantity corresponding to an arbitrary percentage of elongation. As was mentioned in the paper, the subject treated here deals only with that yield point which expresses a natural limit or a physical property of the material, and not with that stress which is based on an arbitrary

---

[29] Ludwik, "Die Bedeutung des Gleit- und Reisswiderstandes für die Werkstoffprüfung," *Zeit. V.D.I.*, vol. 71, p. 1532 (1927).

[30] Czochralski, "Die Grundlagen der Verfestigungsvorgänge," *Zeit. fur Metallkunde*, vol. 15, p. 7 (1923).

[31] Research Laboratories, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

[32] P. Ludwik and Scheu, "Über die Streckgrenze von Elektrolyt- und Flusseisen," Werkstoffausschussbericht no. 70 des V.D.E. Düsseldorf, 1925; cf. also *Stahl und Eisen*, 1925, and other papers by Ludwik and his collaborators.

[33] "Zur Frage des Stickstoffs im technischen Eisen," *Stahl und Eisen*, vol. 50, no. 19, p. 629, May 8, 1930, and subsequent papers.

rule. The term "limit of plasticity" expresses clearly and exactly what type of yield point we are then dealing with, and excludes the possibility of confusion with such conventional definitions as are used in the testing of materials. The upper yield point is in truth the limit of plasticity of the material. The term "limit of plasticity" expresses the fact that in such



FIG. 20 TENSILE-TEST BAR OF TYPE *a* HAVING A HEAVY OXIDE SCALE OVER ITS SURFACE

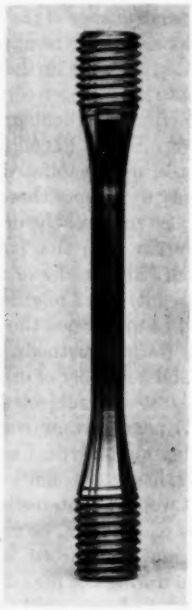


FIG. 22 ROUND TYPE OF TENSILE-TEST BAR USED IN TESTS TO DETERMINE EFFECT OF POLISHING ON UPPER YIELD POINT

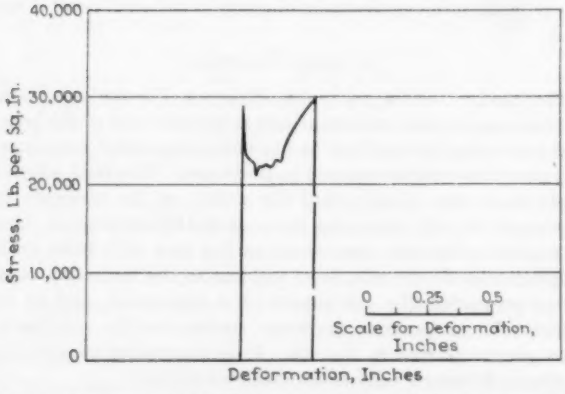


FIG. 21 PORTION OF AUTOGRAPHIC STRESS-DEFORMATION DIAGRAM FOR A TENSILE-TEST BAR OF TYPE *a* HAVING A HEAVY OXIDE COATING

metals as mild steel the permanent deformations begin to become of a large magnitude at this stress. This term is not a new one in the English language or in other languages, and the author does not claim in any respect to be responsible for originating it. At the present he has on his desk a few of the Engineering Index Service cards, and on looking through them he found the word "plasticity" mentioned several times. The term "limit of plasticity" may be found, for instance, on card number 25,458, serial No. 31, of recent publication, and also in a large number of recent publications which cannot be listed here. However, to further clear up this point the author would like to call to Professor Van den Broek's attention the statement on the first page of the paper: "It is not the purpose of this paper to analyze the conventional methods of defining the yield point or to present new definitions in this connection." It was more the intention to contribute to the physical knowledge of the various conditions which have been found to have an influence on the magnitude of the yield stress in mild steel.

Concerning the word "ductility," the author preferred not to apply this word in this connection on account of the various meanings which are generally attributed to it in the testing of materials.

Professor Van den Broek also states that "there are two terms to describe the stress condition represented by S_1, Fig. 1, namely, elastic limit and proportional limit. He (the writer) fails to see any gain in offering some more terms such as upper yield stress." Here the author wishes to remark that the terms "elastic limit" and "proportional limit" were not used for a similar reason as was stated above for the word ductility. Again, the author does not claim to introduce a new term in using the expression "upper yield point." This term appears to have been introduced by C. von Bach, as was mentioned earlier in the paper, and has been used very frequently since that time.

Concerning Professor Van den Broek's remark regarding the buckling of beams, the author wishes to point out that the statement referred to in the paper in the first column of the first page refers to the well-known theory of buckling after the yield point has been passed. The phenomenon of the buckling of columns and beams is not confined to purely elastic deformation, as several cases have been treated in the technical literature in which buckling occurs after the yield stress has been passed.[34]

In the oral discussion Professor Van den Broek brought up the fact that the stress-strain curves included in the paper seemed to be of an unusual shape. With regard to this, the author believes that this may perhaps best be answered by reference to a large amount of work already done in this same field. This previous work is described in the various papers published by M. Moser, F. Körber, W. Kuntze, G. Sachs, P. Ludwik, R. Scheu, C. von Bach and others which were referred to in the paper itself. These various publications contain many stress-strain curves for mild steel having the same shape and form as those included in this paper, which indicates that they are not of an unusual shape but are the same as those usually received for mild steel.

The author would like to also point out in this connection that the type of testing machine used plays an important part in the determination of the lower yield point. The hydraulic testing machine with a continuous equilibrium of the force-measuring device provides a much quicker determination of this lower yield point than does the beam-type machine. If the beam-type machine has been used by Professor Van den Broek, this may perhaps account to a certain extent for his difficulty in observing this drop in load at the yield point.

The question of the effect of the polish on the test bar on the behavior of the specimen is next discussed by Professor Van den Broek. It is believed by him that in polishing the test bars so highly, the bars were thus coated with a layer of tough, amorphous steel which seriously affected the behavior of the specimen. In other words, he believes that the steel bars were mild at the core, but that the core was supported by the amorphous steel which changed the behavior of the material under stress. Professor Priester mentioned several tests which he had made on hot rolled bars having a heavy mill scale on them. He obtained

[34] For a discussion of some of these cases see A. Nádai, "Der Bildsame Zustand der Werkstoffe," J. Springer, Berlin, 1927, p. 128, and also H. M. Westergaard and W. R. Osgood, "Strength of Steel Columns," Trans. of A.S.M.E., vol. 49–50, p. 65.

stress-strain curves similar to that shown for the highly polished bar, type *a* in Fig. 8. This shows clearly, then, that the polish had no appreciable effect on the behavior of the material.

In order to further check this since the paper was presented, several different tests have been made, as this question is rather important.

First a round bar of mild steel of type *a* was machined, and then annealed in the customary manner at 900 deg. cent. The annealing produced a heavy oxide scale over the test piece as shown in Fig. 20. This scale was left on the bar, and it was tested in this condition. The autographic diagram is shown in Fig. 21. This shows that even with an unpolished bar that has only a heavy oxide scale over the surface, the same type of stress-deformation curve is obtained as with the highly polished bar in Fig. 8 and having a high peak of similar magnitude. Second, five round tensile-test bars were made similar to the one shown in Fig. 22. Three of these were highly polished, one was well machined but not polished, and one was only roughly machined and not polished. Two of the three well-polished test pieces were placed in concentrated hydrochloric acid for 1 hour and $1^1/_2$ hours, respectively. In both cases the polish was completely removed by the acid. When these bars were tested, stress-deformation diagrams were obtained as shown in Fig. 23. The results of the tests are tabulated in Table 3. It was found that there was no difference in the values of upper yield point obtained for the highly polished bar and the unpolished but well-machined bar. The rough-machined bar showed a small decrease in the upper yield point due to the presence of a large number of stress concentrations produced by the lathe tool. This effect in reducing the upper yield point is somewhat similar to that produced in the grooved flat bars, type *d*, mentioned in the paper, but of smaller magnitude. Slip layers form at the grooves in both cases, and local yielding reduces the upper yield point. The effect of the etching on the polished bars was of a similar nature. Minute stress concentrations here also tended to slightly reduce the upper yield point. In all of the cases, however, Fig. 23 shows essentially the same type of high peak with sudden drop in load, indicating that the polish had no effect whatsoever in producing the peak. An additional test was made to further emphasize this. A cylinder of hard-drawn copper was annealed at 500 deg. cent. for two hours and then tested in compression. The copper was quite soft after this annealing treatment. The deformation was measured by means of two dial gages. When a load of 20,000 lb. was reached, the cylinder was unloaded, and the deformation was also measured during unloading. The cylinder was then polished highly on the lathe and reloaded. The stress-strain curve obtained is shown in Fig. 24. According to what was said, if polishing had any effect on increasing the load, there would be a sudden discontinuity in the curve at the point *A* in the figure. None was obtained on reloading in this case, however, since the curve continued along the same path it was traveling before being interrupted by unloading.

From these various tests it seems apparent, therefore, that polishing had no effect in producing the peak in the stress-strain curves as thought by Professor Van den Broek.

Further, in regard to Professor Van den Broek's compression tests to determine the crushing strength of steel balls, it might be said that the results of these tests could hardly be applied here to the yield point, since the crushing strength in such cases is governed by a very complicated three-dimensional stress distribution, and depends besides more on the ultimate or breaking strength than on the yield point determined in simple tension.

It was mentioned in the oral discussion by Professor Timoshenko that through certain precautions the yield point could be raised even to the ultimate strength of the material. This no doubt refers to Ludwik's age-hardening tests. The author would also like to add in this connection that such precautions as the careful design of the test piece in order to prevent

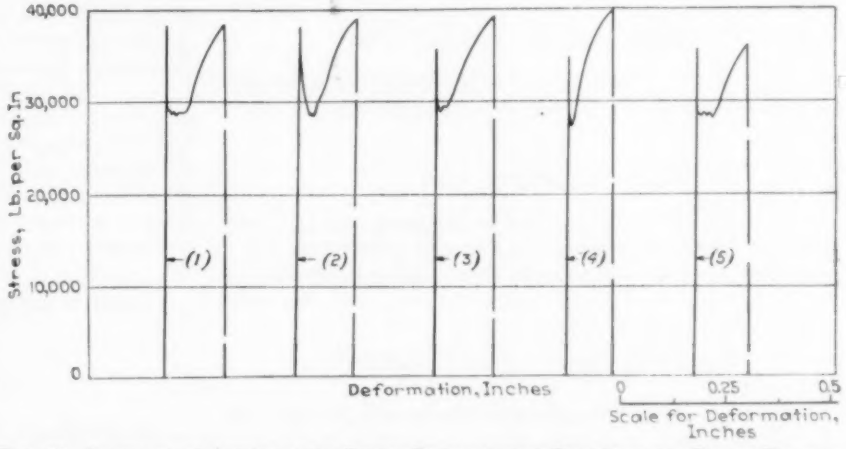


FIG. 23 PORTIONS OF AUTOGRAPHIC STRESS-DEFORMATION DIAGRAMS FOR ROUND TENSILE-TEST BARS HAVING VARIOUS SURFACE CONDITIONS

[(1) Highly polished bar. (2) Well-machined bar, not polished. (3) Rough-machined bar, not polished. (4) Same as (1) but etched in HCl for 1 hr. (5) Same as (1) but etched in HCl for $1^1/_2$ hr.]

TABLE 3 RESULTS OF TESTS ON FIVE DIFFERENT TENSILE TEST BARS WITH A VARIETY OF SURFACE CONDITIONS

(All bars included here were round of the type shown in Fig. 22.)

Type of test bar	Upper yield stress, lb. per sq. in.	Lower yield stress, lb. per sq. in.	Diff. between upper and lower yield points, lb. per sq. in.
Highly polished bar	37,900	28,400	9,500
Well-machined bar (no polish)	37,900	28,200	9,700
Roughly machined bar (no polish)	35,700	28,900	6,800
Well-polished bar etched for 1 hr. in conc. HCl	35,400	28,600	6,800
Well-polished bar etched for $1^1/_2$ hr. in conc. HCl	36,000	28,600	7,400

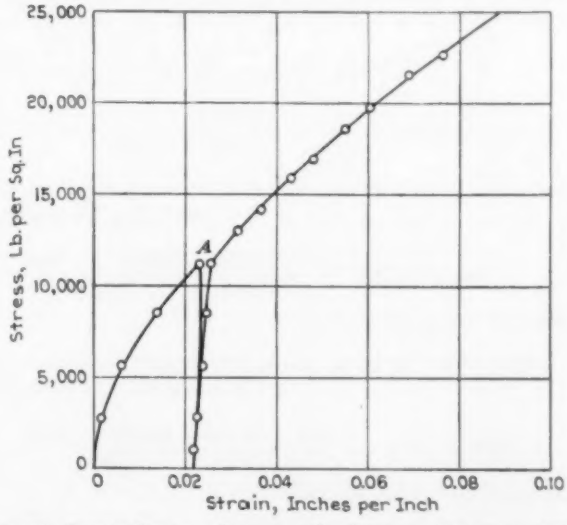


FIG. 24 STRESS-STRAIN CURVE FOR COPPER CYLINDER WITH REPEATED LOADING

stress concentrations is very important to attain this high peak. If a test bar has sharp fillets between the head and the cylindrical length, both M. Moser and F. Körber have shown that the peak will be greatly reduced, as was mentioned earlier in the paper.

A very important additional point was brought out in the discussion by Dr. Nádai, in that the designing engineer should keep in mind not only the value of the yield stress, but also the various factors affecting it. In doing this the engineer will arrive nearer to the true condition present in his structure. It is also to be noted that Dr. Nádai has indicated a number of other interesting questions in this connection which may still be investigated, and it is hoped that more research will thus be stimulated in this field.

It was pointed out by Mr. Peterson that it is not apparent from Fig. 17 what the percentage of elongation of $A - B$ becomes at zero width of groove. This fact is indeed true. The author, however, was more interested in the percentage of elongation as used in the ordinary stress-strain curve than in total elongation in an effort to compare the actual lengths of the portion $A - B$, as would be shown in the diagram. The total elongation, if plotted, would pass through the origin, as was suggested by Mr. Peterson.

APM-53-16

Stresses in Curved Bars

By H. C. PERKINS,[1] ITHACA, N. Y.

Because of the difficulty of computing accurately the stresses in curved bars by the ordinary method, the author has derived simple formulas which give sufficiently reliable results with the use of a slide rule. The paper sets forth these derivations and gives several examples of the application of the formulas.



CURRENT methods of computing stresses in curved bars are complicated and, unless carried out with extreme care, likely to give inaccurate results. It is the object of this paper to describe simpler methods and to propose formulas which will give reliable results with the use of a slide rule. Statement of the formulas will be followed by illustrative applications, after which the methods of derivation will be described.

Fig. 1 shows part of a curved bar above the cross-section GH in equilibrium under the action of a load P and the stress on the cross-section GH. The plane of the cross-section GH passes through the axis of curvature O of the centroidal axis of the bar. The load P is applied in a plane of symmetry of the bar and perpendicular to the cross-section GH. In this figure and in Figs. 2 and 3, let

R = radius of curvature of the centroidal axis of the bar
A = area of the cross-section GH
dA = area of an elementary strip
S = unit normal stress on the element dA
r = radius from O to the element dA
v = distance from centroidal axis of A to element dA
x = distance from neutral axis ($S = 0$) to element dA
(both v and x are positive when measured toward O)
x_1 = distance to centroidal axis from neutral axis
M = moment of P about the centroidal axis of A
M_0 = moment of P about the axis of curvature O
h = radial depth of the cross-section, and
n = the ratio R/h of centroidal radius to depth of section.

The proposed formula for unit stress at any point in the cross-section is

$$S = \frac{P - \frac{M}{R}}{A} + \frac{M}{RF}\frac{v}{r} \qquad [1]$$

in which

$$F = \int \frac{v}{r}\,dA$$

The limits of integration are such as to include the entire area A. Since F appears in the denominator of the expression for bending stress and its units are those of area, F may be called the effective area of the cross-section for bending about the centroidal axis, or simply "the effective area."

Convenient Approximate Formulas

Integration yields such awkward forms for F that the following formulas are to be preferred both for convenience and probable accuracy of result, provided that the least value of r for the given section is greater than one-tenth of the depth of section h. This limit is seldom likely to be exceeded.

For the effective area of a rectangle use

$$F = \frac{An^2}{12(n^2 + 0.04)(n^2 - 0.188)} \qquad [2]$$

The value of n should not be less than 0.6.

For the effective area of a circular or elliptical section use

$$F = \frac{An^2}{16(n^2 + 0.035)(n^2 - 0.161)} \qquad [3]$$

The value of n should be not less than 0.6.

For the effective area of a triangular section, base parallel to the axis of curvature, vertex directed toward the axis of curvature, use

$$F = \frac{A}{18(n^2 - 0.145)} + \frac{A}{135\,n(n^2 - 0.342)} \qquad [4]$$

The value of n should not be less than 0.77.

Examples

Example 1. To illustrate the use of these formulas, consider the curved bar in Fig. 4. Compute the maximum unit normal stress on the cross-section GH, which is circular. $P = 5000$ lb., $A = 12.57$ sq. in., $R = 4$ in., $h = 4$ in., $n = 1$, $M = 40{,}000$ lb-in., and $M/R = 10{,}000$ lb.

To evaluate F, use Equation [2], or

$$F = \frac{12.57(1)}{16(1 + 0.035)(1 - 0.161)} = 0.904 \text{ sq. in.}$$

The maximum tensile stress is at H. Substituting this value in Equation [1] and taking $v = 2$, $r = 2$

$$S = \frac{5000 - 10{,}000}{12.57} + \frac{10{,}000\,(2)}{0.904\,(2)} = -398 + 11{,}050$$
$$= 10{,}652 \text{ lb. per sq. in. tension}$$

The maximum compressive stress is at G. Substituting in [1], with $v = -2$ and $r = 6$,

$$S = \frac{5000 - 10{,}000}{12.57} + \frac{10{,}000\,(-2)}{0.904\,(6)} = -398 - 3685$$
$$= -4083 \text{ lb. per sq. in. compression}$$

In case the section is a composite of several standard parts, such as a T-section composed of two rectangular parts, it is necessary to obtain the effective area of each part for bending about the centroidal axis of the complete section. The effective

---

[1] Assistant Professor of Mechanics of Engineering, Sibley School, Cornell University. Mr. Perkins received the M.E. degree from Cornell University in June, 1915. After a year as machinist in the toolroom of the Morse Chain Works at Ithaca, N. Y., he was appointed instructor in mechanics in Sibley College. He was for one year instructor in the Department of Heat Power in Sibley College and one year as instructor in the Department of Mechanics of the University of Wisconsin.

Contributed by the Applied Mechanics Division and presented at the Hartford, Conn., Meeting, June 1–3, 1931, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

area of the composite section is then equal to the sum of the effective areas of the parts when referred to this axis.

$$F = F' + F'' + \ldots \quad [5]$$

To find the value of the effective area F' for bending about an axis at radius R', from the area A_1 whose centroid is at radius R_1 and whose effective area is equal to F_1, use the formula

$$F' = \frac{F_1}{R_1} R' + \frac{A_1}{R_1}\left(R' - R_1\right) \quad [6]$$

Example 2. To illustrate the use of formulas [5] and [6], find the maximum unit normal stress on the T cross-section

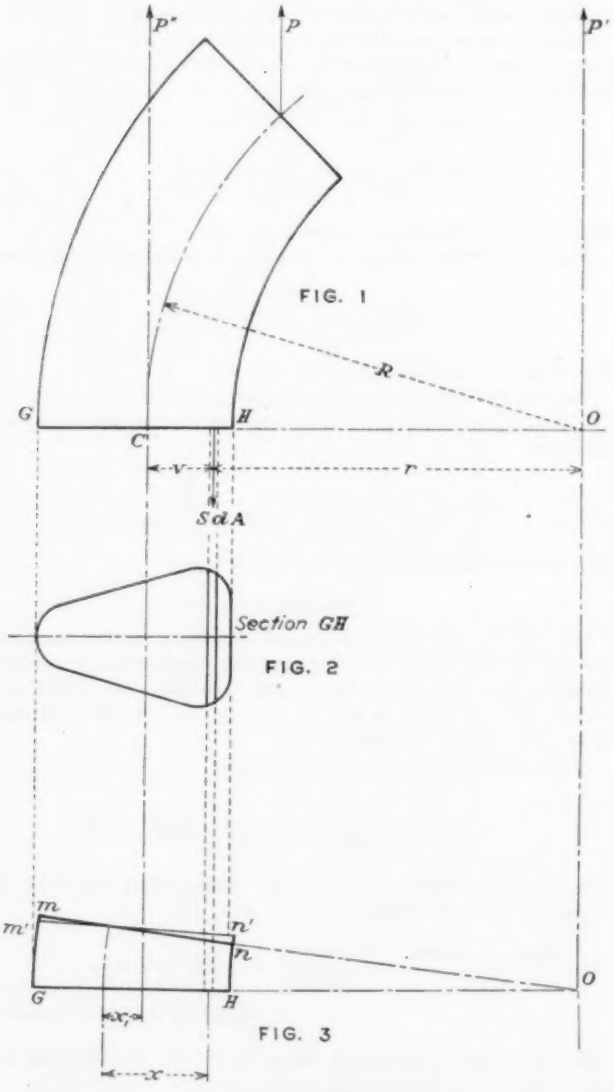


FIG. 1

FIG. 2

FIG. 3

FIGS. 1–3

of Fig. 5 when $P = 4000$ lb. For the rectangle at the left, $A_1 = 6$ sq. in., $R_1 = 6$ in., $h_1 = 6$ in., and $n_1 = 1$. From [2]

$$F_1 = \frac{6\ (1)}{12(1 + 0.04)(1 - 0.188)} = 0.592 \text{ sq. in.}$$

Transferring to the combined axis, $R' = 4.6$ in., and using Equation [6],

$$F' = \frac{0.592}{6}\left(4.6\right) + \frac{6}{6}\left(4.6 - 6\right) = -0.946 \text{ sq. in.}$$

For the rectangle on the right, $A_2 = 4$ sq. in., $R_2 = 2.5$ in., $h_2 = 1$ in., and $n_2 = 2.5$, whence

$$F_2 = \frac{4\ (6.25)}{12(6.25 + 0.04)(6.25 - 0.188)} = 0.0547 \text{ sq. in.}$$

Transferring to the combined axis, $R'' = 4.6$ in.,

$$F'' = \frac{0.0547}{2.5}\left(4.6\right) + \frac{4}{2.5}\left(4.6 - 2.5\right) = 3.467 \text{ sq. in.}$$

The effective area of the T-section is therefore

$$F = F'' + F' = 3.467 - 0.946 = 2.52 \text{ sq. in.}$$

and the maximum tensile stress, for which $v = 2.6$ in., and $r = 2$ in., is, from Equation [1],

$$S = \frac{4000 - 7480}{10} + \frac{7480\ (2.6)}{2.52\ (2)} = -348 + 3860$$
$$= 3512 \text{ lb. per sq. in.}$$

For the maximum compressive stress, $v = -4.4$ in., and $r = 9$ in., so

$$S = \frac{4000 - 7480}{10} + \frac{7480\ (-4.4)}{2.52\ (9)}$$
$$= -348 - 1452 = -1800 \text{ lb. per sq. in.}$$

Example 3. As a further illustration, consider the hook described in article 133 of Morley's "Strength of Materials." The section is trapezoidal as shown in Fig. 6. $P = 1.25$ tons applied through the axis of curvature O, $A = 3.375$ sq. in., and $R = 3$ in. Regard the trapezoid as composed of two parts, a positive rectangle 2 in. $\times$ 2.25 in., and two negative triangles (treated as one part) each of 0.5 in. base $\times$ 2.25 in. altitude. For the rectangle, $A_1 = 4.5$ sq. in., $R_1 = 3.125$ in., $n_1 = 3.125/2.25 = 1.389$, and $n_1^2 = 1.929$. Substituting in Equation [2],

$$F_1 = \frac{4.5\ (1.929)}{12(1.929 + 0.04)(1.929 - 0.188)} = 0.2109 \text{ sq. in.}$$

Transferring to the axis of the trapezoid, using Equation [6],

$$F' = 0.2109\ \frac{3}{3.125} + \frac{4.5}{3.125}\left(3 - 3.125\right) = 0.2027 - 0.18$$
$$= 0.0227 \text{ sq. in.}$$

For the triangles together, $A_2 = -1.125$ sq. in., $R_2 = 3.5$ in., $n_2 = 3.5/2.25 = 1.556$, and $n_2^2 = 2.42$. Substituting now in [4]

$$F_2 = \frac{-1.125}{18(2.42 - 0.145)} + \frac{-1.125}{135(1.556)(2.42 - 0.342)}$$
$$= -0.0275 - 0.00258 = -0.0301 \text{ sq. in.}$$

Transferring to the axis of the trapezoid, using Equation [6],

$$F'' = (-0.0301)\ \frac{3}{3.5} + \left(-1.125\right)\frac{(3 - 3.5)}{3.5} = -0.0258$$
$$+ 0.1607 = 0.1349 \text{ sq. in.}$$

The effective area of the trapezoid is then

$$F = 0.0227 + 0.1349 = 0.1576 \text{ sq. in.}$$

The maximum tensile stress occurs where $v = 1$ and $r = 2$. Substituting in Equation [1] $P = 1.25$ tons, $M = 1.25 \times 3$ ton-in., and $M/R = P = 1.25$ tons,

$$S = \frac{1.25 - 1.25}{3.375} + \frac{1.25\ (1)}{0.1576\ (2)} = 0 + 3.97 \text{ tons per sq. in.}$$

The maximum compressive stress occurs where $v = -1.25$ and $r = 4.25$ in., or

$$S = \frac{1.25 - 1.25}{3.375} + \frac{1.25(-1.25)}{0.1576(4.25)} = -2.33 \text{ tons per sq. in.}$$

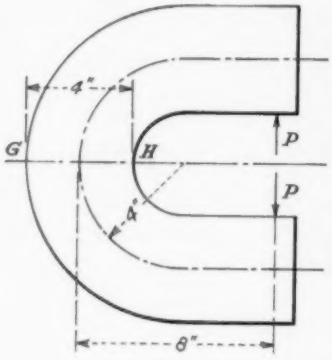


FIG. 4

These stresses differ from Morley's by about three per cent. It is an unexpected demonstration of the comparative reliability of the new formulas to discover that the error is in Morley's numerical work. Computing the value of A' from his figures, $A' = 3.5328$, so that his value for the effective area should be $A' - A = 0.1578$ sq. in., which makes the unit stresses, respectively, 3.96 and 2.33 tons per sq. in. The magnitude of the error is of course too small to affect the practical value of the result, but it does demonstrate that the older formulas are very sensitive to small errors of computation and likely to yield inaccurate results.

DERIVATION OF EQUATION [1]

Fig. 3 represents a short portion of the bar included between two successive cross-sections GH and (before loading) mn. After loading, the section mn has rotated about its neutral axis into a new position $m'n'$, thus extending each longitudinal fiber of the bar in proportion to its distance x from the neutral axis. The original length of each fiber is proportional to its distance r from the axis of curvature O, so the unit deformation of each fiber is proportional to the ratio x/r. Within the elastic limit, unit stress is proportional to unit deformation, so, in terms of a constant of proportionality K,

$$S = K\frac{x}{r} \qquad [7]$$

For equilibrium, the moments of all forces in Fig. 1 taken about the axis O must be equal to zero, or

$$M_0 = \int rS\,dA = K\int x\,dA = Kx_1A$$

One conclusion to be drawn from this equation is that the action line of a load P', which bends the bar in such a way that the neutral axis passes through the centroid of A, must pass through O for $x_1' = 0$, consequently $M_0' = 0$. Since $x' = v$ for this case, Equation [7] may be written

$$S' = K'\frac{v}{r} \qquad [8]$$

Summing vertical components of the forces applied to the bar

$$P' = K'\int \frac{v}{r}\,dA = K'F$$

in which the value of the integral F depends only upon the form of a given curved bar. Solving for K' and substituting in [8],

$$S' = \frac{P'}{F}\frac{v}{r} \qquad [9]$$

To make use of this result, resolve P in Fig. 1 into two parallel components, P' acting through the center of curvature, and P'' acting through the centroid of the cross-section. Writing moments about C, $P'R = M$, so

$$P' = \frac{M}{R}$$

and, from a force summation,

$$P'' = P - \frac{M}{R}$$

Since P'' acts through the centroid of A, it produces a uniform stress at all points of the cross-section, or

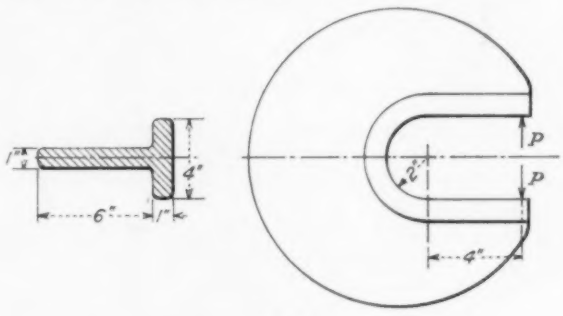


FIG. 5

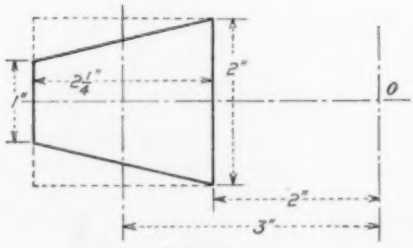


FIG. 6

$$S'' = \frac{P - \dfrac{M}{R}}{A}$$

Substituting the value of $P' = \dfrac{M}{R}$ in [9],

$$S' = \frac{Mv}{RFr} \qquad [10]$$

The stress at any point in the cross-section may now be written in the desired form.

$$S = S'' + S' = \frac{P - \dfrac{M}{R}}{A} + \frac{Mv}{RFr} \qquad [1]$$

DERIVATION OF EQUATIONS [2], [3], AND [4]

For a rectangular cross-section, area $= A$, depth $= h$, and radius from center of curvature to centroid $= R$, a value of F may be found by integration:

$$F = \frac{A}{Rh}\log\left(\frac{R + \frac{h}{2}}{R - \frac{h}{2}}\right) - A \qquad [11]$$

This expression requires careful calculation with the aid of tables and is quite unsuited to use with a slide rule. The more convenient form [2] may be obtained as follows. Expand [11] in the form of a series, or

$$F = \frac{Ah^2}{12R^2} + \frac{Ah^4}{80R^4} + \frac{Ah^6}{448R^6} + \ldots\ldots\ldots [11a]$$

Place the series in the denominator by division, giving

$$F = \frac{A}{12\dfrac{R^2}{h^2} - 1.8 - 0.0516\dfrac{h^2}{R^2} + \ldots\ldots\ldots} \quad [11b]$$

Neglect further terms, adjust the constants empirically for best agreement with [11], factor for convenience in computation, and put $R/h = n$, giving

$$F = \frac{An^2}{12(n^2 + 0.04)(n^2 - 0.188)} \quad [2]$$

When $R/h = n$ is greater than 1, this expression gives results which differ from results of [11] by less than 0.2 per cent. The greatest errors, when n is less than 1 but greater than 0.6, are less than 0.6 per cent.

Equation [3] for an elliptical or circular section may be derived as follows, integration shows that

$$F = 8A\frac{R}{h}\left(\frac{R}{h} - \sqrt{\frac{R^2}{h^2} - 0.25}\right) - A \quad [12]$$

Expansion, division, adjustment, and factoring as for the rectangle result in the form

$$F = \frac{An^2}{16(n^2 + 0.035)(n^2 - 0.161)} \quad [3]$$

When $n = 0.6$ this expression differs from [12] by about 0.7 per cent in value, the error decreasing to 0.2 per cent and less when n is greater than 1.

Equation [4] for a triangular section with vertex toward O may be obtained as follows. By integration,

$$F = 2A\frac{R}{h}\left(1 - \frac{3R - 2h}{3h}\log\frac{3R + h}{3R - 2h}\right) - A \quad [13]$$

Expanding in series,

$$F = \frac{An^2}{18} + \frac{An^3}{135} + \frac{An^4}{135} + \frac{4An^5}{1701} + \ldots\ldots [13a]$$

Summing the even-powered terms and odd-powered terms separately and dividing,

$$F = \frac{A}{18n^2 - 2.4 + \ldots\ldots} + \frac{A}{135n^3 - \dfrac{300}{7}n + \ldots\ldots} \quad [13b]$$

Neglecting further terms and adjusting the constants,

$$F = \frac{A}{18(n^2 - 0.145)} + \frac{A}{135\,n(n^2 - 0.342)} \quad [4]$$

Values obtained from this expression will differ less than 0.5 per cent from the corresponding value of [13], if n is less than 0.77.

Formulas for other forms of cross-section could be derived but would be less useful than Equations [2], [3], and [4]. In particular, an expression for the effective area of a triangle with the vertex directed away from O would be similar to [4] with the negative sign between terms but inferior to [4] in range and accuracy.

Derivation of Equations [5] and [6]

For a composite section, the integral F may be extended over the parts separately and the results added, for integration is a form of addition.

$$F = F' + F'' + \ldots\ldots\ldots\ldots [5]$$

Given the value of the effective area $F_1 = \int \frac{v}{r}dA$ for bending about a centroidal axis at radius $r = R_1$, to find the value of the effective area $F' = \int \frac{x}{r}dA$ for a parallel axis at radius $r = R'$. Put for x its value $R' - r$, then

$$F' = \int \frac{x}{r}dA = \int \frac{R' - r}{r}dA = R'\int \frac{dA}{r} - A$$

$$\int \frac{dA}{r} = \frac{F' + A}{R'}, \text{ and for the special case where } x = v$$

$$= \frac{F_1 + A}{R_1}$$

Solving for F', there results the desired expression,

$$F' = \frac{F_1}{R_1}R' + \frac{A}{R_1}(R' - R_1) \quad [6]$$

The effective area may be evaluated graphically or by approximate integration, using Simpson's Rule. Morley expresses the effective area as the difference of two areas $A' - A$, and in article 131 of his "Strength of Materials" describes a graphical construction. In using Simpson's Rule it will be seen that v changes sign at the centroidal axis, so some terms in the rule will be negative. This may be avoided if desired by using the expression,

$$F = \frac{1}{R}\int \frac{v^2}{r}dA \quad [14]$$

which may be derived as follows: $\int vdA = v_1A = 0$, so that

$$F = \int \frac{v}{r}dA - \int \frac{v}{R}dA = \int\left(\frac{v}{R - v} - \frac{v}{R}\right)dA = \frac{1}{R}\int \frac{v^2}{r}dA$$

Discussion

E. O. Waters.[2] The author's formula [1] is very interesting, by way of showing how a given mathematical process may be simplified for purposes of computation. Theoretically, the standard formulas, such as those on p. 483, Marks' Handbook, 3d ed., which are derived from Morley, should give identical results. Actually, however, it is necessary to compute three differences in these formulas, as against one difference in the author's formula. The difficulty of obtaining accurate results from formulas involving the differences of two or more terms is well known, when the slide rule is used for computation. Even with five-place tables and computing machines, the percentage error of the result is appreciable, if the differences are small in comparison with the magnitude of the separate terms.

Apparently, this desirable transformation has been secured by replacing the integral $R\int \frac{dA}{r}$ (Morley's "modified" area) with the integral $\int \frac{v}{r}dA$ (the author's "effective" area), the two being related by the equation:

[2] Associate Professor of Mechanical Engineering, Yale University, New Haven, Conn. Assoc-Mem. A.S.M.E.

$$R \int \frac{dA}{r} - A = \int \frac{v}{r} dA$$

Another form in which the equation for stress in a curved bar is sometimes written is (using the author's notation)

$$S = \frac{P}{A} + \frac{Mx}{Ax_1(R + x_1 - x)}$$

This looks simple, but before it can be used it is necessary to calculate not only the location of the centroidal axis, but also that of the neutral axis in pure bending.

In contrast with the foregoing, it would seem that the author's method for evaluating the "effective" area of irregular shapes, while theoretically sound, would introduce as great a chance of error as is obviated by using formula [1]. It also seems pertinent to ask on what basis the constants in Equations [2], [3], and [4] have been empirically adjusted. In making such adjustments, it is likely to be the case that high accuracy over a certain range of the variable in question is obtained only at the expense of considerable inaccuracy beyond this range.

C. H. KENT.[3] No exact solution of the problem of stresses in curved bars applicable to all sections has so far been discovered. For the special case of a thin rectangular section, an exact solution based on the theory of elasticity was obtained by Golovin in 1880. The approximate solutions in general use which neglect radial stresses are based on the work of Winkler published in 1858. These solutions also make the assumption that a cross-section of the curved bar remains plane as the bar is bent, which the exact solution for the section referred to shows to be actually the case.

The author has presented an approximate solution in a form which is at once easy to demonstrate and reliable to use. He is no doubt familiar with the following equation presented by the writer[4] in a different form from his own, but similarly derived:

$$S = \frac{P}{A}\left(1 - \frac{R}{r}\right)\frac{A - eX}{A - r_g X} \quad \ldots\ldots\ldots\ldots [15]$$

where

$R = A\,\dfrac{r_g - e}{A - eX}$ = radius of curvature of neutral axis

r_g = radius of curvature of gravity axis

e = distance from center of curvature of bar to line of action load

$$X = \int \frac{dA}{r}$$

and other terms have the same meaning as in this paper.

In this form the equation has the advantage that the value of $\int \dfrac{dA}{r}$ is readily obtained for regular sections, making unnecessary the use of the approximation for F as in this paper. It has, however, the disadvantage of additional complexity and sensitivity to numerical errors. The equation for R shows at once that the neutral axis coincides with the centroidal axis when e is zero, as the author has pointed out.

When applied to Example 1 of this paper, Equation [15] gives stresses of 10,350 and —4010 lb. per sq. in. at the points H and G, respectively. In applying this formula it should be remembered that e in Example 1 is negative. Comparison with the author's results indicates the accuracy of the expressions he has given for F.

One may compare the results of these formulas with the exact solution for the case of a thin rectangular section. Let $n = 1.5$, and assume that the line of action of the load passes through the center of curvature. The exact solution in this case gives $S = 12.88P/A$. Equation [15] gives $S = 12.63P/A$. The author's Equation [1], using his approximation for F and $V/F = {}^1/_2$, gives $S = 12.60P/A$.

We may conclude that formula [1] and the approximations for F which have been given are sufficiently accurate for practical purposes and possess the distinct advantage of yielding reliable results with the use of the slide rule.

AUTHOR'S CLOSURE

After the interesting and pertinent comments of Mr. Waters and Mr. Kent, further discussion would seem to be unnecessary, but for the question of the method of adjusting the constants in Equations [2], [3], and [4]. As to the effect of this adjustment, it should perhaps have been stated that the adjustment is small and does not appreciably affect the results in most practical cases. In Equation [11*b*], for instance, the third term in the denominator is usually small enough to neglect, but in the unusual case where n is less than 1, this term becomes important. It is therefore possible to select a suitable value for this term, which with an extremely small change in the second term will permit the use of the final formula for all values of n greater than 0.6, and without introducing appreciable error into the range where n is greater than 1. It is true that the result is very much in error for values of n less than 0.6, but no engineer is likely to meet such an extreme case (inside radius of curvature less than one-tenth of the depth of section).

Mr. Waters also suggests that computations for the effective area of an irregular section are subject to chances of error when computed by the method used in Examples 2 and 3. The main advantage of that method is its similarity to that used in computing the moment of inertia of a composite section. An engineer who can compute the moment of inertia of a composite section should be able to obtain equally reliable results when he uses this method of computing effective area. It is also well to keep in mind the advantages of Simpson's Rule, especially when used in connection with Equation [14].

---

[3] Professor of Mechanical Engineering, Pennsylvania State College, State College, Pa. Present address, Ann Arbor, Mich. Mem. A.S.M.E.

[4] *Machine Design*, March, 1931, pp. 30–32.

APM-53-17a

Strength of Thin Cylindrical Shells Under External Pressure

By HAROLD E. SAUNDERS[1] and DWIGHT F. WINDENBURG,[2] WASHINGTON, D. C.

The scope of existing formulas for the design of cylindrical tubes and vessels under external pressure, especially with regard to the determination of empirical constants and coefficients by systematic and comprehensive experimental data, is rather limited. The introduction and expansion of industrial processes, involving the treatment of materials under a vacuum, or within double-walled chambers whose jackets are filled with steam under high pressure, has, in conjunction with reduction in operating costs by the use of large units, resulted in the building of cylindrical vessels of great size, in the design of which engineers have had to be guided largely by what little past experience they could muster. The dimensions and proportions of these units fall far beyond the scope of any previous experiments. With fabrication costs so high and with satisfactory performance in service so important an item, the use of inadequate formulas can hardly be justified. Engineers called upon to execute these designs very rightly feel that they must have at their command workable scientific formulas, based upon logical application of accepted theories and capable of establishing proportions which will result in proper distribution and economy of material. It was because of the lack of adequate design formulas of this kind that the A.S.M.E. Special Research Committee on the Strength of Vessels Under External Pressure was organized. This paper represents the first attempt of the committee to meet the needs of designer and engineer. It is being published in advance of completion of research work on this project, to fill urgent needs in the profession and to bring forth suggestions, comment, and criticism from those interested in this work.

THE lack of scientifically proved and generally accepted formulas for the design of cylindrical vessels that must withstand external pressure is the result not so much of lack of ingenuity or enterprise in the theoretical treatment of the design problem, but of the rather limited field of application and the dearth of comprehensive experimental data which would permit the establishment of satisfactory constants and coefficients in existing formulas.



H. E. Saunders

To be sure, certain portions of the field have been carefully explored and are well covered. For example, the scantlings of tubes and cylindrical furnaces for fire-tube boilers are being selected by reference to simple formulas which have been proved by many years of practical experience. In these cases, however, the design is so involved with allowances for fabrication, for corrosion, for unequal heating, and for varying degrees of auxiliary support, that the fundamental strength formulas can scarcely be recognized.



D. F. Windenburg

The classic experiments of Fairbairn, Stewart, Carman, Cook, and others provide simple and useful design formulas for pipes subjected to external pressure, but for the most part they apply only to commercially manufactured tubular members of moderate diameter and long length, which are in many cases assumed free from end or axial loading and from other restraints. Specifically, the diameters covered by these tests are 12 in. or less, and the tubes generally have a length-diameter ratio of 4 or more. The thickness-diameter ratios, however, are in every case so small that the pipes may be expected to collapse by instability.

The present-day problem involves diameters of the order of 20 to 50 ft., length-diameter ratios of from 4 down to 1 or less, and working conditions which in almost every case involve end loading. The problem then becomes one of the strength of thin shells of pressure vessels rather than the resistance of long pipes and tubes to external pressure. The simple conventional formulas are found no longer applicable when the length-diameter ratio falls below about 6, since the collapsing pressure rises very rapidly with a decrease in this ratio. For the majority of modern designs, the thickness-diameter ratio may be assumed to lie below 0.020, and the ultimate collapsing pressure, for a tube of infinite length, below 500 lb. per sq. in. These limits will be adhered to in the present paper.

General Theory

The two-dimensional collapse of an infinitely long, thin tube has been carefully investigated. The formula

$$p_e = 2\,\frac{m^2 E}{m^2 - 1}\left(\frac{t}{d}\right)^3 \qquad [1]$$

---

[1] Senior Assistant to Officer in Charge, U. S. Experimental Model Basin, Navy Yard, Washington, D. C. Commander Saunders is a graduate of the United States Naval Academy and of the Massachusetts Institute of Technology, Department of Naval Architecture. For three years he was head of the submarine-design section in the Bureau of Construction and Repair, Navy Department, and for five years in charge of the construction of large V-class submarines at the Navy Yard, Portsmouth, N. H. Since October, 1929, he has been in his present position.

[2] Assistant Physicist, U. S. Experimental Model Basin, Navy Yard, Washington, D. C. Mr. Windenburg was graduated from Cornell College, Iowa, with the M.A. degree, followed by two years of graduate work in physics and mathematics at the University of California. For three years he was head of the mathematics department of the Polytechnic College of Engineering, Oakland, Calif., and for two years head of the mathematics department of the Long Beach Junior College, Long Beach, Calif. Since January, 1929, he has been in his present position.

Contributed by the A.S.M.E. Special Research Committee on the Strength of Vessels Under External Pressure and presented at a joint session with the Applied Mechanics Division during the Hartford, Conn., Meeting, June 1–3, 1931, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

given by Bryan (1)[1] is generally accepted. Here,

p_c = collapsing pressure
E = Young's modulus
$1/m$ = Poisson's ratio
t = the thickness of the tube
d = the diameter of the tube to the neutral axis (middle of the shell or wall).

All units are given in inches and pounds per square inch.

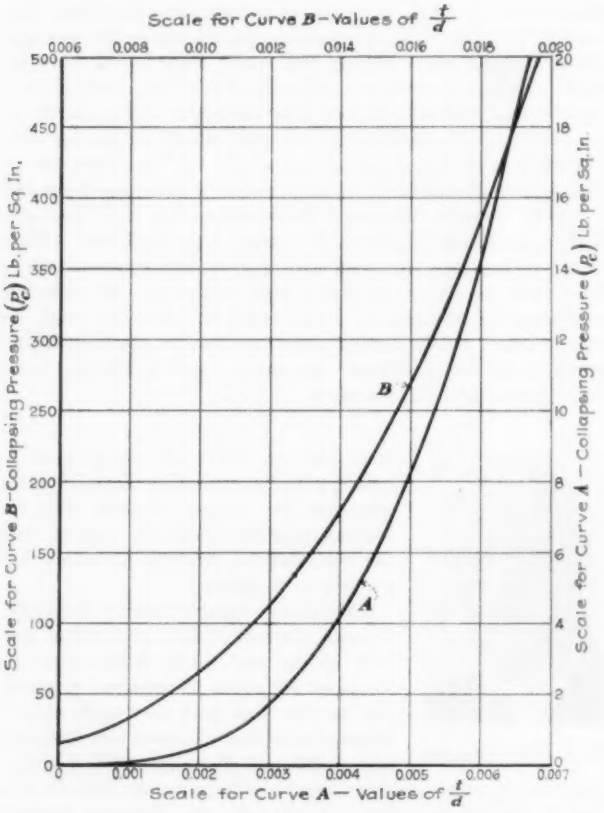


FIG. 1 RELATION BETWEEN COLLAPSING PRESSURE AND t/d RATIO IN THE FORMULA $p_c = 66 \times 10^6 \left(\frac{t}{d}\right)^3$

This is essentially the same as the well-known Föppl formula for the buckling of a circular ring under external pressure:

$$p = \frac{3EI}{r^3L} \quad \ldots\ldots [2]$$

where

p = collapsing pressure
I = moment of inertia of the cross-section of the ring
r = radius of the ring to the neutral axis
L = length of the ring.

When the ring is of rectangular cross-section and unit length, Equation [2] reduces to

$$p = 2E\left(\frac{t}{d}\right)^3 \quad \ldots\ldots [3]$$

This latter differs from Equation [1] only by the factor $\frac{m^2}{m^2-1}$, which is approximately 1.1 when $1/m$ (Poisson's ratio) = 0.3.

[1] Numbers in parentheses are those of references given at the end of the paper.

The pressures represented by these formulas are attainable only with geometrically perfect tubes, when the basic assumptions are fulfilled. For thin steel tubes, where $1/m = 0.3$ and $E = 30 \times 10^6$ lb. per sq. in., Equation [1] becomes

$$p_c = 66 \times 10^6 \left(\frac{t}{d}\right)^3 \quad \ldots\ldots [1a]$$

Stewart's corresponding empirical formula for commercial lap-welded steel tubes (2) and Carman's formula for thin seamless steel tubes (3) is

$$p_c = 50.2 \times 10^6 \left(\frac{t}{D}\right)^3 \quad \ldots\ldots [4]$$

where D is the outside diameter instead of the diameter to the neutral axis as given in all theoretical formulas. This value is some 25 per cent lower than that given by Equation [1a], but it must be remembered that Equation [4] represents the *average* of a great many commercial tubes taken at random from stock. In several individual tests the values predicted by Equation [1a] were obtained. For example, as pointed out by Slocum (4), Stewart's tests Nos. 92 and 94 (2) give the following:

Test No.	t/d ratio	Collapsing pressure: By experiment	Collapsing pressure: By [1a]
92	0.0312	2030	2004
94	0.0323	2200	2224

A graphic representation of Equation [1a] for general use is given in Fig. 1.

Theoretical solutions to the problem of collapse of short tubes have been obtained by Lorenz (5), Southwell (6), and von Mises (7). Southwell developed a formula for the collapsing pressure of a short tube, taking into account the effect of limited length on the elastic behavior of the cylinder. He obtained an expression involving not only all the usual factors, but the number of lobes n and a constant Z, which latter was dependent upon the type of end constraints. This equation is as follows:

$$p = 2E\frac{t}{d}\left[\frac{Z}{n^4(n^2-1)}\left(\frac{d}{L}\right)^4 + \frac{1}{3}\frac{m^2}{m^2-1}\left(n^2-1\right)\frac{t^2}{d^2}\right] \quad \ldots\ldots [5]$$

where n is the number of lobes in a complete circumferential belt at collapse, and Z is a factor depending upon the type of end constraints.

The preceding formula gives distinctly separate curves for successive integral values of n when p is plotted against L/d with a constant ratio t/d. Theoretically, the number of lobes will be that value of n which will produce the minimum collapsing pressure by Equation [5] for any given length. Southwell (8), however, showed that the single hyperbola

$$p = \frac{32}{9}E\frac{t^2}{Ld}\sqrt[4]{\frac{Z}{36}\left(\frac{m^2}{m^2-1}\right)^3\frac{t^2}{d^2}} \quad \ldots\ldots [6]$$

represented very closely the envelope of the family of curves given by various values of n in Equation [5], and that this hyperbola could be used with safety to determine the collapsing pressure of short tubes.

Later, Cook (9) evaluated the constant Z for an ideal type of end constraints. He obtained the value $Z = \pi^4/16$, and showed a very remarkable experimental verification of Equation [5] when this value of Z was used.

If we assume this value of Z and substitute the values $E = 30 \times 10^6$ lb. per sq. in. and $1/m = 0.3$ in Equation [6], we obtain

$$p = 73.4 \times 10^6 \frac{t}{L} \frac{t}{d} \left(\frac{t}{d}\right)^{1/3} \quad \ldots\ldots\ldots\ldots [6a]$$

Curve C of Fig. 2 represents the hyperbola of this equation for $t/d = 0.003$.

The vertical scale represents the collapsing pressure of a tube of given L/d ratio in multiples of the collapsing pressure of a similar tube of infinite length. Denoting this multiplying factor by M, we may then write

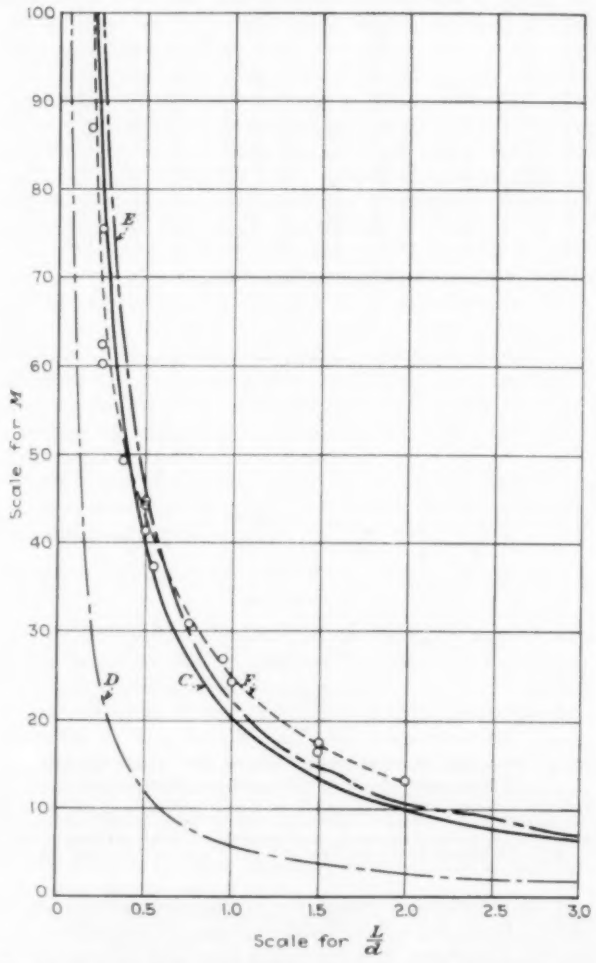


FIG. 2 CURVES ILLUSTRATING COMPARATIVE THEORETICAL AND EXPERIMENTAL DATA

(Curve C, Southwell's single hyperbola, Equation [6]; D, Equation [9] for critical length = $6D$; E, von Mises' Equation [10]; F, experimental curve for best models. These curves are for t/d ratios of approximately 0.003.)

$$M = \frac{p}{p_e} \quad \ldots\ldots\ldots\ldots [7]$$

where p is the collapsing pressure of any tube and p_e is the collapsing pressure of a tube having the same t/d ratio but of infinite length. However, if we divide Equation [6a] by Equation [1a], we obtain

$$M = \frac{p}{p_e} = \frac{1.11}{L} d \sqrt{\frac{d}{t}} \quad \ldots\ldots\ldots\ldots [8]$$

It is evident that Equation [8] will give a different curve for M against L/d for each value of t/d. This is actually seen to be the case from the experimental data obtained by Cook (10).

It will also be noted that Equation [8] can be written as

$$p = \frac{p_e}{L} \; kd \sqrt{\frac{d}{t}} \quad \ldots\ldots\ldots\ldots [8a]$$

where the factor $kd\sqrt{d/t}$ is the expression for the "critical length" given by Southwell (6). Equation [8a] then becomes

$$p = \frac{p_e L_c}{L} \quad \ldots\ldots\ldots\ldots [9]$$

where L_c is the critical length, defined as the minimum length beyond which the resistance of the tube to collapse is independent of the length, and p_c is the pressure at which a tube of critical length will collapse as calculated by Equation [1].

This is identical with the expression given by Carman (11). Carman, however, says that the critical length is equal to about $6D$, and hence is independent of the t/d ratio. Curve D, in Fig. 2, shows the hyperbola represented by Equation [9] when the critical length is placed equal to $6D$. If this hyperbola were used for the ratio $t/d = 0.003$, the experimental values would exceed the theoretical by as much as 400 per cent.

Cook's experimental work (10) seems to confirm Southwell's theoretical expression for the critical length, but he obtains the numerical value 1.73 for the constant k, instead of 1.11 as indicated by Equation [8]. Cook's values of t/d ranged from 0.0097 to 0.0146. The values used in the tests conducted thus far at the Experimental Model Basin are from 0.003 to 0.004.

The first to develop a theoretical expression for the pressure at which short tubes *under end load as well as circumferential load* would collapse by instability was von Mises (12), who gave the following:

$$p = \left\{ \frac{\frac{E}{n^2}\frac{t}{r}}{\left[1 + \left(\frac{n}{\pi}\frac{L}{r}\right)^2\right]^2} + \frac{n^2}{12}\left[1 + \left(\frac{\pi}{n}\frac{r}{L}\right)^2\right]^2 \frac{m^2 E}{m^2 - 1}\frac{t^3}{r^3} \right\} \times \frac{1}{1 + \frac{1}{2}\left(\frac{\pi}{n}\frac{r}{L}\right)^2} \quad \ldots\ldots [10]$$

The evaluation of this formula is rather complicated, for its use involves finding the value of n which gives the lowest collapsing pressure.

Curve E, Fig. 2, represents values calculated from von Mises' Equation [10].

Curves C, D, and E, Fig. 2, cannot be extended indefinitely to low values of the ratio L/d because, under these conditions, collapse by pure instability will no longer occur. Von Sanden and Günther (12), who have made a theoretical investigation of this phase, find that collapse of very short tubes (L/d less than about 0.20) is brought about by development of high stresses adjacent to the stiffening or supports, which stresses exceed the proportional limit and reach the yield point of the material.

Actual measurements on many models, similar to those listed in Table 1, having low L/d ratios (of about 0.15), indicate that collapse is brought about by overstressing the material before there is any indication of the lobe formation which precedes collapse by instability.

Since all the preceding formulas contain E, they cannot be expected to hold strictly under such conditions. Von Kármán (13) has brought out to excellent advantage the variation in modulus of elasticity in the vicinity of the yield point. These matters, however, pertain to a fundamentally different type of collapse, and it is proposed to defer discussion of them to a subsequent paper.

(NEW) TABLE 1

Model dentification number, series III	t, in.	d, in.	L_e, in.	t/d	L/d	p by expt.	p_c Eq. [1a]	$M = p/p_c$	p_5 Eq. [5]	p_{10} Eq. [10]	Number of lobes: By expt.	Calculated: Eq. [12]	Eq. [5]	Eq. [10]	Maximum variation in radius expressed in fractions of thickness: Entire circum.	Region of one lobe	Angle between maximum and minimum radii: Entire circum.	Region of one lobe	Angle between center of first lobe and minimum radius
2000D50T1	0.0500	16.0	32.0	0.00312	2.00	26.5	2.00	13.2	20.8	21.7	5	4.88	5	5	0.47	0.47	11°	11°	8°
1500D50T1	0.0530	16.0	24.0	0.00331	1.50	41.5	2.39	17.4	33.2	34.6	6	5.55	6	6	..	..	..	..	..
1500D50T2	0.0510	16.0	24.0	0.00319	1.50	35.5	2.14	16.6	29.6	31.4	6	5.60	6	6	0.53	0.53	23°	23°	4°
1000D50T1	0.0530	16.0	16.0	0.00331	1.00	58.0	2.39	24.3	48.8	51.1	8	6.79	7	7	0.36	0.32	40°	23°	21°
938D50T1	0.0500	16.0	15.0	0.00312	0.938	53.5	2.00	26.8	44.9	46 7	7	7 24	7	7	0.33	0.33	77°	42°	9°
750D50T1	0.0505	16.0	12.0	0.00316	0.75	64.5	2.08	31.0	57.9	60.4	9	7.94	8	8	0.30	0.16	144°	24°	7°
546D50T1	0.0476	16.0	8.75	0.00297	0.54	64.5	1.73	37.3	68.8	71.8	9	9.50	9, 10	9	0.50	0.50	25°	25°	6°
500D50T1	0.0520	16.0	8.0	0.00325	0.50	94.0	2.27	41.4	94.0	99.3	10	9.65	10	9	0.43	0.37	130°	36°	10°
500D50T2	0.0510	16.0	8.0	0.00319	0.50	96.0	2 14	44.9	89.5	94.6	9, 10	9.70	10	9	0.33	0.25	105°	30°	45°
500D50T3	0.0490	16.0	8.0	0.00306	0.50	83.5	1 89	44.2	80.5	85.8	9, 10	9.80	10	9	0.26	0.18	73°	30°	95°
375D50T1	0.0510	16.0	6.0	0.00319	0.37	106.0	2 14	49.5	118.5	128.1	11	11.30	11	11	0.43	0.33	113°	14°	35°
250D50T1	0.0516	16.0	4.0	0.00323	0.25	134.0	2 22	60 4	182.0	204.7	13, 14	13.67	14	13	0.17	0.11	49°	9°	2°
250D50T2	0.0518	16.0	4.0	0.00324	0.25	140.0	2.24	62.5	183.0	205.8	13, 14	13.66	14	13	0.19	0.19	78°	8°	3°
250D50T3	0.0512	16.0	4.0	0.00320	0.25	163.0	2.16	75.5	180.5	202.3	13, 14	13.72	14	13	0.18	0.13	166°	9°	20°
188D50T1	0.0493	16 0	3.0	0.00308	0.188	168.0	1.93	87 0	218.5	252.6	16	15.96	16	14	0.29	0.22	66°	23°	6°
125D50T1	0.0450	16.0	2.0	0.00281	0.125	195.0	1.46	133.6	260.6	323.0	19	20.02	20	17	0.23	0.16	37°	19°	6°

$E = 30 \times 10^6$ lb. per sq. in. $1/m$ (Poisson's ratio) = 0.3 All models subject to full axial as well as circumferential loading.

Elastic Behavior of a Cylindrical Vessel Under External Pressure

When a long, thin tube is subjected to external pressure, it first contracts uniformly around the circumference; that is, the circumferential elements of the shell, loaded similarly to long slender columns, will suffer uniform contraction or shortening before bending.



Fig. 3 Model of Thin Cylindrical Vessel With Closely Spaced Internal Stiffening Rings, Showing Symmetrical Lobe Formation at Collapse

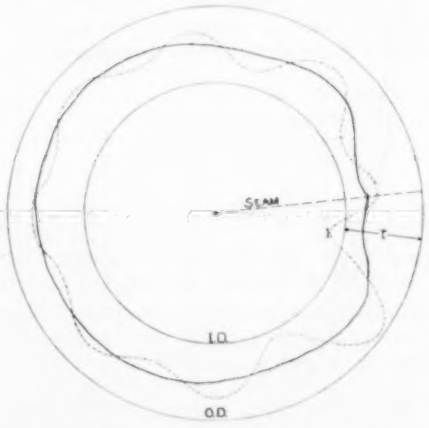


Fig. 4 Diagram Representing Shape of Thin Cylindrical Vessel and Mode of Collapse by Instability

(Full line, original contour of middle of shell; broken line, contour just prior to collapse. Position of lobe formation of entire model is influenced by position of lobe at the point 1, where the original local variation in radius was great. Collapse finally occurred at point 1 and at the two adjacent lobes.)

As the pressure is increased, the circular elements will assume elliptical forms and will eventually collapse by instability. The tube will flatten out, with two bulges extending outward beyond the original circular form and two extending inward, forming what is known as two-lobe collapse.

If circumferential supports are provided in the tube, at a distance apart less than the critical length, the presence of these supports causes a reduction in the length of the equivalent circumferential columns. This is analogous to the well-known subdivision of a simple slender strut into bays, with the actual neutral axis crossing the original axis at intermediate points.

The circumferential elements of the tube between supports take the form of long, thin plates, loaded longitudinally and stiffened along their edges. These plates will buckle into numerous bays or lobes, depending generally upon the proportion of width to length of the plates.

The subdivision of the original circumferential columns in a cylindrical vessel evidences itself in an increase in the number of lobes above two, and a considerable increase, as in the simple column divided into numerous bays or bulges, of the collapsing or buckling load. It is this phenomenon which gives rise to the increased strength of a tube with decrease in length.

Von Sanden and Günther (12) state that Equation [10]

"has always given the number of bulges on the circumference, and, for isolated bulges, the calculated length of the bulge has always checked with the observed length, even in cases where the experimental bodies were far from being ideal in form and scantlings." On the other hand, Cook (9), in referring to Equation [5] declares that "the number of lobes in the collapsed tube does not correspond with that indicated by theory. . . . . It is to be expected that the observed number of lobes will usually be *less* than that predicted by theory, and this is actually seen to be the case."

The tests thus far conducted at the Experimental Model Basin, as shown in Table 1, have in nearly every case given the correct number of lobes as predicted by both Equations [10] and [5]. The one notable exception is the model No. 1000D1 shown in Fig. 4, where the actual number of lobes is seen to be 8, while Equations [10] and [5] both predict 7. Thus, the number of lobes is *greater* than predicted by theory.

It is possible that Cook, who states that "in most cases the collapse was fairly symmetrical," may have been misled by the fact that all the lobes in any one circumferential belt did not appear upon collapse, for it is by no means certain that a complete circle of bulges will be found. The photograph, Fig. 3, represents a typical case, where only a few bulges appear. If, however, the center-to-center distance for each pair of bulges is stepped off around the circumference the correct number of lobes (in this case 16) will be found.

The number of lobes will in service be found to vary from 2 for a long tube to as high as 17 for tubes with closely spaced circumferential stiffeners (14).

Effective Unsupported Length of a Tube and Method of Stiffening

It is fairly well established that the length L may be taken as the length of the unsupported cylinder between adjacent transverse or circumferential supports, whether those supports are the solid ends of the vessel, whether they are bulkheads or partitions, or whether they are only circumferential rings or stiffeners, provided they possess rigidity in their own plane relatively greater than the shell of the vessel, and provided they extend continuously around the circumference. Hemispherical ends may be reckoned effective in preserving the circular section at their point of junction with the cylindrical shell, in the same manner as flat or dished ends.

As a measure of the necessary rigidity for a circumferential stiffener, recourse may be had to Equation [2] by inserting for the factor L the length of the unsupported wall or shell *between* stiffeners. This will give the required moment of inertia for a continuous stiffening ring of uniform section which will collapse by instability simultaneously with the shell. It has been customary, however, to make the stiffener 10 per cent stronger than the shell (14), in which case the formula becomes $1.1p = 3EI/r^3L$, or

$$I = \frac{0.37\, r^3 L}{E}\, p \qquad [11]$$

The stiffening rings may take on manifold forms as shown in Fig. 5. Circumferential supports for trays, platforms, and other accessories may be made to serve as stiffeners, but for this purpose they should be continuous and attached to the shell.

Frames, stiffening rings, and supports may be either internal or external, and may be attached by riveting, welding, or other convenient means. In special cases where the shell must remain intact, it may be possible to design the internal stiffening rings so that they simply rest against the shell. Shell connections to external rings must be substantial enough to cause the rings to contract under load with the shell.

Effect of Variation From True Cylindrical Form

One would not expect columns with varying amounts of eccentricity to support equal loads before collapse, nor to support the maximum load of which a straight column is capable. It should not therefore be expected that cylindrical tubes or vessels having variations from circular sections will withstand external pressure alike nor to the maximum degree of which the material is capable.

This is borne out by the experience of many experimenters,

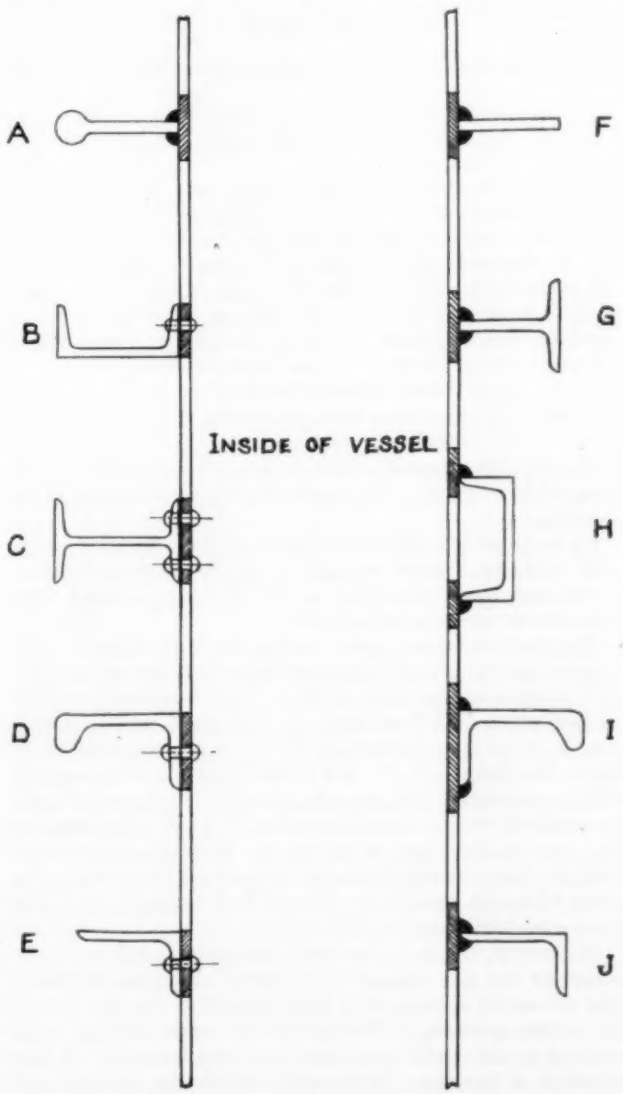


Fig. 5 Forms for Stiffening Rings

(The hatched portions of the shell represent approximately the amounts which may be reckoned effective in calculating the moment of inertia of the frame. Note that for riveted construction, reckon only that portion of shell effective which lies under the faying flange. Compare types *I* and *D*. Types shown as riveted may also be welded. Welding need not be continuous; should preferably be intermittent in many cases. *A*, bulb plate, welded. Probably the lightest, simplest, and most effective design. At present this section is only rolled abroad, but should be introduced into this country. *B*, channel, riveted (or welded). Simple design but unbalanced section, which will tip under heavy load. *C*, I-beam, riveted (or welded). Symmetrical, but probably unnecessarily heavy and difficult to fabricate. *D*, bulb angle, riveted. Good section except for unsymmetrical position of web and bulb. *E*, angle, riveted. Simple but unbalanced, in that the neutral axis lies too close to the faying flange. *F*, flat-plate ring, welded. Light, simple, and effective. *G*, T-bar, welded on stem. Good section, but somewhat difficult to fabricate. *H*, channel, welded on toes. Attractive design, but probably needlessly heavy and involving dead space for corrosion. *I*, bulb angle, welded. Good section except for unsymmetrical position of web and bulb. *J*, angle, welded on toe. Simple but lacking in symmetry. Will tip under extreme load.)

notably Stewart and Carman, who found it necessary to reduce the coefficient 66×10^6 in the rational Equation [1] to about 50×10^6 for *average* commercial tubes. Numerous tubes in Stewart's tests (2) collapsed at much lower pressures, which led Slocum (4) to state that "instead of substituting the actual radius of the tube in Equation [1], there should be substituted its radius of curvature at the *flattest portion*. Considering the tube to be elliptical in section, this will be the radius of curvature of the ellipse at the extremity of its minor axis, which is $R = a^2/b$, where a and b are the semi-axes. Since $a = \frac{D_{max.}}{2}$ and $b = \frac{D_{min.}}{2}$, this becomes $2R = \frac{D^2_{max.}}{D_{min.}}$, and substituting this corrected value of the diameter in Equation [1] it becomes

$$p_c = 2\frac{m^2}{m^2 - 1} E \left(t \frac{D_{min.}}{D^2_{max.}} \right)^3 \quad \ldots\ldots\ldots\ldots [1b]$$

This correction is of course only approximate, but should give good results for the range of eccentricities in question."

This was a step in the right direction, but it probably applies only to two-lobed collapse, although this feature was not brought out at the time. For the case of collapse of tubes into higher-lobed formations, it is readily seen that an initial dent or bulge approximating the shape and size of one part of a lobe will be the equivalent of eccentric column loading at that point.

In the model whose cross-section is given in Fig. 4, there were noticeable departures from the circular form in the vicinity of the seam. There was, however, no appreciable reduction in the collapsing pressure, because the variations from circular form were not excessive as compared with the thickness of the material.

On early models, having variations of the order of twice the shell thickness, collapse occurred irregularly at pressures much lower than those calculated, so that the tests had no value whatever for comparative purposes.

This leads to the conclusion that for short tubes, simple variation in diameter is a less important factor than is local variation in the region of one bulge or lobe. This is strikingly evident in tests which have been made on both small- and large-scale models, up to 8 ft. diameter, at the U. S. Experimental Model Basin, Washington, D. C., and at the Navy Yard, Portsmouth, N. H.; so much so that the point of initial failure can generally be predicted from a careful inspection of the diagram depicting the exact initial form, as in Fig. 4. However, experimental work at these two establishments has not yet progressed to the point where the quantitative effect of local or general variations from cylindrical form can be predicted.

In practice, the departure from circular form is found to increase as the size (diameter) decreases. Working allowances and tolerances decrease at a lesser rate than the size or scale of the tube or vessel; stiffening rings and supports are generally omitted in the smaller sizes, and dents and bulges are of more common occurrence. Fortunately, allowances for hot and cold working, corrosion, and the like are greater on small than on large tubes, so that some compensation is thereby obtained.

It is perhaps somewhat premature to set up any definite tolerances for the fabrication of tubes and vessels subjected to external pressure, especially of the built-up variety, but it may be safe to state that for thin tubes the *total variation in radius* at all points shall not exceed the shell thickness; that the middle of the shell plating in any circular section shall lie between two circles representing the outside and the inside diameter, drawn from the geometrical center of the shell. Furthermore, as brought out previously, extreme variations in radius within the region of any one bulge are to be avoided.

For comparative and systematic tests on models and structures, the fabrication tolerance should be reduced to one-half—or better, to one-third—of this amount. The model from which the data in Fig. 4 were taken complies with the latter requirement.

Manifestly, the establishment of such a requirement involves the use of some method whereby compliance with the given specifications or whereby variation in circular form of existing structures may be indicated. A simple device of this kind, as shown in Fig. 6, has been developed and used at the Navy Yard, Portsmouth, N. H. It consists of a temporary central bar or spindle upon which is mounted a swinging arm of pipe or tubing, carrying at its outer end a tracing wheel with pointer and multiplying mechanism, and a chart. The chart is graduated in radial lines representing degrees or divisions of a whole circle, and in concentric circles representing given absolute radial displacements of the tracing wheel.

A belt (or chain) mounted on two pulleys, one clamped on the central spindle and one on the chart holder, maintains the chart in a given direction in space as the arm is rotated. The simple operation of rolling the tracing wheel around the inside of the tank causes an automatic record, to a magnified scale,

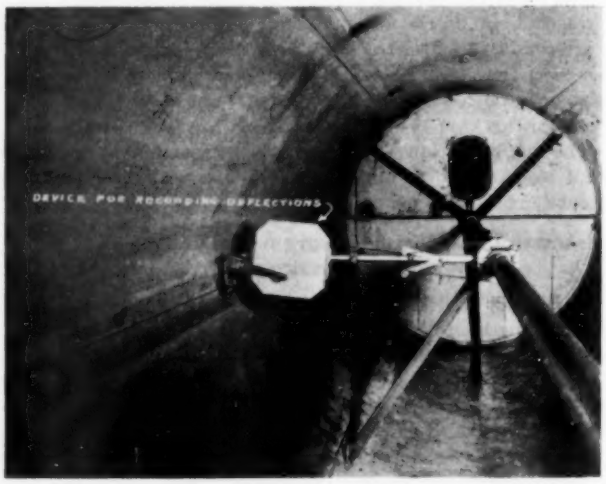


FIG. 6 DEVICE FOR TRACING CONTOURS OF SECTIONS

to be made of the exact shape of the section corresponding to the heavy full line on Fig. 4. By moving the arm along the spindle, similar records may be made at other sections.

EFFECT OF LOCAL STIFFENING, CONCENTRATED LOADS, ETC.

Much more extensive study and research than have already been undertaken will be necessary before definite answers can be given on this phase of the subject.

For guidance, however, it may be stated that since the shell or wall of a cylindrical vessel contracts nearly uniformly under external pressure to about the elastic limit of the material, anything which *prevents* the shell from contracting in this manner may cause unnatural deformation and may result in the equivalent of eccentric loading, thus weakening the structure. Hence the paradox that very stiff frames or rings, numerous solid bulkheads, and heavy longitudinal seams *may* result in a *reduction* of the collapsing pressure. The best and most economical design is that in which the various members are each just strong and rigid enough to carry the designed collapsing load.

SCALE EFFECTS IN TESTS OF MODELS AND IN EXTRAPOLATION OF DATA

Since t, L, and d in the various formulas are all functions of a

linear dimension, the constant k, as found by a relation such as $50.2 \times 10^6 = k \frac{m^2}{m^2 - 1} E$, as well as the factor Z for types of end constraints, are both non-dimensional. Collapsing pressure p is therefore independent of scale *so long as geometrical similarity is maintained* and so long as the physical properties E and $1/m$ of the material remain constant. To comply with the requirements for geometrical similarity, variations in form should be of the order of the scale. For example, the diagram, Fig. 4, would apply equally to any tube or vessel, whether 1 in. or 100 ft. in diameter, simply by expanding the scale. Available data on reliable tests with models and cylinders of various sizes verify the conclusion that scale effect is practically non-existent.

Experimental Data

The experiments of Fairbairn (15) were made with riveted steel pipes, which do not represent geometrically similar models of the pressure vessels being constructed at the present time.

The tests of Stewart (2) were conducted on commercial lap-welded steel tubes, which were probably inferior in respect to adherence to cylindrical form to those being turned out today.

The tests of Carman (3 and 11) were all with quite small tubes of various metals, and those of Cook (9 and 10) were on small tubes of brass and steel.

The tests made in Germany during the World War (14), although on a scale more nearly commensurate with the large cylindrical vessels under discussion in this paper, were made on admittedly inferior models, and the results show it. The German experimenters (16) stated that owing to unavoidable departure from circular form the collapsing pressures as found by Equation [10] should be reduced by the use of coefficients as low as 0.4, 0.5, and 0.6. It is to be noted, however, that they were applying this formula to very low values of L/d where, as stated above, instability formulas such as Equation [10] are not directly applicable.

The great majority of the published data are for pipes or tubes having an L/d ratio of 4 or more; hence they are of little assistance in this problem. Physical characteristics of the materials used are in general not given, although they may be assumed with reasonable accuracy.

The great stumbling block to careful analysis of previous experimental data is the almost total lack, especially for large diameters, of information as to "out of roundness." For example, the German experimenters (14) report that their vessels were not strictly circular in cross-section, but they do not describe in detail such irregularities as were found to cause 2-to-1 variation in collapsing pressure for nearly identical models.

Table 1 gives the results of tests made on structural models of simple cylindrical forms by the U. S. Experimental Model Basin, Washington, D. C., under the cognizance of the Bureau of Construction and Repair, Navy Department. The models are built of high-grade sheet steel as used for metal-furniture manufacture, to obtain uniform thickness and curvature throughout.

They are made with heavy jigs which serve to hold the shell in correct shape during assembly. Difficulties are experienced in obtaining the proper curvature adjacent to the longitudinal seam, but the irregularities are probably no greater than those obtaining in the best commercial practice. Charts similar to Fig. 4 are constructed for several stations in each model, and curves showing variation in circular form are drawn for successive pressures.

Curve F of Fig. 2 represents values of the multiplier M for these models, based upon the experimental collapsing pressure and Equation [1a]. The t/d ratio for the present series has been limited to about 0.003; it is hoped to give similar data for other ratios in subsequent papers.

It will be noted that for values of the L/d ratio greater than 0.5, the experimental collapsing pressure is considerably above the theoretical pressure for ideal tubes, while for values of the L/d ratio below 0.5, the experimental pressures fall below the theoretical.

In all computations, the values $E = 30 \times 10^6$ lb. per sq. in. and $1/m = 0.3$ were assumed. The shell plating of the models so far tested is rather thin to permit accurate determination of these physical properties.

It will also be noted (by reference to Table 1) that those models which collapsed at pressures below the theoretical values were inferior in adherence to circular form.

It must be emphasized that in experiments and analyses of this kind, the matter of geometrical and physical similarity amounts to very much more than fine talk on the subject. Nonconformity to cylindrical form and non-similarity between specimens is unquestionably responsible for 90 per cent or more of the discrepancies found by experimenters in the past. For this reason, experimenters in the future should always determine and give the exact shape of the cylinders they are testing and the physical properties of the material they use.

Factors of Safety

It will be noted that, throughout this discussion, the *working* pressure of the tube or vessel has not yet been mentioned. It is considered preferable to execute the design upon the basis of *collapsing* pressure, as having a much more direct bearing, for example, upon the physical properties of the material and as offering an opportunity for confirmation by model tests.

The designed collapsing pressure may be arrived at by a combination of the working or test pressure and the factor of safety.

The latter is to be selected by judgment or by experience, embodying allowance for reduction in actual overcalculated collapsing pressure due to unavoidable out-of-roundness, unknown effects of local or non-uniform stiffening, and indeterminate stresses due to loads or forces foreign to the static pressure load.

In this case, $p = FP$, where F is the factor of safety and P is the working pressure.

For the selection of the factor of safety F in his empirical formula, Stewart (2) gives some excellent general rules. It must be remembered, however, that the numerical values of the factors which he gives must be modified for use in the rational formulas recommended in this paper. For example, for steel tubes, the coefficient of Stewart's Equation [4] would be 50.2×10^6, whereas that of the rational formula for ideal tubes would be 66×10^6. Definite rules for the selection of this factor can hardly be given, since the complex circumstances surrounding each design will have to be considered and properly evaluated.

It may be pointed out here that careful study of the influence of variation of circular section on factor of safety shows that a decrease in that factor with an increase in diameter is entirely reasonable. As previously brought out in somewhat different form, it will be found that tolerances of several per cent of the diameter are customary and perhaps necessary on small diameters of the order of 1 ft., whereas it will be economically possible to limit variations on large vessels, of a diameter of 10 ft. or more, to fractions of 1 per cent. Likewise, any deviation from circular form due to damage or excessive pressure is much more easily detected on the larger than on the smaller sizes.

Practical Application of Design Formulas

For the present it appears advisable to use the combination of Bryan's rational formula, Equation [1], for tubes of length greater than the critical length, and Southwell's approximate hyperbola, Equation [6], for shorter tubes. This was the suggestion of Southwell in 1915 (8), but at that time he had not determined a value for Z and had no experimental verification of his formula, Equation [6]. A comparison of the curves C and E with the experimental curve F (Fig. 2) shows that either Equation [6] or Equation [10] can be used with safety for values of L/d between 0.50 and 2.0 and for a t/d ratio of 0.003.

It must be emphasized that these curves represent only one

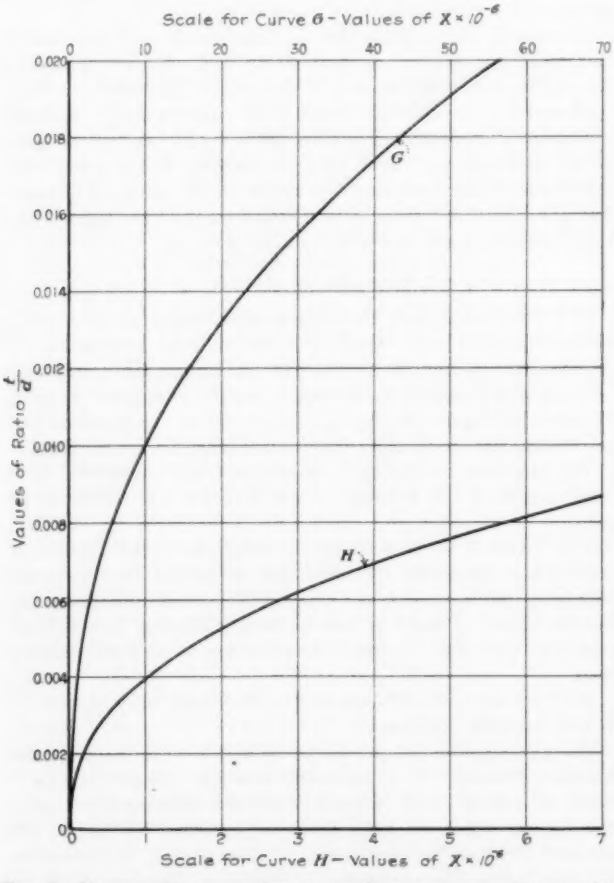


Fig. 7 Curves Showing Relation Between the Ratio t/d, p, and L/d in the Equation: $t/d = \left(\frac{p\ L/d}{\text{constant}}\right)^{2/5} = X^{2/5}$

value of t/d. However, this value was chosen at random, and there is no reason why any other value should not fit the theoretical formulas as well. Nor is there any reason to believe that the theoretical formulas will not hold when the range of L/d is extended to the critical length. However, more comprehensive tests should be made.

From the present meager data available, there seems to be little choice between Equations [5], [6], and [10]. Equations [5] and [10] are undoubtedly more accurate than [6], which, as previously stated, was intended only as a hyperbolic curve approximating the envelope represented by Equation [5]. However, the simplicity with which Equation [6] can be used in making computations is very desirable. When Z is replaced by $\pi^4/16$ and the proper values for E and $1/m$ are substituted for the material used, an equation of the type [6a] is obtained, which can be expressed as

$$\frac{t}{d} = \left(\frac{p\ L/d}{\text{constant}}\right)^{2/5} = X^{2/5} \qquad [6b]$$

where the constant for any particular material is given by

$$2.28\ E\left(\frac{m^2}{m^2 - 1}\right)^{3/4}$$

For steel tubes where $E = 30 \times 10^6$ lb. per sq. in. and $1/m = 0.3$, the constant takes the value 73.4×10^6 as given in Equation [6a].

When the design conditions have been stated and p, L, and d are known, X can be evaluated and the correct thickness determined from the relation

$$\log \frac{t}{d} = \frac{2}{5} \log X$$

or from the curve, Fig. 7.

Equations [5] and [10] have the great inconvenience that they cannot be solved directly for t. However, once t has been calculated by Equation [6], it is desirable to use them as a check on the collapsing pressure. As already stated, their use involves finding the number of lobes n which cause the collapsing pressure to be a minimum for a given t, L, and d.

It can be shown that the relation

$$n = \sqrt[4]{\frac{7.06}{\left(\frac{L}{d}\right)^2 \frac{t}{d}}} \qquad [12]$$

determines approximately the required value of n in Equation [5]. Since n represents the number of lobes, the integral number nearest to the value of n given by Equation [12] should be used. Methods for finding the required value of n in Equation [10] have also been developed.

The exact values of n obtained by the use of Equation [12] are given in Table 1, together with the integral values of n by Equations [5] and [10], and the number of lobes obtained by experiment.

The discrepancies in integral values of n in Table 1 are more apparent than real, since n is determined for a minimum value of p at a point where the variation of p with n is small. Substituting either value of n in the formulas will be found to give approximately the same value of p.

The only case where neither Equation [5] nor Equation [10] gave the correct number of lobes is model No. 1000D1. This is as yet unexplained.

Application to Practice

Example 1. Required: To design the shell of a closed cylindrical steel tank, working under full vacuum.

Given: $D = 10$ ft., $L = 12$ ft., $P = 14.7$ lb. per sq. in. No stiffeners or supports except the ends. Take $F = 4$, $E = 30 \times 10^6$ lb. per sq. in., and $1/m = 0.3$.

Solution:

$p = 4 \times 14.7 = 58.8$ lb. per sq. in. (say, 60 lb.)
$d = 120$ in. (approximately)
$L = 144$ in.
$L/d = 144/120 = 1.2$

Since $E = 30 \times 10^6$ lb. per sq. in., and $1/m = 0.3$, the constant in Equation [6b] becomes

$$\text{constant} = 2.28 \times 30 \times 10^6 \left(\frac{1}{0.91}\right)^{3/4} = 73.4 \times 10^6$$

$$\frac{t}{d} = \left(\frac{p\,L/d}{73.4 \times 10^6}\right)^{2/5} = \left(\frac{60 \times 1.2}{73.4 \times 10^6}\right)^{2/5}$$

$$= (0.9809 \times 10^{-6})^{2/5}$$

From Fig. 7, $t/d = 0.00393$, whence $t = 0.47$ in. Analytically, from formula $t/d = (X)^{2/5}$,

$\log t/d = {}^2/_5 \times \log (0.9809 \times 10^{-6}) = 0.4(3.99162 - 10)$

$t/d = 0.003950$

whence

$t = 0.00395 \times 120 = 0.474$ in.

Check by Equations [5] and [10].

From Equation [12],

$$n = \sqrt[4]{\frac{7.06}{\left(\frac{L}{d}\right)^2 \frac{t}{d}}} = \sqrt[4]{\frac{7.06}{(1.2)^2 \times 0.00395}} = \sqrt[4]{1241}$$

$$= 5.94$$

$n = 6$

Substituting in Equation [5] and assuming $Z = \frac{\pi^4}{16}$,

$$p = 2E\frac{t}{d}\left[\frac{Z}{n^4(n^2-1)}\left(\frac{d}{L}\right)^4 + \frac{1}{3}\frac{m^2}{m^2-1}\left(n^2-1\right)\frac{t^2}{d^2}\right]$$

$$p = 2 \times 30 \times 10^6 \times 0.003950\left[\frac{\frac{\pi^4}{16}}{1296 \times 35} \times \frac{1}{(1.2)^4} + \frac{1.1}{3} \times 35 \times (0.00395)^2\right]$$

$$= 237{,}000\ (0.0000647 + 0.0002002)$$

$p = 237{,}000 \times 0.0002649 = 62.8$ lb. per sq. in.

A similar substitution in Equation [10] gives

$p = 65.6$ lb. per sq. in.

Since $t/d = 0.00395$, we have, by Equation [1],

$$p_e = 2\frac{m^2}{m^2-1}E\left(\frac{t}{d}\right)^3$$

$$= 66 \times 10^6 \times 0.06163 \times 10^{-6}$$

$$= 4.07 \text{ lb. per sq. in.}$$

By Fig. 2, the multiplier M for $L/d = 1.2$ from curve C is 16.6.

$$p = Mp_e = 16.6 \times 4.07 = 67.6 \text{ lb. per sq. in.}$$

It must be remembered that this method gives only a rough check when t/d differs appreciably from 0.003, for which the curve C of Fig. 2 is drawn. For higher values of t/d this method gives collapsing pressures which are too high.

Example 2. Required: To determine the thickness of the cylindrical inner shell of a jacketed heater, reinforced by stiffening rings spaced to suit local conditions.

Given: $D = 6$ ft., $L = 2$ ft., $P = 150$ lb. per sq. in. Take $F = 6$. The material of the shell and stiffening rings is nickel alloy, for which $E = 26 \times 10^6$ and $1/m = 0.3$.

Solution:

$p = 6 \times 150 = 900$ lb. per sq. in.
$d = 72$ in. (approximately)
$L = 24$ in.
$L/d = 24/72 = 0.33$
$E = 26 \times 10^6$ lb. per sq. in.
$1/m = 0.3$

Using Equation [6b]

$$\text{constant} = 2.28 \times 26 \times 10^6\left(\frac{1}{0.91}\right)^{6/4} = 63.6 \times 10^6$$

$$\frac{t}{d} = \left(\frac{pL/d}{63.6 \times 10^6}\right)^{2/5} = \left(\frac{900 \times 0.33}{63.6 \times 10^6}\right)^{2/5} = (4.67 \times 10^{-6})^{2/5}$$

From Fig. 7, $t/d = 0.0074$, whence $t = 0.53$ in. Analytically, from formula $t/d = (X)^{2/5}$,

$\log t/d = {}^2/_5 \log (4.67 \times 10^{-6}) = 0.4(4.66932 - 10)$

$t/d = 0.00737$

whence

$t = 0.00737 \times 72 = 0.53$ in.

Check by Equations [5] and [10].

From Equation [12],

$$n = \sqrt[4]{\frac{7.06}{\left(\frac{L}{d}\right)^2 \frac{t}{d}}} = \sqrt[4]{\frac{7.06}{0.1089 \times 0.00737}} = \sqrt{8796} = 9.68$$

$n = 10$

Substituting in Equation [5], we get

$p = 954$ lb. per sq. in.

Substituting in Equation [10], we get

$p = 1086$, for $n = 9$

and $p = 1149$, for $n = 10$

By Equation [1], $p_e = 57.2 \times 10^6 \times (0.00737)^3 = 22.9$ lb. per sq. in.

The multiplier from Fig. 2 for $L/d = 0.33$ from curve C is 61, whence

$$p = Mp_e = 22.9 \times 61 = 1397 \text{ lb. per sq. in.}$$

As stated under Example 1, the collapsing pressure thus obtained is too high, because the t/d ratio greatly exceeds 0.003 for which curve C is drawn.

To calculate the required moment of inertia of the stiffening rings, assuming they are of the same material as the shell, we have, from Equation [11],

$$I = \frac{0.37 \times (36)^3 \times 24 \times 900}{26 \times 10^6} = 14.34 \text{ in.}^4$$

After selection of a type, as in Fig. 5, the necessary scantlings for the rings can be determined by application of the rules there given.

For example, suppose a channel frame is desired for riveted construction. The American standard 5-in. channel C-8 has the required scantlings, thus: Area of section, 2.63 sq. in.; width of flange, 1.885 in.; moment of inertia, 8.8 in.^{4} We consider a strip of the shell plating equal in width to the flange of the channel to work with the channel as a portion of the stiffener; it is therefore reckoned effective in calculating the moment of inertia of the ring as a whole. (See *B*, Fig. 5.) The centroidal axis of this compound section is shifted from the center of the web of the channel toward the shell a distance of 0.761 in., while the total moment of inertia of the compound section becomes 14.358 in.^{4}, distributed as follows:

I of channel about new axis $= 10.323$ in.^{4}
I of shell about new axis $= 4.035$ in.^{4}
I of compound section $= 10.323$ in.$^4 + 4.035$ in.^{4}
$= 14.358$ in.^{4}

Conclusion

In the last analysis, the acceptability of any formula or procedure used by engineers for design purposes rests largely upon the degree of exactitude with which results can be predicted. To permit the final selection of a suitable formula or formulas for the strength of vessels under external pressure and the insertion of the correct constants and coefficients, it will be necessary to accumulate many times more data than are available at present.

This can be accomplished only by the active cooperation of all who have designed, built, used, and tested structures of this kind.

The submission of complete data by which Table 1 might be filled out for each such example or installation would enable analysis of these data and preparation of a definite code to proceed at once, with hope of a satisfactory conclusion in the not distant future.

References

(1) Bryan, G. H., "Application of the Energy Test to the Collapse of a Long Thin Pipe Under External Pressure," Cambridge Philosophical Society Proceedings, Vol. VI, 1888, pp. 287–292.
(2) Stewart, R. T., "Collapsing Pressure of Bessemer Steel Lap-Welded Tubes, Three to Ten Inches in Diameter," Trans. A.S.M.E., Vol. 27, 1906, pp. 730–822.
(3) Carman, A. P., and Carr, M. L., "Resistance of Tubes to Collapse," University of Illinois Engineering Experimental Station, Bulletin No. 5, June, 1906.
(4) Slocum, S. E., "The Collapse of Tubes Under External Pressure," London *Engineering*, Vol. 87, Jan. 8, 1909, pp. 35–37.
(5) Lorenz, R. v., "Die nicht achsensymmetrische Knickung dünnwandiger Hohlzylinder," *Physikalische Zeitschrift*, No. 12, Apr., 1911, pp. 241–260.
(6) Southwell, R. V., "On the General Theory of Elastic Stability," Philosophical Transactions, Vol. 213-A, pp. 187–244.
(7) Mises, R. v., "Der kritische Aussendruck zylindrischer Rohre," *Z.V.D.I.*, Vol. 58, No. 19, May, 1914, pp. 750–755.
(8) Southwell, R. V., "On the Collapse of Tubes by External Pressure," *Philosophical Magazine*, January, 1915, pp. 67–77.
(9) Cook, G., "The Collapse of Short Thin Tubes by External Pressure," *Philosophical Magazine*, Oct., 1925, pp. 844–848.
(10) Cook, G., "The Collapse of Short Thin Tubes by External Pressure," *Philosophical Magazine*, July, 1914, pp. 51–56.
(11) Carman, A. P., "The Collapse of Short Thin Tubes," University of Illinois Engineering Experiment Station, Bulletin No. 99, 1917.
(12) Sanden, K. v., und Günther, K., "Ueber das Festigkeitsproblem querversteifter Hohlzylinder unter allseitig gleichmässigem Aussendruck," *Werft und Reederei*, No. 8, 1920, pp. 163–168, No. 9, 1920, pp. 189–198, und No. 10, 1920, pp. 216–221 (von Mises' equation given on p. 220).
(13) Kármán, T. v., "Untersuchungen über Knickfestigkeit," *Mitteilungen über Forschungsarbeiten*, No. 81, 1910.
(14) Sanden, K. v., und Günther, K., "Uber das Festigkeitsproblem querversteifter Hohlzylinder unter allseitig gleichmässigem Aussendruck," II, Teil, *Werft und Reederei*, No. 17, 1921, pp. 505–510.
(15) Fairbairn, W., "The Resistance of Tubes to Collapse," Philosophical Transactions, Vol. 148, 1858, pp. 389–413.
(16) Johow-Foerster, "Hilfsbuch für den Schiffbau," Berlin, 1928, p. 929.

Discussion

[A discussion of this paper jointly with the one by T. McL. Jasper and J. W. W. Sullivan, from R. G. Sturm, will be found in the discussion of the latter paper, immediately following.]

Ernest F. Miller.[4] This appears to be the first attempt to evaluate the effect of circumferential reenforcement of the shell. The writer's interest is its direct application to the design of vacuum apparatus, particularly steam condensers. Until very recently condensers of medium and large sizes have been made of cast iron, and in a large majority of cases, the wall thickness determined by the casting requirements was sufficient to render the problem of collapse to be of little importance.

Many condensers are today being fabricated from steel plate, a process that permits the use of the minimum wall thickness determined by the theory of stability.

It is necessary in the construction of medium and large condensers to provide intermediate support for the cooling tubes. These supports may be utilized to reenforce the shell and to reduce the L/d ratio to small values.

The adaptation of formula [11] to the design of reenforcing rings is a distinct contribution. The degree of support obtained by circumferential bracing in relation to the ideal has heretofore been considered as the relation to a support of infinite rigidity. It is gratifying to find that this is not the case.

The restriction of limiting the total variation in radius to one shell thickness seems to be rather severe for commercial practice. In one design with which the writer is familiar the diameter is 24 ft. and the wall thickness is $^3/_4$ in. The external pressure is one atmosphere. This thickness was determined with an ample factor of safety so that greater discrepancies should be permissible.

There are two problems in the design of condensers that the writer wishes to present for consideration. One problem is the discontinuity of the cylindrical shape at the steam inlet. Here the cylinder becomes tangent to or joins a flat surface. To reduce the possibility of collapse, it has been a practice to reinforce the flat surfaces so that the bending moment due to the pressure loading approaches zero at the junction of the flat and the cylinder. The other problem is that of supporting the weight of a horizontal condenser by feet attached to the shell. No doubt the structure is reenforced more than necessary at these points because of a fear that the radial components of the load will initiate a collapse.

The authors promise that such problems as these may be adequately handled in the near future. It is hoped that this may be a small contribution to a work so well undertaken.

D. S. Jacobus.[5] The authors should be commended for the care and skill with which they conducted their experiments and for presenting rational rules for the design of certain types of vessels under collapsing pressure. Recommendations for the relative strength of the reenforcements, the bringing out of the fact that very stiff reenforcements may weaken the shell, and the specifying of the allowable tolerance in the diameter add much to the knowledge of the subject. The proof that the law of geometric symmetry holds if care is taken to make the proportionate amount of out-of-roundness the same in the model as in the full-sized vessel is a very notable feature, and taken all in all the paper is a most valuable contribution in a field where data are much needed.

The conclusion is reached that, to permit the final selection of a suitable formula or formulas for the strength of vessels under external pressure with correct constants and coefficients, it will necessitate the accumulation of more data than are available at present. As stated in the preamble of the paper, the lack of adequate design formulas led to the formation of the A.S.M.E. Special Research Committee on the Strength of Vessels Subjected to External Pressure. There is a need of more comprehensive formulas of the sort for inclusion in the A.S.M.E. Boiler Code. It is hoped that the committee will be in a position to recommend such formulas without an undue amount of delay.

D. B. Rossheim,[6] F. G. Kleiber,[7] and C. D. Read.[8] The authors are to be complimented on their paper on the stability phase of this problem which finds almost daily application in the design of vacuum and jacketed vessels, boiler flues, etc. With the work of von Sanden, Günther, von Mises, Southwell, and Cook available for many years, it is surprising that so many engineers adhere to old empirical formulas, which while known to be safe within certain limits, usually err greatly on large-diameter vessels with bracing rings spaced at intervals much smaller than the vessel diameter.

This company has designed large towers (21 ft. in diameter)

[4] Development Engineer, Westinghouse Elec. & Mfg. Co., South Philadelphia, Pa. Jun. A.S.M.E.

[5] Advisory Engineer, Babcock & Wilcox Company, New York, N. Y. Mem. A.S.M.E.

[6, 7, 8] M. W. Kellogg Company, Jersey City, N. J.

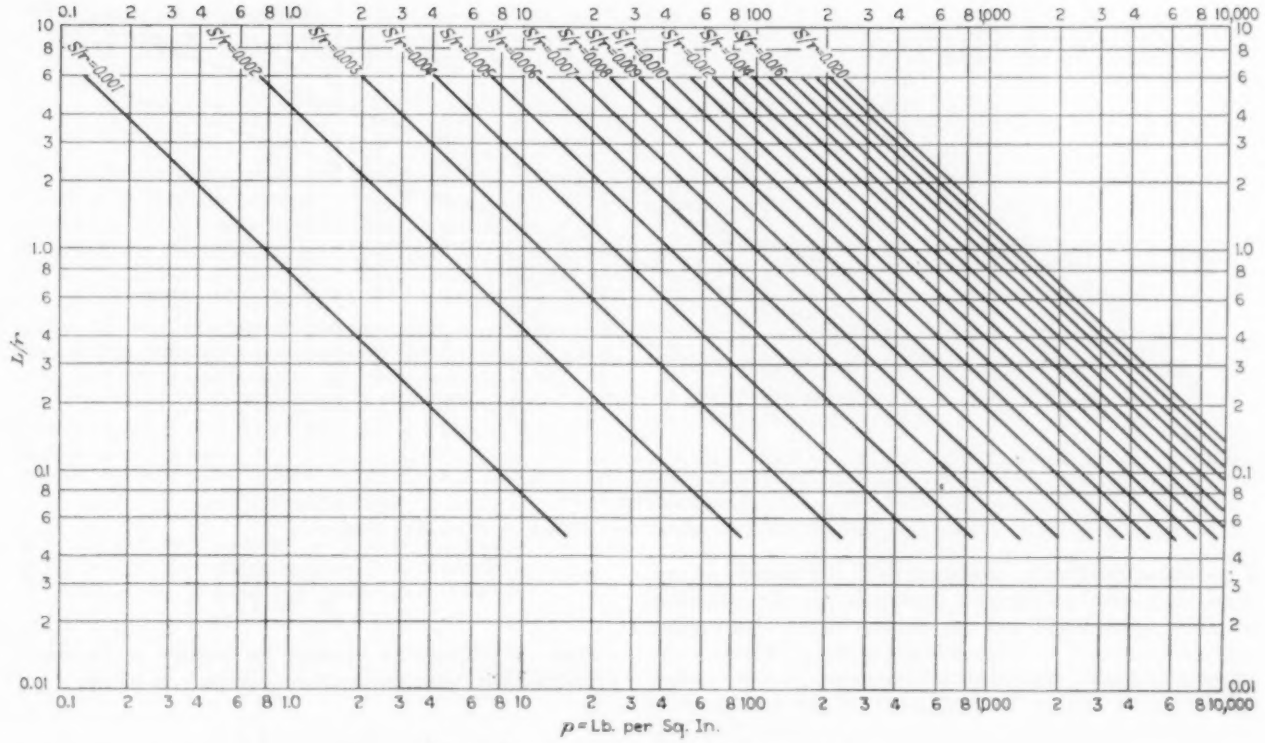


FIG. 8 GRAPH A—SOUTHWELL'S EQUATION [6]

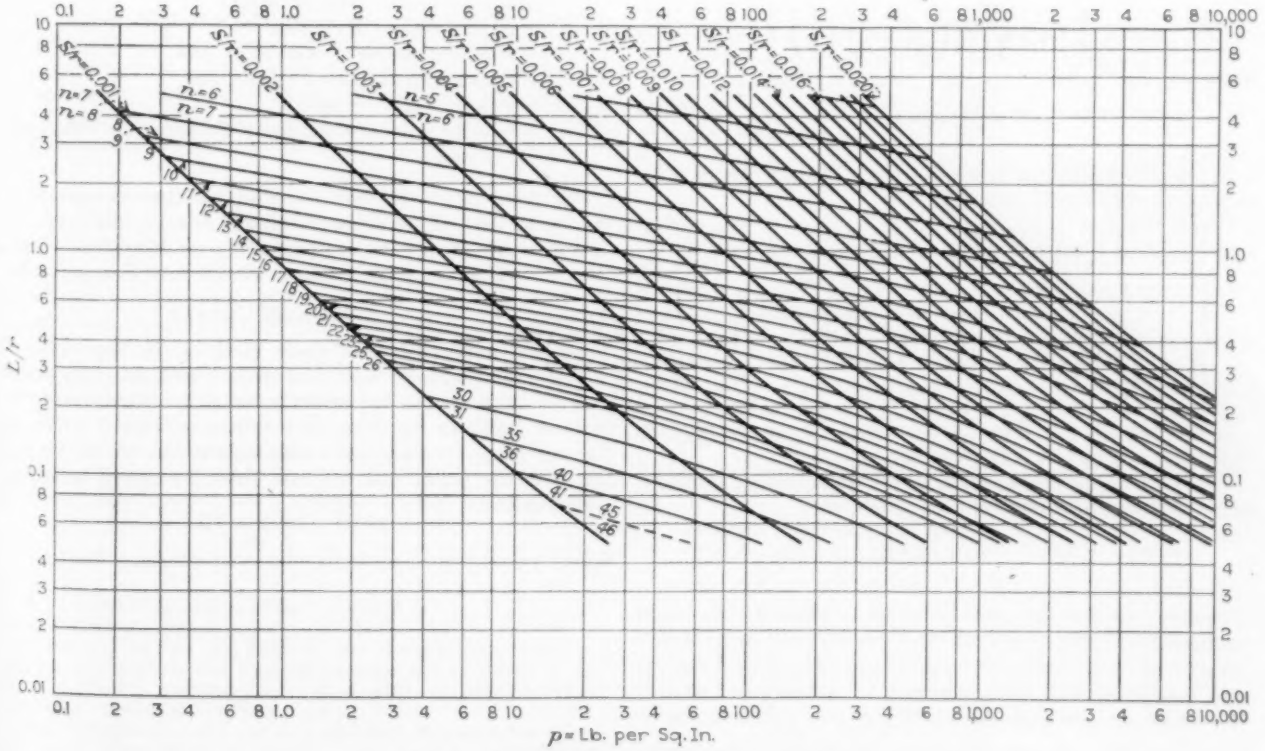


FIG. 9 GRAPH B—VON MISES' EQUATION [10]

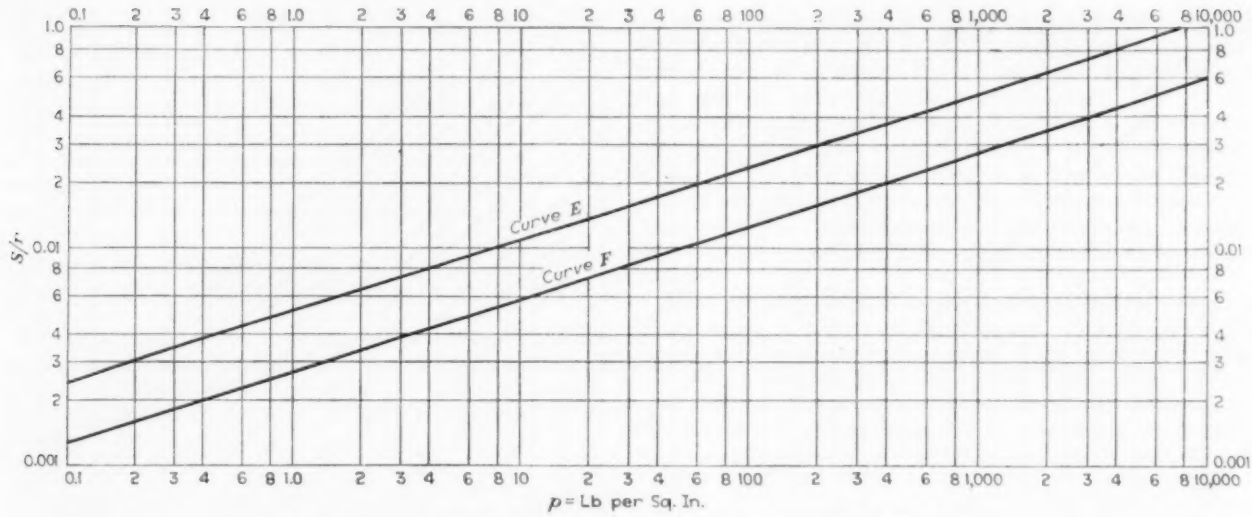


FIG. 10 CURVE E, BRYAN'S EQUATION; CURVE F, STEWART'S EQUATION

of considerable height in accordance with the formula of von Mises, with entire satisfaction. Vessels of this size containing hot inflammable liquid and gas involve considerable expense and hazard, so that to insure absolute safety in operation and economy in design, much time has been spent in investigating and developing data on the many phases of this subject. Also to accelerate calculations and design of such vessels, it has been necessary to compile tables and graphs of the various formulas and equations, which for the most part are somewhat awkward and involve trial and error methods of solution. Three of these curves are presented as Figs. 8, 9, and 10.

Graph A is the approximate Southwell equation for collapse of a short, thin tube under external radial pressure, as follows:

$$p = \frac{32}{9} E \frac{S^2}{Ld} \sqrt[4]{\left[\frac{Z}{36}\left(\frac{m^2}{m^2-1}\right)^3\left(\frac{S}{d}\right)^2\right]}$$

where

$$Z = \frac{\pi^4}{16} \text{ (Co·k)}$$

In this case a family of curves of constant S/r are plotted with an abscissa of p and an ordinate of L/r.

Graph B is von Mises' equation for collapse of thin closed cylinders under equal unit radial and end pressure, as follows:

$$p = \left\{ \frac{\frac{E}{n^2}}{\left[1+\left(\frac{n}{\pi}\frac{L}{r}\right)^2\right]^2}\frac{S}{r} + \frac{n^2}{12}\left[1+\left(\frac{\pi}{n}\frac{r}{L}\right)^2\right]^2 \frac{m^2E}{m^2-1}\left(\frac{S}{r}\right)^3 \right\} \frac{1}{1+\frac{1}{2}\left(\frac{\pi}{n}\frac{r}{L}\right)^2}$$

In this case a family of curves of constant S/r are plotted with an abscissa of p and an ordinate of l/r.

Curve E is the Bryan equation for the collapse of an infinitely long, thin tube under external load, as follows:

$$p = {}^1/_4 \frac{m^2E}{m^2-1}\left(\frac{S}{r}\right)^3$$

Curve F is Stewart's equation for collapse of commercial lap-welded steel tubes under external pressure, as follows:

$$p = 50.2 \times 10^6 \left(\frac{S}{D}\right)^3$$

Nomenclature:

- p = collapsing pressure (lb. per sq. in.)
- E = Young's modulus = (28×10^6)
- S = thickness of cylinder or tube wall
- r = mean radius of cylinder or tube
- d = mean diameter of cylinder
- D = outside diameter of cylinder or tube
- L = length of unsupported ring or tube
- $1/m$ = Poisson's ratio = (0.3)
- n = number of lobes resulting in the minimum collapsing pressure.

The three discussers have compiled the greater part of a treatise on this subject which is believed to cover many phases to which little or no attention has been given heretofore. This will be in shape for presentation in the near future.

AUTHORS' CLOSURE

Additional tests made at the U. S. Experimental Model Basin since the paper was prepared have given results that should be included. These are shown in a new graph for Fig. 2 and in additions to Table 1. [NOTE: The new graph mentioned has been substituted for the former Fig. 2 of the paper, and the new Table 1 is given herewith —EDITOR.]

APM-53-17b

The Collapsing Strength of Steel Tubes

By T. McLEAN JASPER[1] and JOHN W. W. SULLIVAN,[2] MILWAUKEE, WIS.

In this paper the authors present a method for determining the collapsing strength of steel tubes on the basis of the physical and elastic properties of the metal in the tube, and the dimensions, out-of-roundness, and variation in thickness of the tube. This method applies to long, short, thick, or thin tubes. It was developed by considering the collapse of tubes in terms of compressive stress rather than external pressure, by reevaluating the work of other investigators, and from experimental results of the authors. Its application to the determination of the collapsing strength of commercial tubes is shown by illustrative examples.

I—INTRODUCTION

THE object of this paper is to present a method for determining the collapsing strength of steel tubes on the basis of the physical and elastic properties of the steel in the tube, the tube dimensions, and the out-of-roundness and variation in thickness of the tube. The collapsing strength can be determined in terms of (*a*) the maximum compressive stress in the tube walls at the moment of collapse, or (*b*) the external pressure on the tube at the moment of collapse. This method applies to tubes that are sometimes designated as "long," "short," "thick," or "thin."

The physical and elastic properties of the steel used in the construction of tubes which are of importance in determining the collapsing strength of tubes are the yield point, modulus of elasticity, and Poisson's ratio. The tube dimensions which are of importance in determining the collapsing strength consist of the length, diameter, and thickness. The out-of-roundness also affects the pressure necessary to develop collapse.

In order to arrive at a method for determining the collapsing strength of steel tubes, an investigation was made of the factors that influence the collapsing strength of the tubes, and tests were performed in which the effects of these factors were evaluated.

In addition, a study of the experimental work and theoretical considerations of other investigators on the collapsing strength of steel tubes was made.

The method for determining the collapsing strength of steel tubes which is presented in this paper, however, has been derived largely from experimental results. Theoretical considerations have been relied upon only in so far as they have been substantiated by experimental evidence.

The application of this method to the determination of the collapsing strength of commercial tubes is indicated in this paper by illustrative examples.

II—FACTORS THAT AFFECT THE COLLAPSING STRENGTH OF A STEEL TUBE

As a result of the experimental work of the present and previous investigations (1, 2)[3] the following factors have been found to influence the collapsing strength of a steel tube.

- *a* Ratio of length of tube to diameter, L/D
- *b* Ratio of diameter of tube to wall thickness, D/T
- *c* Variations in thickness
- *d* Yield point of the steel in the tube
- *e* Elastic constants of the steel in the tube, represented by the modulus of elasticity and Poisson's ratio
- *f* Variations in roundness
- *g* Method of manufacturing the tube.

In this section the influence of these factors on the collapsing strength of steel tubes will be presented in a general way, and in subsequent sections will be discussed in detail.

As the ratio L/D of length of tube to diameter of tube increases up to about 8, the collapsing strength decreases; above 8 the collapsing strength is practically unaffected by this ratio. For the purposes of this paper, tubes having an L/D ratio of less than 8 will be considered as "short" tubes, and those having an L/D ratio of 8 or over as "long" tubes.

When the ratio of diameter to wall thickness, D/T, increases, the collapsing strength decreases; the actual relationship between this ratio and the collapsing strength being also affected by the yield point of the steel in the tube for the lower D/T ratios. Tubes having lower D/T ratios are sometimes designated "thick" tubes, and those having higher ratios, "thin" tubes. The dividing line between the lower and higher D/T ratios, which separates the thick and thin tubes, is governed by the yield point of the steel in the tubes, as will be subsequently discussed.

Variations in thickness below the average thickness tend to weaken the tube and adversely affect the collapsing strength; whereas variations above the average thickness, under special conditions, may strengthen the pipe and tend to increase the collapsing strength.

The yield point of the steel in the tube has a limiting effect on the collapsing strength for the lower D/T ratios.

The elastic constants of the steel in the tube, represented by the modulus of elasticity and Poisson's ratio, affect the collapsing strength at the higher D/T ratios. But the effect upon the collapsing strength of variations in these constants, for the several types of steel used in tubes, is small.

Variations in the diameter of the tube produce out-of-roundness which tends to lower the collapsing strength.

The method of manufacturing the tube influences the collapsing strength in so far as it affects the roundness and uniformity of thickness of the tube. Methods of manufacturing which tend to increase or decrease the yield point of the steel also have an influence on the collapsing strength of thick tubes.

III—ANALOGY BETWEEN THE COLLAPSE OF A LONG TUBE AND THE FAILURE OF A LONG COLUMN

In this section there will first be discussed the equations of Sturm and Timoshenko for determining the collapsing strength of steel tubes, the curve of Euler for determining the collapsing strength of steel columns, and experimental results on the collapse of tubes and columns. A comparison will then be made of the collapse of a long tube and the failure of a long column on the

---

[1] A. O. Smith Corporation. Mem. A.S.M.E.

[2] A. O. Smith Corporation.

Contributed by the A.S.M.E. Special Research Committee on Strength of Vessels Under External Pressure and presented at the Hartford, Conn., Meeting, June 1 to 3, 1931, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[3] Numbers in parentheses are those of references given in section X of the paper.

basis of the theoretical considerations of Sturm and Euler and the experimental results on tubes and columns.

a Sturm's and Timoshenko's Equations for Determining the Collapsing Strength of Long Tubes

On the basis of the elastic constants and tube dimensions the following equations have been developed by Sturm (3) and Timoshenko (4), respectively, for determining the collapsing strength of long steel tubes.

Sturm's Equation:

$$P = \frac{2}{3} E \frac{t^3}{D^3}\left[\frac{N^4 - N^2 + a^4K^2 + a^2K(2N^2 - m)}{N^2(1 - m^2)}\right] \quad \ldots [A]$$

where P = collapsing pressure, lb. per sq. in.
E = modulus of elasticity of steel, taken as 30,000,000 lb. per sq. in. in this investigation, because it is appropriate to the metal used in the experimental work
t = thickness of tube, inches
D = outside diameter of tube, inches
N = number of lobes in which the tube collapses, taken as 2 for long tubes
$K = \pi^2R^2/L^2$, where R = outside radius of tube, inches, and L = length of tube, inches
a = a parameter for end conditions, taken as 1 for long tubes; and
m = Poisson's ratio, taken as 0.260, which is appropriate for the class of steel used in the tubes.

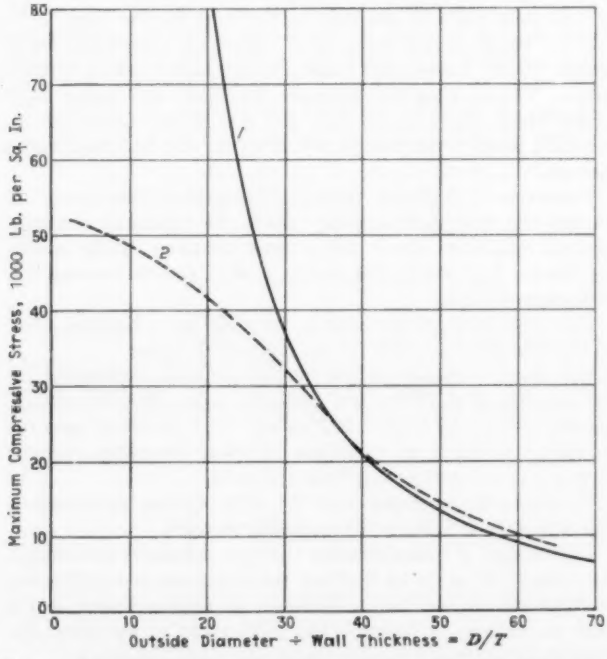


Fig. 1 Curves Showing the Relation Between Maximum Compressive Stress at Collapsing Pressure and D/T, for A. O. Smith Arc-Welded Pipes

1—Calculated from Sturm's equation.
2—Plotted from experimental results.
(Yield point of steel, 52,000 lb. per sq. in.)

For the long tubes tested in the present investigation, this equation reduces to

$$P = 5{,}360{,}000 \frac{t^3}{D^3}(12 + K^2 + 7.74K)$$

where $K = 0.000488\ R^2$.

Timoshenko's Equation:

$$P = \frac{Et^3}{4(1 - m^2)R^3} \quad \ldots [B]$$

where the meanings of the terms are as previously indicated for Sturm's equation. Sturm's and Timoshenko's equations give practically the same values of collapsing pressure for long tubes.

By means of the following Equation [C] the maximum compressive stress in the tube at the collapsing pressure may be determined (5).

$$S = \frac{2PR_1^2}{R_1^2 - R_2^2} \quad \ldots [C]$$

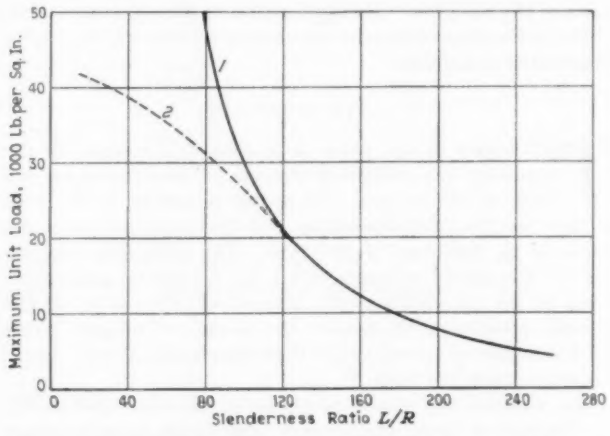


Fig. 2 Relation Between Maximum Unit Load and Slenderness Ratio for Tetmajer's Mild-Steel Columns With Pivoted Ends

1—Calculated from Euler's equation.
2—Plotted from experimental results on Tetmajer's columns.
(Yield point of steel, 39,200 lb. per sq. in.)

where S = maximum compressive stress (at inner surface of tube), lb. per sq. in.
P = collapsing pressure, lb. per sq. in.
R_1 = outer radius of tube, inches, and
R_2 = inner radius of tube, inches.

The theoretical collapsing strength of long steel tubes having various D/T ratios can be obtained from Sturm's Equation [A] and Equation [C]. If the collapsing strengths thus obtained are plotted against the D/T ratios of the corresponding long tubes, a curve results of the form indicated in Fig. 1. This curve will hereinafter be referred to as Sturm's curve. In Fig. 1 there are also plotted the experimental results of one brand of commercial tubes tested in the present investigation. (In Appendix No. 1 there are described the methods and apparatus used in measuring the specimens for out-of-roundness and variations in thickness, and the apparatus and procedure employed in the collapsing tests.)

It is evident in Fig. 1 that Sturm's curve and the curve of experimental results are in agreement for the higher D/T ratios but not for the lower ones. The explanation of this fact is that at the higher D/T ratios the collapsing strength of a steel tube is controlled by one set of factors, whereas at the lower ratios it is controlled by another set, as will be shown later in section IV.

b Euler Curve for Determining the Collapsing Strength of Long Steel Columns Pivoted at Both Ends

In Fig. 2 there is plotted the curve calculated from Euler's

equation for determining the collapsing strength of a long steel column pivoted at both ends. The maximum unit load that may be withstood by the column is plotted against the slenderness ratio; the former being the ratio of total load to cross-sectional area, and the latter being the ratio of length of the column to the least radius of gyration of the cross-section. There are also plotted the experimental results of Tetmajer's mild-steel columns with pivoted ends (6).

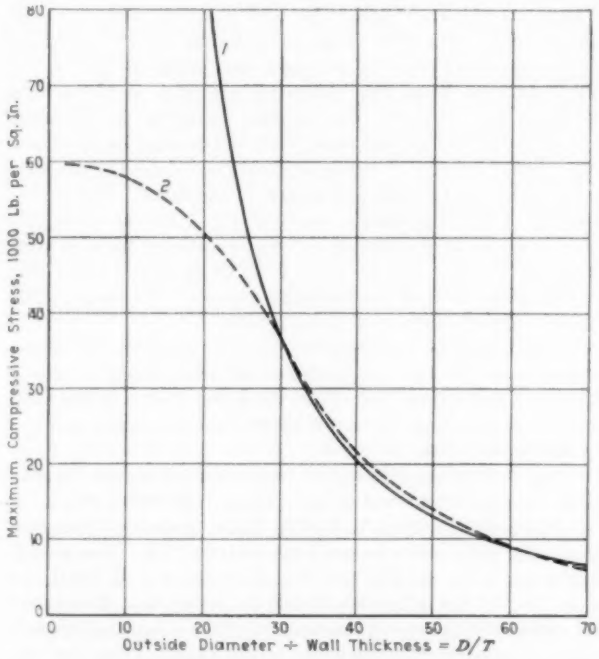


FIG. 3 RELATION BETWEEN MAXIMUM COMPRESSIVE STRESS AT COLLAPSING PRESSURE AND D/T, FOR CARMAN'S SEAMLESS COLD-DRAWN TUBES

1—Calculated from Sturm's equation.
2—Plotted from experimental results.
(Yield point of steel, 59,600 lb. per sq. in.)

It is evident in Fig. 2 that Euler's and Tetmajer's curves are in agreement for the higher slenderness ratios, but not for the lower ones. As in the case of the steel tubes represented in Fig. 1, the collapsing strength of the steel columns is governed by one set of factors at the higher slenderness ratios, while at the lower ratios it is governed by another set.

c COMPARISON OF THE COLLAPSE OF LONG STEEL TUBES AND THE FAILURE OF LONG STEEL COLUMNS PIVOTED AT BOTH ENDS

Only long tubes ($L/D \geq 8$) will be considered in this comparison. The collapse of short tubes will be discussed in a subsequent section. In comparing the collapse of long steel tubes with the failure of long steel columns pivoted at both ends, the ratio D/T of the tubes is analogous to the slenderness ratio of the columns. Hence thin tubes may be compared to columns having higher slenderness ratios, and thick tubes to columns having lower slenderness ratios.

In Figs. 1 and 2 it is evident that the theoretical curves of Sturm and Euler conform to the experimental results on thin tubes and columns having large slenderness ratios. It would therefore appear that the elastic properties of the steel and the ratio D/T control the collapsing strength of thin tubes, and that the elastic properties of the steel and the slenderness ratio L/R control the collapsing strength of columns having large slenderness ratios and which are pivoted at both ends.

For thick tubes and columns having small slenderness ratios, however, the theoretical curves of Sturm and Euler do not conform to the experimental results. In Figs. 1 and 2 it is apparent that the collapsing strengths of thick tubes and columns having small slenderness ratios approach the yield point of the steel as the ratios D/T and L/R decrease. It would therefore appear that the yield point of the steel in these structures controls the collapsing strength. The effect of yield point will be discussed further in the following section.

In the cases of steel columns and thin and thick long steel tubes, failure results from buckling. The slightest degree of eccentricity in the loading of a column lowers the stress at which buckling begins; while in a tube that is subjected to external pressure, out-of-roundness lowers the pressure at which collapse occurs. Also variation in the thickness of a column or tube affects the stress at which failure occurs.

From the foregoing considerations it may be concluded that the collapse of a long steel tube under external pressure is analogous to the failure of a long steel column under a compressive load.

The analogy between the collapse of a tube and the failure of a column is discussed because the relationship between the theoretical considerations and experimental evidence is so well established for column failure.

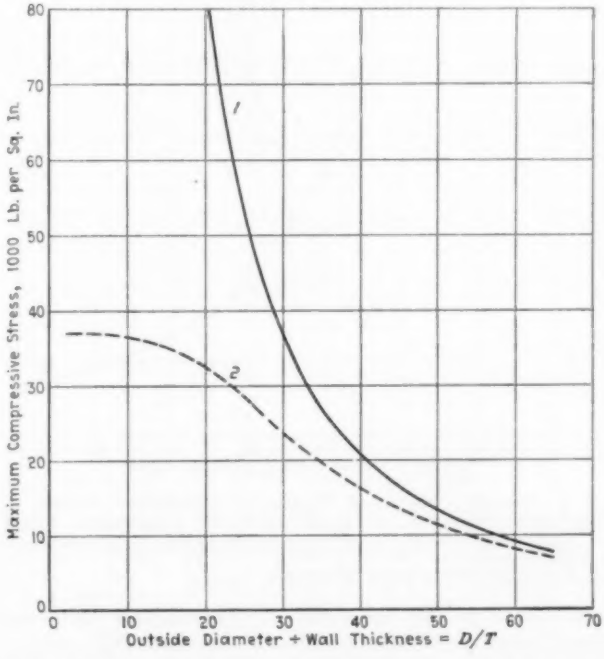


FIG. 4 RELATION BETWEEN MAXIMUM COMPRESSIVE STRESS AT COLLAPSING PRESSURE AND D/T, FOR STEWART'S LAP-WELDED PIPES

1—Calculated from Sturm's equation.
2—Plotted from experimental results.
(Yield point of steel, 37,000 lb. per sq. in.)

IV—FUNDAMENTAL RELATIONSHIP BETWEEN (*a*) THE COLLAPSING STRENGTH OF A LONG STEEL TUBE AND (*b*) THE RATIO D/T, THE ELASTIC CONSTANTS, AND THE YIELD POINT OF THE STEEL

This fundamental relationship is based upon the experimental work of three independent investigations by Carman (2), Stewart (1), and the A. O. Smith Corporation on the collapsing strength of long steel tubes, that is, tubes having an L/D ratio of 8 or over.

a Carman's Experimental Results

In Fig. 3 there are plotted results of Carman's tests on 21 cold-drawn seamless steel tubes. These tubes were 1 in. to 3 in. in diameter, 0.014 in. to 0.065 in. thick, and 7.6 to 14.1 diameters long. The D/T ratios of these tubes ranged from 42.8 to 68.7. The dimension ratios and collapsing strengths of these tubes are listed in Table 1.[4] These tubes were carefully selected and machined to the desired dimensions, and consequently the out-of-roundness and variation in thickness were exceedingly small. The average yield point of the steel in these tubes was 59,600 lb. per sq. in.

The curve of experimental results in Fig. 3 is drawn through the experimental points at the higher D/T ratios and extrapolated back to the yield point through the lower D/T ratios.

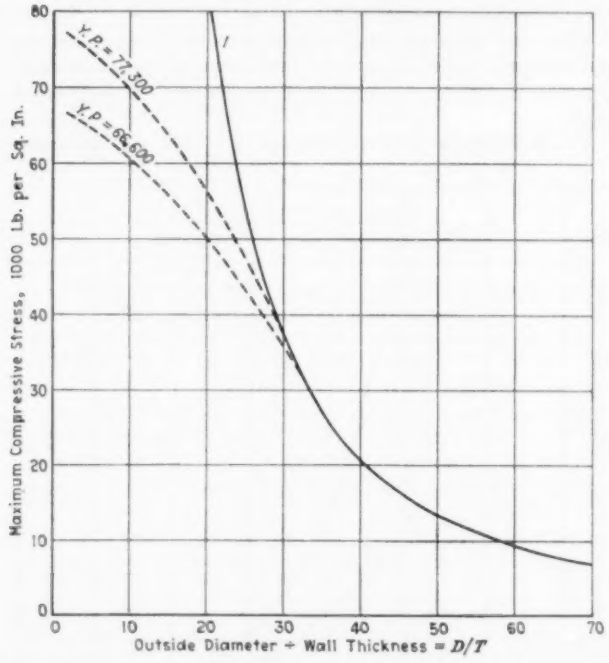


Fig. 5 Relation Between Maximum Compressive Stress at Collapsing Pressure and Ratio D/T, for Commercial Tubes

Note: Tubes are 8 or more diameters in length and of uniform roundness and thickness.
1—Calculated from Sturm's equation.

It is evident in this figure that the experimental curve and Sturm's curve are in close agreement at the higher D/T ratios.

b Stewart's Experimental Results

In Fig. 4 there are plotted the results of Stewart's tests on 256 lap-welded steel tubes. These tubes were 3 in. to 13 in. in diameter, 0.11 in. to 0.51 in. thick, and over 8 diameters long. The D/T ratios ranged from 12.3 to 63.0. The average yield point of the steel was 37,000 lb. per sq. in. The dimensions and collapsing strengths of these tubes are listed in Table 2. The average variation in thickness $[(T_{avg.} - T_{min.}) \times 100/T_{avg.}]$ was 12 per cent, while the average out-of-roundness $[(D_{max.} - D_{min.}) \times 100/D_{min.}]$ was 1.4 per cent. These variations were based on measurements made at the collapsed section of the tubes.

The curve of experimental results in Fig. 4 does not conform to Sturm's curve so closely as does the experimental curve in Fig. 3, for the higher D/T ratios. At the lower D/T ratios it is evident that the curve in Fig. 4 approaches the yield point of the steel in the tubes. The lack of agreement between Sturm's curve and the experimental curve in Fig. 4 is due to the out-of-roundness and variation in thickness of the tubes. The quantitative effect of out-of-roundness in reducing the collapsing strength will be discussed in a subsequent section.

[4] See Appendix No. 3 for tables.

c A. O. Smith Corporation's Experimental Results

In Fig. 1 there are plotted the results of A. O. Smith Corporation's tests on 14 arc-welded tubes. These tubes were $8^5/_8$ in. to 16 in. in diameter, 0.25 in. to 0.38 in. thick, and over 8 diameters long. The D/T ratios ranged from 34.3 to 61.5, and the average yield point of the steel was 52,000 lb. per sq. in. The dimension ratios and collapsing strengths of these tubes are listed in Table 3. The average variation in thickness[5] of these tubes was 2.5 per cent, while the average out-of-roundness was 0.77 per cent. These variations were based on measurements made at the collapsed section of the tubes. The general direction of the experimental curve in Fig. 1 for the lower D/T ratios was further established by eight additional tests on tubes having D/T ratios of 24.4 to 25.9. The results of these tests are not classified with the foregoing group of 14 because of the slightly different method of manufacture.

As mentioned in the foregoing section, the curve of experimental results in Fig. 1 is in agreement with Sturm's curve at the higher D/T ratios, but not at the lower ones. At the lower ratios, the experimental curve approaches the yield point of the steel in the tubes.

In Fig. 5 there are plotted the results of A. O. Smith Corporation's tests on 4 seamless tubes. These tubes were $8^5/_8$ in. to 9 in. in diameter, 0.40 in. to 0.50 in. thick, and over 8 diameters long. The D/T ratios ranged from 18.0 to 21.5. The average yield point of the steel in two of these tubes was 77,300 lb. per sq. in., and in the other two, 66,600 lb. per sq. in. The dimension ratios and collapsing strengths of these tubes are shown in Table 4. The average variation in thickness of the first two was 4.8 per cent, and of the second two, 11 per cent, while the average out-of-roundness of the first two was 0.23 per cent and of the second two, 0.27 per cent.

The experimental results of these tubes at the lower D/T ratios lie in a curve drawn similarly to the experimental curves in Figs. 1 and 3, thus indicating that at the lower D/T ratios the experimental curve approaches the yield point of the steel in the tubes.

d Fundamental Relationship Between the Collapsing Strength of Long Steel Tubes and the Factors Controlling This Strength

From a consideration of Sturm's equation and of the experimental curves in Figs. 1 and 3, it is evident that Sturm's curve approximately defines the collapsing strength of long steel tubes at the higher D/T ratios. As Sturm's curve is based upon the elastic properties of the steel and the dimensions of the tube, it follows that the elastic constants and the D/T ratio are the controlling factors in determining the collapsing strength of tubes having the higher D/T ratios.

The elastic constants are represented by the modulus of elasticity and Poisson's ratio. The effect upon the collapsing strength of variations in these elastic constants, for the several types of steel used in tubes, is small. The controlling factors in determining the collapsing strength of long steel tubes having the higher D/T ratios therefore reduce to merely the ratio D/T itself.

Since the maximum stress in the tube at the collapsing pressure varies directly as the collapsing pressure for a given ratio

[5] The variation in thickness was measured at the collapsed section of the tube, and is equal to $[(T_{avg.} - T_{min.}) \times 100/T_{avg.}]$.

D/T, it follows that the ratio D/T controls the collapsing pressure of tubes having the higher D/T ratios.

Referring to Figs. 1, 3, 4, and 5, it is evident that the maximum stress in the tubes at the collapsing pressure approaches the yield point of the steel in the tubes, as the ratio D/T decreases. It also appears that the higher the yield point the greater the maximum stress, at the lower D/T ratios. Thus, for a ratio $D/T = 25$, the yield point and maximum stress of the tubes represented in Figs. 1, 3, 4, and 5 are set forth in the order of increasing maximum stress in Table 5. The effect of the yield point on the maximum stress becomes more pronounced as the ratio D/T decreases.

On the basis of Table 5 and the curves in Figs. 1, 3, 4, and 5, it appears that for steel tubes having the lower D/T ratios, increasing the yield point raises the maximum stress that may be withstood by the tubes under external pressure. Since the maximum stress varies directly with the collapsing pressure for a given ratio D/T, it follows that for long tubes having the lower D/T ratios, the yield point of the steel and the ratio D/T are the controlling factors in determining the collapsing pressure of the tubes. In developing the foregoing relationships, the following assumptions have been made:

1 The tubes consist of homogeneous steel, and are uniformly round and uniformly thick
2 The yield point of the steel in compression is assumed to be equal to the yield point in tension
3 For very low values of the ratio D/T, the collapsing strength approaches the ultimate strength of the steel in the tube.

The fundamental relationship between (*a*) the collapsing strength of a long steel tube and (*b*) the D/T ratio, the elastic constants, and the yield point of the steel, for tubes of steel having various yield points, is set forth in Fig. 6.

From the foregoing discussion it is concluded that the controlling factors in determining the collapsing strength of long steel tubes are: (1) the elastic constants and the ratio D/T for tubes having the higher D/T ratios; and (2) the yield point of the steel in the tube and the ratio D/T for tubes having the lower D/T ratios.

V—FUNDAMENTAL RELATIONSHIP BETWEEN (*a*) THE COLLAPSING STRENGTH OF A SHORT STEEL TUBE AND (*b*) THE RATIOS L/D AND D/T, THE ELASTIC CONSTANTS, AND THE YIELD POINT OF THE STEEL

In the foregoing section the fundamental relationship between the collapsing strength of a long steel tube and the factors that control this strength was presented. In this section the effect of the ratio L/D of length of tube to diameter as an additional factor in controlling the collapsing strength will be considered. It has been shown that for steel tubes having a ratio L/D of 8 or more the collapsing strength is practically unaffected by this ratio (1) (2). This section will therefore concern tubes having a ratio L/D of less than 8.

No tests on short tubes were performed by A. O. Smith Corporation, as the results of an investigation on short tubes by Carman provided sufficient data to enable the effect of the ratio L/D on the collapsing strength to be determined. Carman's work on short tubes was selected as a basis for determining the effect of the ratio L/D on account of the great care exercised in preparing the specimens and in performing the collapsing tests.

Before discussing the effect of the ratio L/D on the collapsing strength of short tubes, a review will be made of the factors that affect the collapsing strength of such tubes in order that Carman's data may be properly evaluated. These factors are as follows:

1 Ratio of length of tube to diameter of tube, L/D
2 Ratio of diameter of tube to wall thickness, D/T
3 Variations in the thickness of the tube
4 Yield point of the steel in the tube
5 Elastic constants of the steel in the tube, represented by the modulus of elasticity and Poisson's ratio
6 Variations in the diameter of the tube, and
7 Method of manufacturing the tube.

On account of the care with which Carman had his specimens machined, the effect of variations in the diameter and thickness on the collapsing strength of the tube may be considered as negligible. Since the effect of variations in the elastic constants of commercial steels on the collapsing strength of steel tubes is

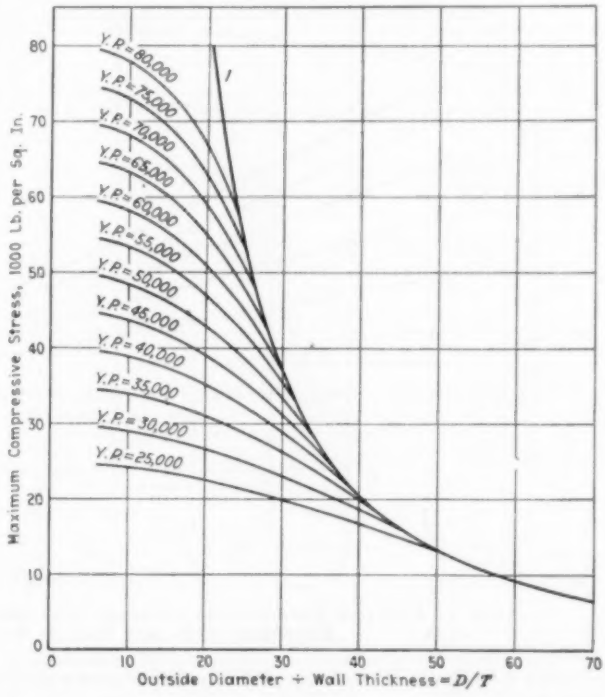


FIG. 6 RELATION BETWEEN MAXIMUM COMPRESSIVE STRESS AT COLLAPSING PRESSURE AND RATIO D/T, FOR HOMOGENEOUS STEEL PIPE HAVING VARIOUS YIELD-POINT VALUES

NOTE: Pipe is 8 or more diameters in length and of uniform roundness and thickness.
1—Calculated from Sturm's equation.

small, the fifth factor may be considered as constant. All of Carman's steel tubes were made in the same way and from the same quality of material; consequently the effect of the method of manufacture may also be considered as constant. Accordingly, there remain for consideration the effects of only three factors on the collapsing strength of Carman's short steel tubes, namely, the ratio of length of tube to diameter, L/D, the ratio of diameter to thickness, D/T, and the yield point of the steel in the tube.

a APPLICATION OF CARMAN'S DATA TO STURM'S EQUATION

The effect of the ratio L/D on the collapsing strength of short tubes was investigated by applying Carman's experimental data to Sturm's equation for determining the collapsing strength of steel tubes. These data cover tubes having an L/D ratio of 1.74 to 6.09. In order to determine the effect of the L/D ratio on the collapsing strength of tubes on the border line between short tubes and long tubes, additional data were used on

tubes having an L/D ratio of 7.60 to 8.08. These data are set forth in Table 6.

In applying Sturm's equation to short tubes, the term N (the number of lobes into which the tube collapses) is the only unknown variable. It is entirely logical that N should be represented by either a whole or compound number. In order to evaluate this term, the foregoing data on the collapsing pressure of short tubes were substituted in Sturm's equation, and N was determined for several L/D ratios. The results of these determinations were plotted in a series of intermediate curves, of which Fig. 7 represents the final result. In Fig. 7 the value of N is plotted against the ratio L/D for various D/T ratios. On the basis of the value of N shown in Fig. 7, the collapsing strengths of short tubes having L/D ratios of $1^1/_2$ to 7 were calculated. The results of these calculations are shown in Table 7.

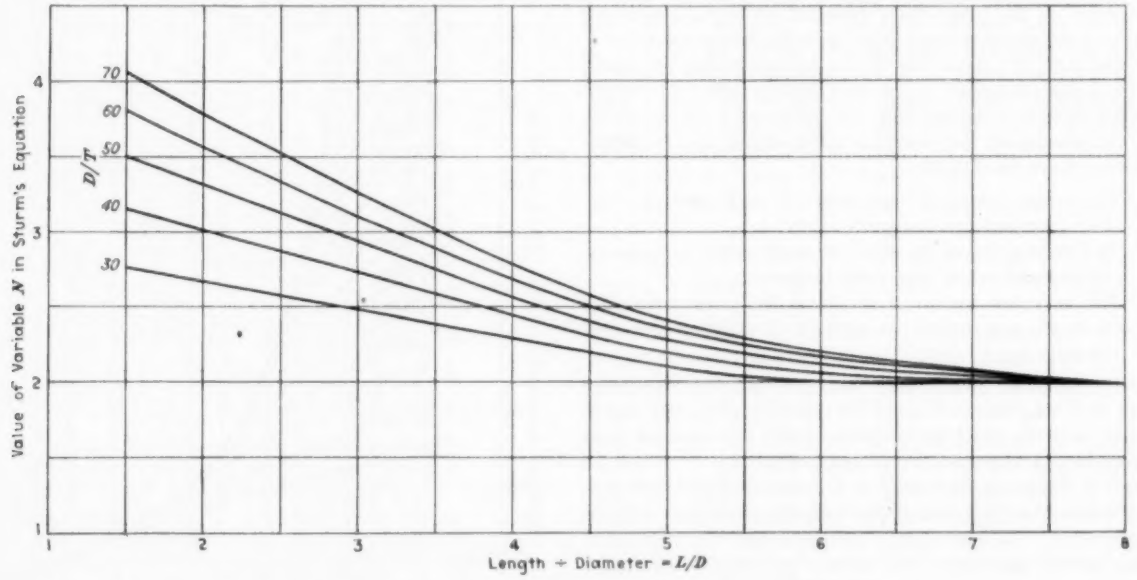


FIG. 7 RELATION BETWEEN THE VARIABLE N IN STURM'S EQUATION AND THE RATIO OF LENGTH OF TUBE TO DIAMETER (L/D), FOR VARIOUS RATIOS OF DIAMETER TO THICKNESS (D/T)

b FUNDAMENTAL RELATIONSHIP BETWEEN THE COLLAPSING STRENGTH OF SHORT STEEL TUBES AND THE FACTORS THAT CONTROL THIS STRENGTH

On the basis of the determination of the collapsing strength of short steel tubes obtained by substituting the proper value of N in Sturm's equation, and on the basis of the experimental data on short tubes, the collapsing strengths of short steel tubes were plotted against the D/T ratios for various L/D ratios, as shown in Fig. 8. The collapsing strength was determined by solving Sturm's equation for collapsing pressure, and then substituting this value of pressure in the following formula.

$$S = \frac{PD}{2T} \quad \text{[D]}$$

where S = maximum compressive stress in the tube, lb. per sq. in.
P = collapsing pressure of the tube, lb. per sq. in.
D = outside diameter of the tube, inches
T = thickness of the tube, inches.

NOTE: The simplified formula [D] may be used in place of the fundamental formula [C], given in section III, for D/T ratios of 32 or more without incurring an error greater than 3 per cent.

From Fig. 8 it is concluded that the collapsing strength of short steel tubes is controlled by the elastic properties and the L/D and D/T ratios for the higher D/T ratios; and by the yield point of the steel and the L/D and D/T ratios for the lower D/T ratios.

Since the collapsing pressure varies directly with the collapsing strength, it follows that these relationships also apply to the collapsing pressure.

In order that the information contained in Fig. 9 may be used more readily, the curves have been replotted for the individual L/D ratios, in Figs. 9 to 15.

As in the case of the long tubes discussed in the previous section, the following assumptions apply to the fundamental relationship between the collapsing strength of short steel tubes and the factors that control this strength, namely:

1 The tubes are made of homogeneous steel, and are uniformly round and uniformly thick.

2 The yield point of the steel in compression is assumed to be equal to the yield point in tension.

3 For very low values of the ratio D/T, the collapsing strength approaches the ultimate strength of the steel in the tube.

c EFFECT OF ENDS ON THE COLLAPSING STRENGTH OF SHORT STEEL TUBES

The parameter a in Sturm's equation concerns the end conditions of the tube. The experimental data used in developing the effect of the ratio L/D on the collapsing strength of short tubes were derived from simply supported tubes, for which $a = 1$.

In determining the collapsing strength of short tubes with ends, the value of a might conceivably be increased somewhat. It is suggested that further tests might be made which would indicate the value of a to be used for vessels having ends of a certain degree of fixity.

However, it is considered that designs based on the curves shown in Fig. 8 will always be on the side of safety.

VI—EFFECT OF OUT-OF-ROUNDNESS ON THE COLLAPSING STRENGTH OF STEEL TUBES

The experimental data obtained from A. O. Smith Corporation's tests on tubes did not reveal the effect of out-of-roundness on the collapsing strength of the tubes. The degree of out-of-

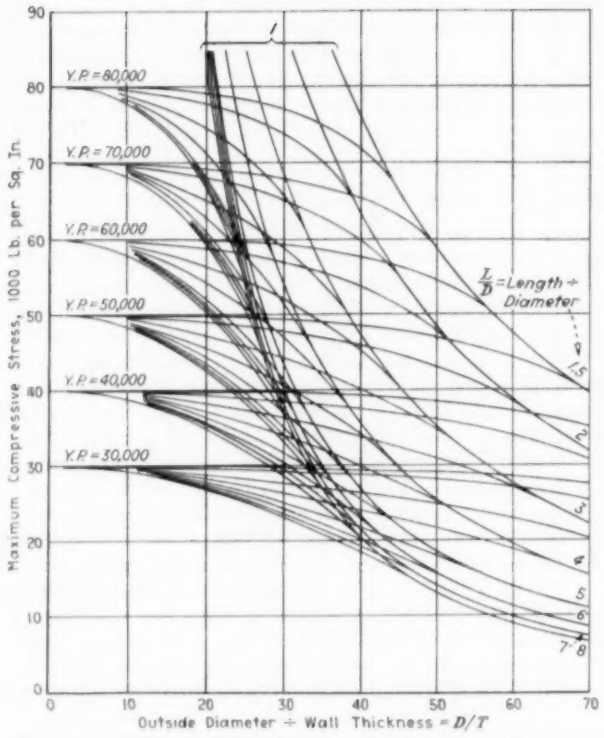


FIG. 8 RELATION BETWEEN MAXIMUM COMPRESSIVE STRESS AT COLLAPSING PRESSURE AND D/T, FOR HOMOGENEOUS STEEL TUBES HAVING VARIOUS YIELD-POINT VALUES AND VARIOUS L/D RATIOS

NOTE: Tubes are of uniform roundness and thickness.
1—Calculated from Sturm's equation.

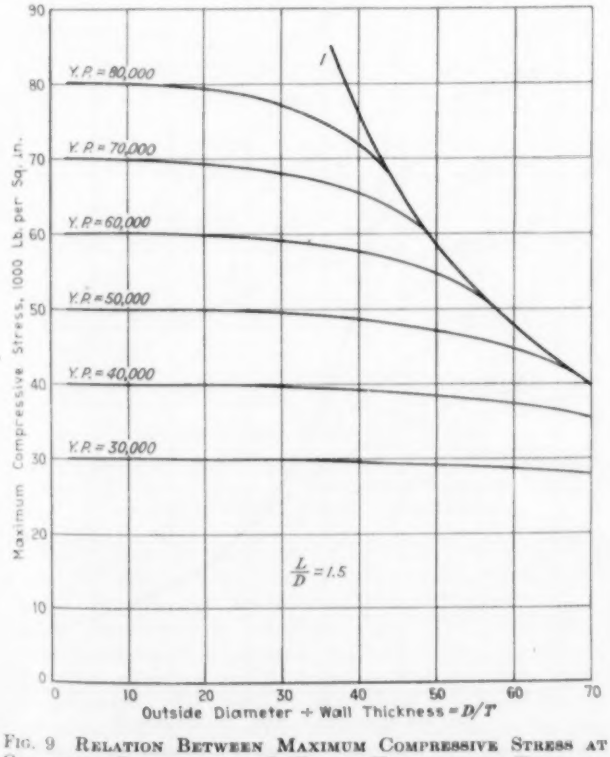


FIG. 9 RELATION BETWEEN MAXIMUM COMPRESSIVE STRESS AT COLLAPSING PRESSURE AND D/T, FOR HOMOGENEOUS TUBES OF STEEL HAVING VARIOUS YIELD-POINT VALUES

NOTE: Tubes are 1.5 diameters in length and of uniform roundness and thickness.
1—Calculated from Sturm's equation.

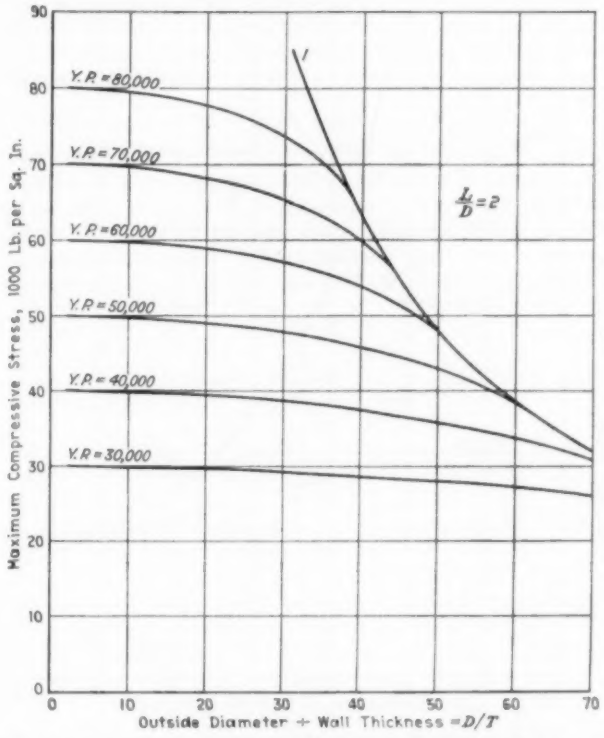


FIG. 10 RELATION BETWEEN MAXIMUM COMPRESSIVE STRESS AT COLLAPSING PRESSURE AND D/T, FOR HOMOGENEOUS TUBES OF STEEL HAVING VARIOUS YIELD-POINT VALUES

NOTE: Tubes are 2 diameters in length and of uniform roundness and thickness.
1—Calculated from Sturm's equation.

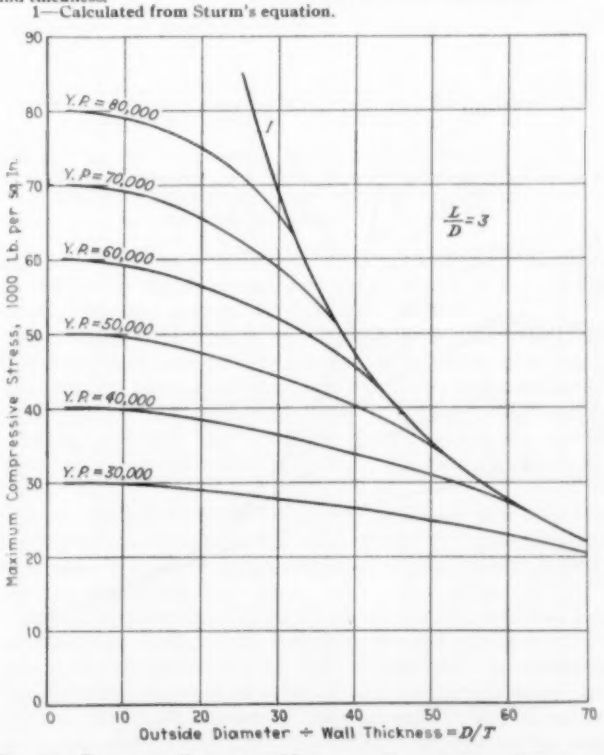


FIG. 11 RELATION BETWEEN MAXIMUM COMPRESSIVE STRESS AT COLLAPSING PRESSURE AND D/T, FOR HOMOGENEOUS TUBES OF STEEL HAVING VARIOUS YIELD-POINT VALUES

NOTE: Tubes are 3 diameters in length and of uniform roundness and thickness.
1—Calculated from Sturm's equation.

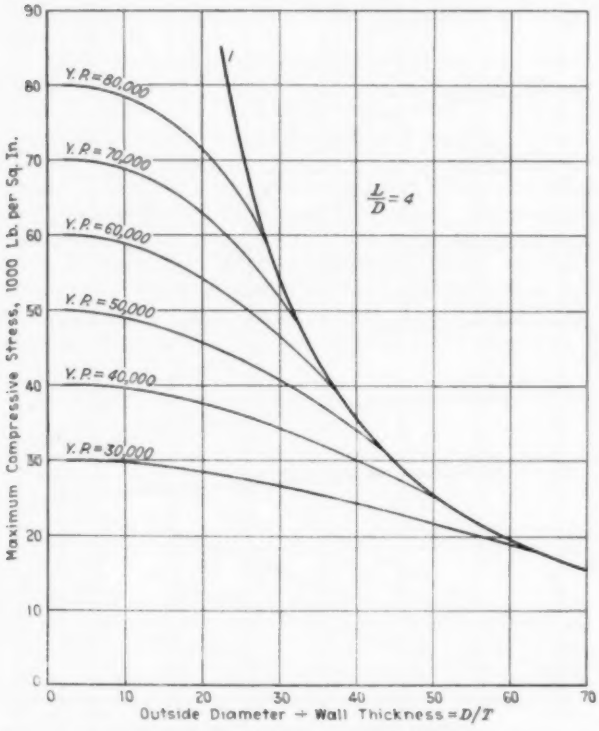


FIG. 12 RELATION BETWEEN MAXIMUM COMPRESSIVE STRESS AT COLLAPSING PRESSURE AND D/T, FOR HOMOGENEOUS TUBES OF STEEL HAVING VARIOUS YIELD-POINT VALUES

NOTE: Tubes are 4 diameters in length and of uniform roundness and thickness.
1—Calculated from Sturm's equation.

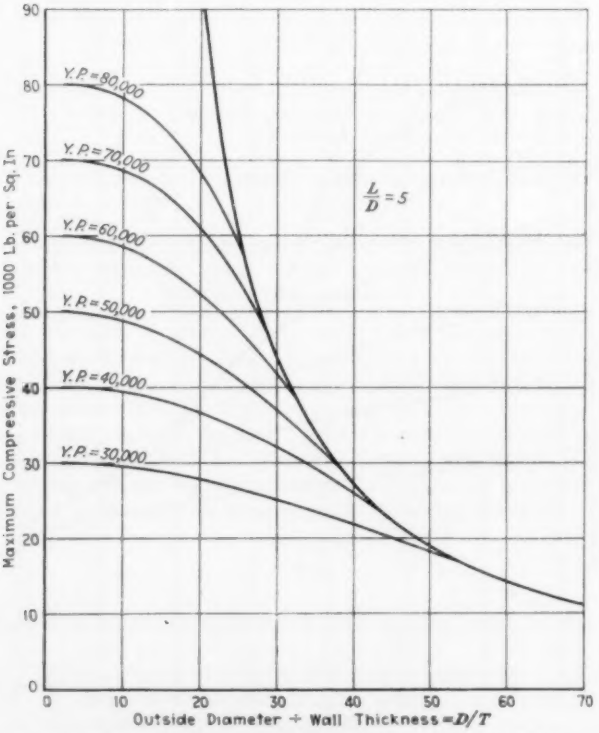


FIG. 13 RELATION BETWEEN MAXIMUM COMPRESSIVE STRESS AT COLLAPSING PRESSURE AND D/T, FOR HOMOGENEOUS TUBES OF STEEL HAVING VARIOUS YIELD-POINT VALUES

NOTE: Tubes are 5 diameters in length and of uniform roundness and thickness.
1—Calculated from Sturm's equation.

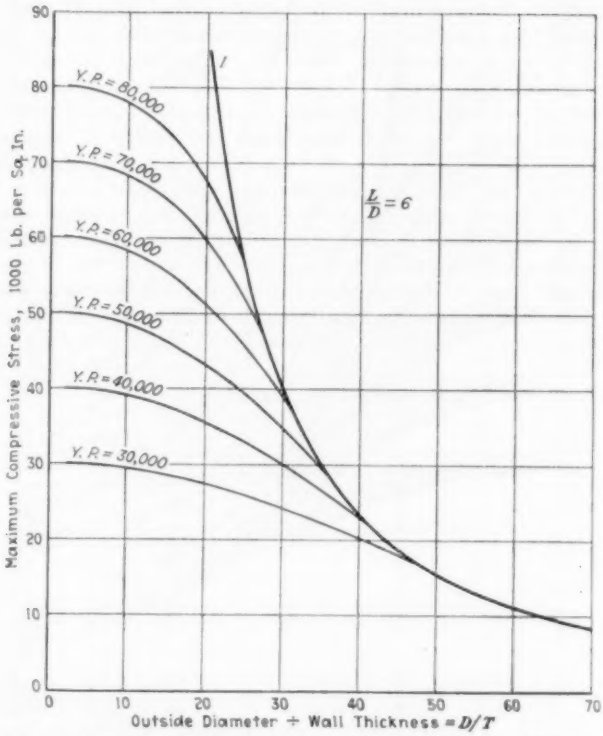


FIG. 14 RELATION BETWEEN MAXIMUM COMPRESSIVE STRESS AT COLLAPSING PRESSURE AND D/T, FOR HOMOGENEOUS TUBES OF STEEL HAVING VARIOUS YIELD-POINT VALUES

NOTE: Tubes are 6 diameters in length and of uniform roundness and thickness.
1—Calculated from Sturm's equation.

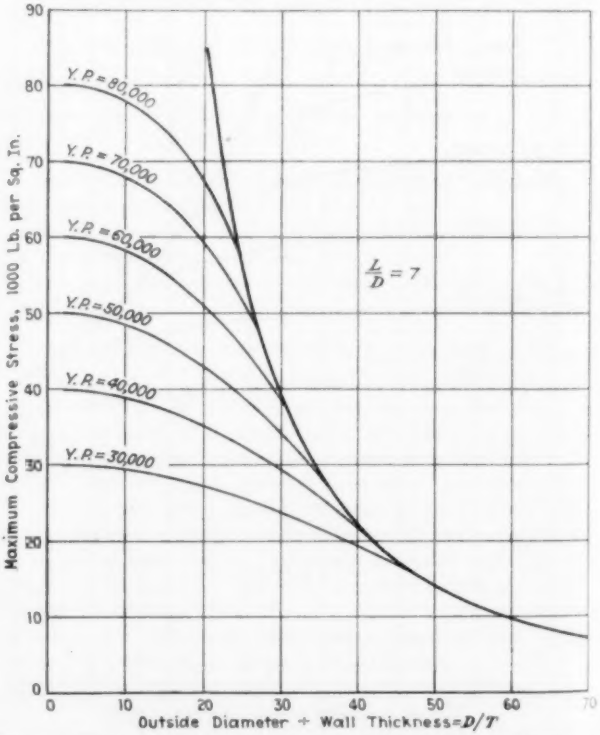


FIG. 15 RELATION BETWEEN MAXIMUM COMPRESSIVE STRESS AT COLLAPSING PRESSURE AND D/T, FOR HOMOGENEOUS TUBES OF STEEL HAVING VARIOUS YIELD-POINT VALUES

NOTE: Tubes are 7 diameters in length and of uniform roundness and thickness.
1—Calculated from Sturm's equation.

roundness in some of the tubes was too small to be evaluated; and in others the effect of out-of-roundness appeared to be repressed by the method of manufacturing the tube. Consequently Stewart's data (1) on the collapse of long tubes were selected to reveal the effect of out-of-roundness on the collapsing strength.

a Effect of Out-of-Roundness in Reducing the Collapsing Strength of Steel Tubes, as Indicated by Theoretical Considerations

An investigation of the effect of out-of-roundness on the basis of theoretical considerations was first made, which later served as a guide for interpreting Stewart's data. The effect of out-of-roundness as indicated by theoretical considerations, however, did not conform quantitatively to the experimental evidence. It did, however, conform qualitatively to the experimental evidence, in so far as it indicated that the effect of out-of-roundness in reducing the collapsing strength of steel tubes increases as the ratio D/T of diameter to thickness increases.

Because of the lack of quantitative agreement between the effect of out-of-roundness based upon theoretical considerations and that indicated by the experimental evidence, the latter was used to evaluate the effect of out-of-roundness in reducing the collapsing strength of steel tubes.

A study of the effect of out-of-roundness based upon theoretical considerations was made by means of Timoshenko's analysis of an elliptical ring submitted to uniform pressure (7). The results of this calculation are presented in Fig. 16.

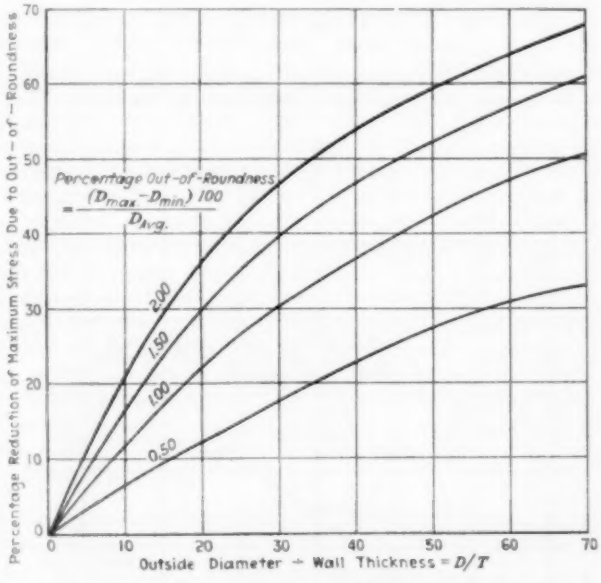


Fig. 16 Showing Effect of Out-of-Roundness in Reducing Maximum Stress in Tube at Collapsing Pressure, as Indicated by Theoretical Considerations

In this figure the percentage reduction of stress in the tube at the collapsing pressure (due to out-of-roundness) is plotted against the ratio of outside diameter to wall thickness, D/T, for various percentages of out-of-roundness. The percentage of out-of-roundness is determined by dividing the difference between the maximum and minimum outside diameters by the average outside diameter and multiplying this result by 100.

It is evident that the effect of out-of-roundness in reducing the collapsing pressure (as derived from the foregoing theoretical considerations) increases as the ratio D/T increases (see Fig. 16). It should be noted, however, as will subsequently be shown, that as the ratio D/T increases and the collapsing strength becomes controlled by the elastic properties of the tube rather than by the yield point of the steel in the tube, the effect of out-of-roundness decreases.

b Application of Stewart's Data to the Determination of the Effect of Out-of-Roundness in Long Steel Tubes

Stewart's data are not ideally suited to an investigation on the effect of out-of-roundness on account of wide variations in the thickness of his specimens. The difference between the collapsing strength of his tubes and that of perfect tubes consequently represents the effects of variations in both roundness and thickness. These data were not sufficiently complete to enable the effects of variations in thickness and roundness on the collapsing strength to be readily separated.

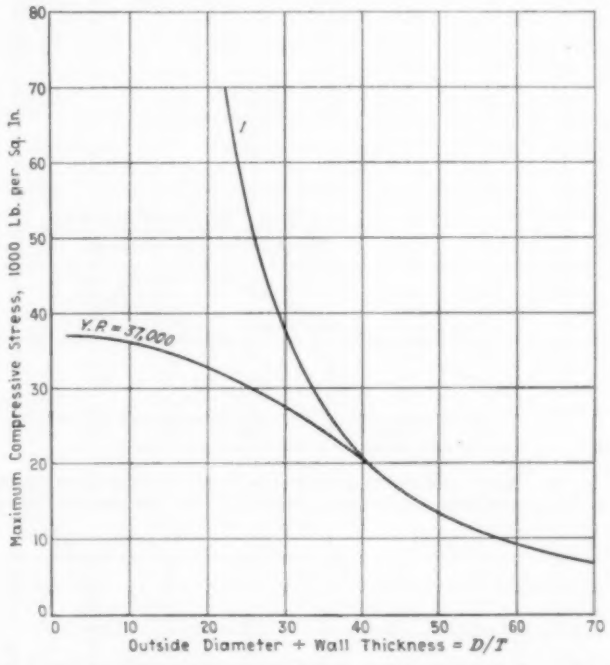


Fig. 17 Relation Between Maximum Compressive Stress at Collapsing Pressure and D/T, for Homogeneous Tubes of Steel Having a Yield Point of 37,000 Lb. per Sq. In.
1—Calculated from Sturm's equation.

However, on the basis of calculations made with these data, it is clearly indicated that the effect of the variations in thickness prevalent in these specimens is in large measure subordinate to the effect of variations in roundness in reducing the collapsing strength, and that the out-of-roundness has a much greater effect than the variation in thickness prevalent in Stewart's tubes.

Since the effects of variations in roundness and thickness are difficult to separate, the total effects of both variations were considered to be a safe evaluation of the maximum effect of out-of-roundness.

The determination of the effect of out-of-roundness presupposes a knowledge of the collapsing strength of perfect tubes, that is, homogeneous tubes of uniform roundness and thickness. The collapsing strength of tubes of steel having the same yield point as the steel in Stewart's tubes, namely, 37,000 lb. per sq. in., is indicated in Fig. 17. This curve is derived on the basis of the fundamental relationship between (*a*) the collapsing strength of long steel tubes and (*b*) the ratio D/T, the elastic constants, and the yield point of the steel, which relationship

is set forth in Fig. 6. In Fig. 17 the maximum compressive stress in the tube at the collapsing pressure is plotted against the ratio D/T of diameter to thickness. As discussed in a previous section, at the lower D/T ratios, the curve marked "Y. P. = 37,000" determines the collapsing strength; while at the higher D/T ratios the curve calculated from Sturm's equation determines the collapsing strength.

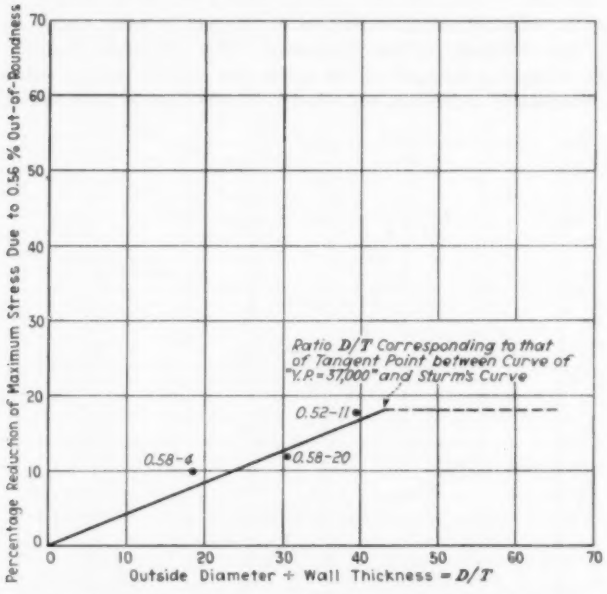


FIG. 18 SHOWING EFFECT OF 0.56 PER CENT OUT-OF-ROUNDNESS IN REDUCING THE COLLAPSING STRENGTH OF STEWART'S TUBES

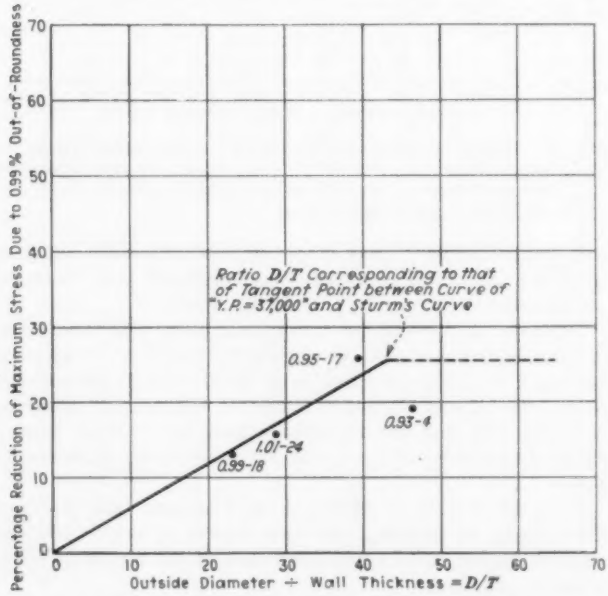


FIG. 19 SHOWING EFFECT OF 0.99 PER CENT OUT-OF-ROUNDNESS IN REDUCING THE COLLAPSING STRENGTH OF STEWART'S TUBES

The maximum stresses in Stewart's tubes were calculated by means of formula [D], Section V, and are set forth in Table 8. The maximum stresses in perfect tubes, having the same D/T ratio as Stewart's tubes, were read off the proper curve in Fig. 17, and are listed in Table 8.

The percentage of out-of-roundness in Stewart's tubes varied from less than 0.25 to more than 2.00. In order to evaluate the effect of this range of out-of-roundness the data were classified into zones of the following percentages, as shown in Table 9: 0.25–0.75, 0.75–1.25, 1.25–1.75, and 1.75–2.25. The average effect of each of these zones in reducing the collapsing strength was plotted in a series of intermediate curves shown in Figs. 18–21.

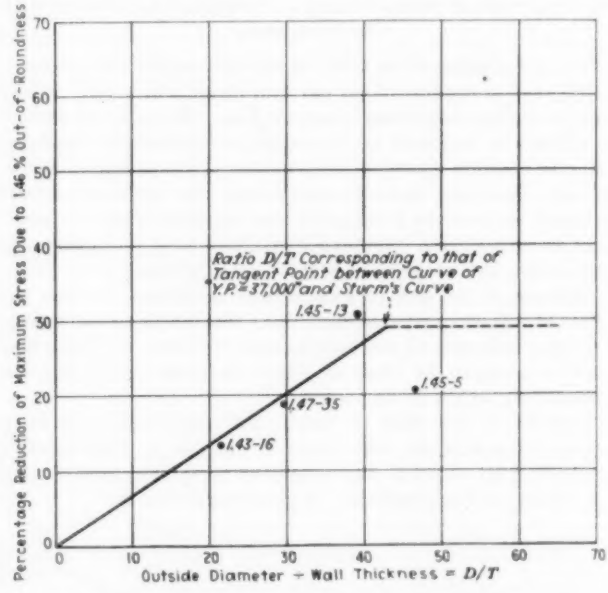


FIG. 20 SHOWING EFFECT OF 1.46 PER CENT OUT-OF-ROUNDNESS IN REDUCING THE COLLAPSING STRENGTH OF STEWART'S TUBES

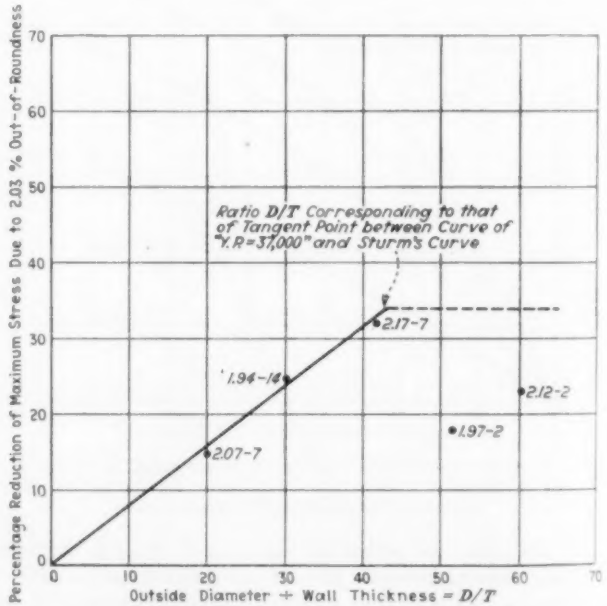


FIG. 21 SHOWING EFFECT OF 2.03 PER CENT OUT-OF-ROUNDNESS IN REDUCING THE COLLAPSING STRENGTH OF STEWART'S TUBES

In Figs. 18–21 the percentage reduction of stress in Stewart's tubes at the collapsing pressure is plotted against the ratio D/T for percentages of out-of-roundness ranging from 0.56 to 2.03.

The numbers accompanying the plotted points indicate first,

the average percentage of out-of-roundness of, and, second, the number of tubes represented by, the point: thus, in Fig. 18 the point having the coordinates of 9.9 per cent reduction in stress and $D/T = 18.5$ represents four tubes whose average out-of-roundness is 0.58 per cent, and whose average D/T ratio and average percentage reduction in stress are indicated by its coordinates.

In Figs. 19, 20, and 21 there are plotted the data on tubes having D/T ratios between 45 and 65. While there are not enough data at these higher ratios to evaluate quantitatively the effect of out-of-roundness, it is nevertheless evident that this effect decreases after attaining a maximum in the vicinity of $D/T = 40$. From a study of the factors that affect the collapsing strength of tubes, it has been concluded that this maximum is located at the point of tangency between the curve of the yield point of the steel in the tubes and the curve calculated from Sturm's equation. These curves and their tangent point are shown in Fig. 17. (In interpreting Fig. 17, it should be noted that the average yield point of the steel in Stewart's tubes is 37,000 lb. per sq. in.) In other words, the maximum effect of any given percentage of out-of-roundness occurs at the ratio D/T at which the yield point of the steel in the tube ceases to control the collapsing strength, and beyond which the collapsing strength is controlled only by the elastic properties of the steel and the tube dimensions. This ratio D/T is 43 for Stewart's tubes.

Since there are insufficient data available to evaluate quantitatively the effect of out-of-roundness at the higher D/T ratios, a horizontal broken line is drawn to indicate that the effect of out-of-roundness beyond the ratio $D/T = 43$ does not increase.

For purposes of design, the effect of out-of-roundness, in reducing the collapsing strength of tubes having D/T ratios greater than that of the tangent point, may safely be taken as that indicated by the ordinate of the horizontal broken line corresponding to the given out-of-roundness.

The curves shown in Figs. 18 to 21 were consolidated into the lower set of curves in Fig. 22. In the upper part of Fig. 22 there are drawn the curves showing the collapsing strength of two sets of perfect tubes having various D/T ratios, the steel in one set having a yield point of 35,000 lb. per sq. in., and that in the other of 75,000 lb. per sq. in. It is evident in Fig. 22 that (*a*) as the ratio D/T increases up to the tangent point, the percentage effect of out-of-roundness in reducing the collapsing strength also increases; and (*b*) the percentage effect of out-of-roundness in reducing the collapsing strength decreases for tubes having higher D/T ratios as the yield point of the steel in the tubes increases.

c Comparison of the Effect of Out-of-Roundness Based Upon Theoretical Considerations and That Indicated by the Experimental Results

A comparison of Figs. 16 and 22 indicates that the percentage effect of out-of-roundness based upon theoretical considerations is greater than that indicated by the experimental results. A small part of the discrepancy arises from the fact that the effect of out-of-roundness indicated by the experimental evidence is diminished by the variation in thickness, more specifically by the thicker portions of the tube.

d Effect of Out-of-Roundness in a Short Steel Tube

The foregoing data on the effect of out-of-roundness in reducing the collapsing strength of steel tubes were obtained from long tubes, that is, tubes having a ratio of length to diameter (L/D) of 8 or over. The data available on short tubes were not sufficient to reveal the effect of out-of-roundness on the collapsing strength of these tubes. However, it is believed that the effect of out-of-roundness in short tubes does not exceed that in long tubes because of the supporting action of the ends, and consequently the effect in long tubes may be regarded as the safe maximum which the effect of out-of-roundness in short tubes does not exceed.

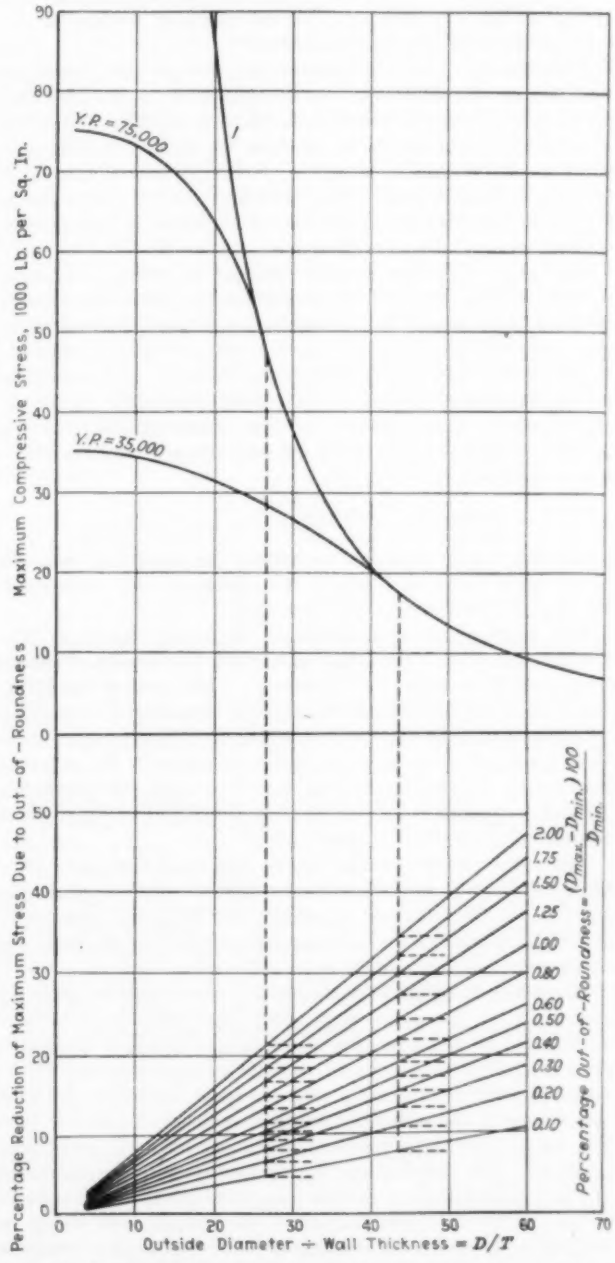


Fig. 22 Showing Effect of Out-of-Roundness of Steel Tubes in Reducing Maximum Compressive Stress at Collapsing Pressure, for Various D/T Ratios, With Particular Reference to Steels Having Yield-Point Values of 35,000 and 75,000 Lb. per Sq. In.

1—Calculated from Sturm's equation.

Since all short tubes in which the stress condition is of importance have closed ends, vessels may be considered as short tubes if their L/D ratios are less than 8, and as long tubes if their L/D ratios are 8 or over.

VII—EFFECT OF OTHER FACTORS ON THE COLLAPSING STRENGTH OF STEEL TUBES

a Variation in Wall Thickness

The variation in wall thickness of the tubes tested by A. O. Smith Corporation did not appear to affect the collapsing strength. The average of the variations in thickness $[(T_{avg.} - T_{min.}) \times 100/T_{min.}]$ of the collapsed sections of the tubes plotted in Fig. 1 was 2.5 per cent.

The data on Carman's tubes do not include the percentage variation in wall thickness. On account of the care with which his specimens were selected and machined, it is probable that the variation in thickness was so small as not to have a noticeable effect on the collapsing strength. A comparison of the curve of the collapsing strength of his tubes and Sturm's curve, shown in Fig. 3, indicates that the effect of variation in thickness in Carman's tubes may be neglected.

The data on Stewart's tubes indicate an average variation in wall thickness between the average and minimum thicknesses of about 12 per cent. As discussed in the section VI on the effect of out-of-roundness, these data are not sufficiently complete to enable the effects of variations in thickness and roundness on the collapsing strength to be readily separated. However, the out-of-roundness prevalent in these tubes appears to have a greater effect in reducing the collapsing strength than the variation in thickness.

b Method of Manufacturing the Tube

The method of manufacture affects the collapsing strength of a tube in so far as it affects the roundness and thickness of the tube and the yield point of the steel.

The average out-of-roundness of the tubes plotted in Fig. 1 is 0.77 per cent. According to Fig. 22, this degree of out-of-roundness (for a tube of steel having a yield point of 52,000 lb. per sq. in.) should produce a maximum reduction in collapsing strength of 18.2 per cent.[6] It is evident in Fig. 1, however, that the experimental curve is not below Sturm's curve, and that this degree of out-of-roundness does not appear to affect the collapsing strength. The explanation of this lack of effect seems to concern the method of manufacture.

The tubes plotted in Fig. 1 were cold-sized externally after being welded. It is believed that this cold sizing represses the effect of out-of-roundness in reducing the collapsing strength of the tubes, on the basis of the reasoning presented in the following paragraphs.

When the sizing press is closed, the tube is almost perfectly round. After the press is released, the tube may become slightly out of round on account of small changes in the elastic recovery, which are produced by slight variations in wall thickness. If uniform external pressure is again applied to the tube (as is done in the collapsing tests), it is believed that the tube will regain its nearly complete roundness before it eventually collapses. The fact that the slight out-of-roundness in the tubes of Fig. 1 does not affect the collapsing strength therefore appears to be due to the beneficial effect of external cold sizing in tending to preserve the roundness under subsequent compression.

This external cold sizing also increases the yield point of the metal in compression, and therefore tends to increase the collapsing strength of tubes having the lower D/T ratios.

[6] This value is obtained by noting that in Fig. 1 the tangent point between the experimental curve and Sturm's curve would theoretically lie at about $D/T = 37.5$, and that in Fig. 22, at $D/T = 37.5$, 0.77 per cent out-of-roundness would produce a reduction in collapsing strength of about 18.2 per cent.

VIII—DETERMINATION OF THE COLLAPSING STRENGTH OF A STEEL TUBE BY MEANS OF THE FUNDAMENTAL RELATIONSHIPS BETWEEN THE COLLAPSING STRENGTH AND THE FACTORS THAT AFFECT THIS STRENGTH

In this section illustrative examples will be presented of the determination of the collapsing strength of long and short tubes, both thick and thin, made of steel of various yield points.

It is believed that on the basis of the following procedure the collapsing strength of any commercial steel tube can be determined from its dimensions, the yield point of the steel, and its degree of out-of-roundness.

a Collapsing Strength of a Long, Thick Steel Tube (Having a Small D/T Ratio)

Let it be assumed that this tube is 8 or more diameters long, has a D/T ratio of 25, a yield point of steel of 35,000 lb. per sq. in., 1 per cent out-of-roundness, and a negligible variation in thickness. The collapsing pressure of this tube will be found from the following procedure.

1 Determine the maximum stress in the tube at the collapsing pressure by consulting Fig. 6. Follow the curve marked "Y. P. = 35,000" until the ratio $D/T = 25$ is reached. At this point read off the maximum stress of 28,800 lb. per sq. in. This is the maximum stress that can be withstood by a homogeneous tube of uniform roundness and thickness, having the aforementioned physical properties.

2 Determine the amount by which the maximum stress should be reduced on account of 1 per cent out-of-roundness by consulting Fig. 22. Note that the ratio $D/T = 25$ is less than the tangent-point ratio $D/T = 43.5$, between the curve marked "Y. P. = 35,000" and Sturm's curve. Hence the percentage reduction of maximum stress may be read from the intersection of the ordinate $D/T = 25$ with the 1 per cent out-of-roundness curve, and is found to be 13.9 per cent.

3 Solve for the collapsing pressure of the tube by substituting in the proper one of the following formulas. Use formula [E] for tubes having a D/T ratio of 32 or over, and [F] for D/T ratios of less than 32.

$$P = \frac{(100 - A)}{100} \frac{ST}{R_1} \quad \text{[E]}$$

$$P = \frac{(100 - A)}{100} \frac{S(R_1^2 - R_2^2)}{2R_1^2} \quad \text{[F]}$$

where P = collapsing pressure of the tube, lb. per sq. in.
A = percentage reduction of the maximum stress due to out-of-roundness
S = maximum compressive stress at the collapsing pressure of a tube with no out-of-roundness, lb. per sq. in.
T = thickness of the tube, inches
R_1 = outside radius of the tube, inches
R_2 = inside radius of the tube, inches.

Using formula [F], the collapsing pressure is found to be 1900 lb. per sq. in.

b Collapsing Strength of a Long, Thin Steel Tube (Having a Large D/T Ratio)

Let it be assumed that this tube is 8 or more diameters long, has a D/T ratio of 50, a yield point of steel of 35,000 lb. per sq. in., 1 per cent out-of-roundness, and a negligible variation in thickness. According to the steps outlined in the previous example:

1 Determine the maximum stress in the tube from Fig. 6 by locating the intersection of the curve marked "Y. P. =

35,000" and the ordinate $D/T = 50$. This stress is found to be 13,400 lb. per sq. in.

2 Determine the reduction in maximum stress due to out-of-roundness by consulting Fig. 22. Note that the ratio $D/T = 50$ is greater than the tangent-point ratio $D/T = 43.5$ for steel having a yield point of 35,000 lb. per sq. in., and consequently the percentage reduction is determined by the intersection of the ordinate $D/T = 50$ with the horizontal broken line drawn from the intersection of the tangent-point ratio $D/T = 43.5$ with the 1 per cent curve. The percentage reduction in stress is found to be 24.2.

3 Solve for the collapsing pressure of the tube by substituting in formula [E]. The collapsing pressure is found to be 406 lb. per sq. in.

c COLLAPSING STRENGTH OF A SHORT, THICK STEEL TUBE (HAVING A SMALL D/T RATIO)

Let it be assumed that this tube is 4 diameters long, has a D/T ratio of 20, a yield point of steel of 60,000 lb. per sq. in., 0.80 per cent out-of-roundness, and a negligible variation in thickness.

1 Determine the maximum stress in the tube by consulting Fig. 12 for $L/D = 4$. The maximum stress is determined by the intersection of the curve marked "Y. P. = 60,000" with the ordinate $D/T = 20$, and is found to be 54,200 lb. per sq. in.

2 Determine the reduction in maximum stress due to out-of-roundness by consulting Fig. 22. Note that the ratio $D/T = 20$ is less than the tangent-point ratio $D/T = 38.7$ between the curve marked "Y. P. = 60,000" and the curve of $L/D = 4$. Accordingly, the percentage reduction is determined by the intersection of the ordinate $D/T = 20$ with the 0.80 per cent out-of-roundness curve, and is found to be 9.9 per cent.

3 Solve for the collapsing pressure of the tube by substituting in formula [F]. The collapsing pressure is found to be 4640 lb. per sq. in.

d COLLAPSING STRENGTH OF A SHORT, THIN STEEL TUBE (HAVING A LARGE D/T RATIO)

Let it be assumed that this tube is 4 diameters long, has a D/T ratio of 55, a yield point of steel of 60,000 lb. per sq. in., 0.80 per cent out-of-roundness, and a negligible variation in thickness.

1 Determine the maximum stress in the tube by consulting Fig. 12. The maximum stress is determined by the intersection of the curve marked "Y. P. = 60,000" with the ordinate $D/T = 55$, and is found to be 22,200 lb. per sq. in.

2 Determine the reduction in maximum stress due to out-of-roundness by consulting Fig. 22. Note that the ratio $D/T = 55$ is greater than the tangent point ratio $D/T = 38.7$, for steel having a yield point of 60,000 lb. per sq. in., and consequently the percentage reduction is determined by the intersection of the ordinate $D/T = 55$ with the horizontal broken line drawn from the intersection of the tangent-point ratio $D/T = 38.7$ with the 0.80 per cent curve. The percentage reduction in stress is found to be 19.2 per cent.

3 Solve for the collapsing pressure of the tube by substituting in formula [E]. The collapsing pressure is found to be 652 lb. per sq. in.

IX—CONCLUSIONS

The method for determining the collapsing strength of steel tubes presented in this paper was developed from the following considerations.

1 The collapsing strength of a steel tube is affected by the following factors:

a Ratio of length of tube to diameter, L/D
b Ratio of diameter of tube to wall thickness, D/T
c Variations in thickness
d Yield point of the steel in the tube
e Elastic constants of the steel in the tube, represented by the modulus of elasticity, and Poisson's ratio
f Variations in roundness
g Method of manufacturing the tube.

2 The collapsing pressure of long steel tubes (having an L/D ratio of 8 or over) is controlled by the elastic properties and the ratio D/T for the higher D/T ratios; and by the yield point of the steel and the D/T ratio for the lower D/T ratios.

3 The collapsing pressure of short steel tubes (having an L/D ratio of less than 8) is controlled by the elastic properties and the L/D and D/T ratios for the higher D/T ratios; and by the yield point of the steel and the L/D and D/T ratios for the lower D/T ratios.

4 The effect of out-of-roundness in reducing the collapsing pressure of steel tubes increases as the ratio D/T increases, up to the tangent point of the curve of the yield point of the steel in the tube and the curve calculated from Sturm's equation; beyond this point the effect of out-of-roundness does not increase as the ratio D/T increases. The effect of out-of-roundness in reducing the collapsing pressure decreases, for tubes having higher D/T ratios, as the yield point of the steel in the tube increases.

X—REFERENCES

(1) "Collapsing Pressures of Bessemer Steel Lap-Welded Tubes, Three to Ten Inches in Diameter," by Reid T. Stewart, Trans. A.S.M.E., vol. 27 (1906), pp. 730–822.
(2) "The Collapse of Short, Thin Tubes," by A. P. Carman, Univ. of Ill. Eng. Exp. Sta. Bul. 99, 1917.
(3) Equation developed by R. G. Sturm, and included in a doctorate thesis which is to be submitted to the Department of Theoretical and Applied Mechanics, College of Engineering, University of Illinois, and described in a private communication dated December 5, 1927.
(4) "Applied Elasticity," by Timoshenko and Lessels, Equation [235], p. 248, Westinghouse Technical Night School Press (1925), East Pittsburgh, Pa.
(5) "Applied Elasticity," Equation [244], p. 252.
(6) "Strength of Materials," by Maurer and Withey, p. 292. John Wiley & Sons, Inc. (1925), New York.
(7) "Applied Elasticity," pp. 242–245.
(8) "Collapsing Pressures of Lap-Welded Steel Tubes," by Reid T. Stewart, Trans. A.S.M.E., vol. 29 (1907), pp. 123–130.
(9) "The Collapse of Short Tubes by External Pressure," by Gilbert Cook, *Philosophical Magazine*, July, 1914, pp. 51–56.
(10) "On the General Theory of Elastic Stability," by R. V. Southwell, Philosophical Trans., Royal Society of London, vol. 213A (Jan., 1913), pp. 187–244.
(11) "Resistance of Tubes to Collapse," by Carman and Carr, University of Illinois Exp. Sta. Bul. 5, 1906.

Appendix No. 1

EXPERIMENTAL WORK OF A. O. SMITH CORPORATION ON THE COLLAPSING STRENGTH OF STEEL TUBES

(*a*) *Method of Testing Tubes for Collapsing Pressure.* The experimental method employed for determining the collapsing pressure of tubes is essentially that used by Stewart, which it is believed was very well thought out by him (1). It consisted of subjecting the outside of the tube to gradually increasing uniform hydrostatic pressure until the tube was permanently deformed through collapse. The rate of expulsion of air from the inside of the tube was measured to indicate the approach of collapse. The maximum pressure attained was the collapsing pressure. It was read off two calibrated gages by two observers so as to minimize instrumental and observational errors. The lower of the two pressures was taken as the collapsing pressure.

The tubes tested were selected at random and were representative of commercial tubes.

Before testing the tube, thickness and roundness measurements were taken over a zone within which it was expected that collapse would occur. After testing the tube a section was cut through the collapsed portion and the thickness of the collapsed section measured. By these means the roundness and thickness of the collapsed section were accurately determined.

(*b*) *Description of Roundness-Measuring Apparatus.* The roundness of the tube is measured by means of the apparatus shown diagrammatically in Fig. 23. The tube *P* is revolved on the rollers *R* and upon rollers at the other end of the tube which are not shown in the figure. Differences in outside diameter from point to point on the periphery are determined by the gage *G*. The gage is attached to the upper arm *UA* through the holder *H*. The upper arm is rigidly connected to the lower arm *LA* through the post *V*. A counterweight *W* maintains

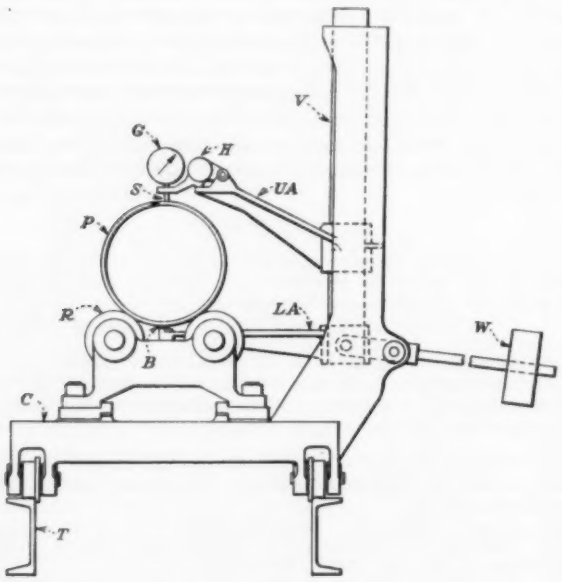


FIG. 23 APPARATUS FOR MEASURING ROUNDNESS OF PIPE

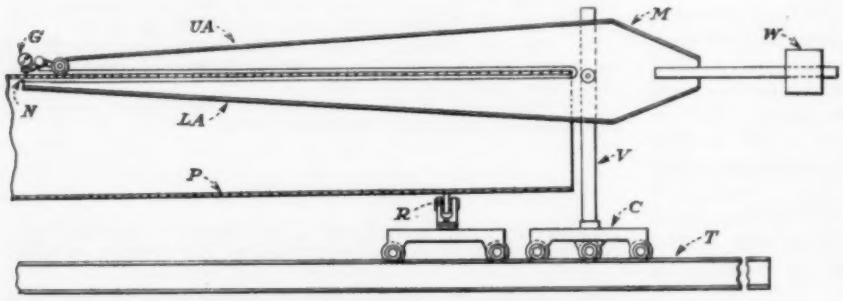


FIG. 24 APPARATUS FOR MEASURING THICKNESS OF PIPE

the lower arm in contact with the tube at *B*, which is diametrically opposite the gage *G*.

The upper arm is adjustable on the post, so that tubes of various diameters can be measured.

In order to measure roundness at intervals along the tube, the apparatus is mounted on the carriage *C* and run along the tracks *T*. The gage reads to 0.001 in.

(*c*) *Description of Thickness-Measuring Apparatus.* The wall thickness of the tube *P* is measured by means of the apparatus shown diagrammatically in Fig. 24. One half of the tube *P* is supported by the rollers *R* and the other half, which is not shown in the figure, by rollers at the other end of the tube. The measuring device *M* is rolled along the track *T* to the point at which a thickness measurement is desired, so that the upper arm *UA* is on the outside and the lower arm *LA* is on the inside of the tube. The measuring knob *N* is held in contact with the inner wall by the counterweight *W*. The wall thickness in thousandths of an inch is indicated on gage *G*.

The arms of the measuring device are adjustable on the vertical post *V*, so that the thickness can be measured for tubes of various diameters.

The carriage *C* enables the measuring device to be rolled back and forth along one half of the tube. The thickness of the other half is determined after the tube is reversed.

The wall thickness at various points on the periphery is measured after the tube is revolved on the rollers *R*.

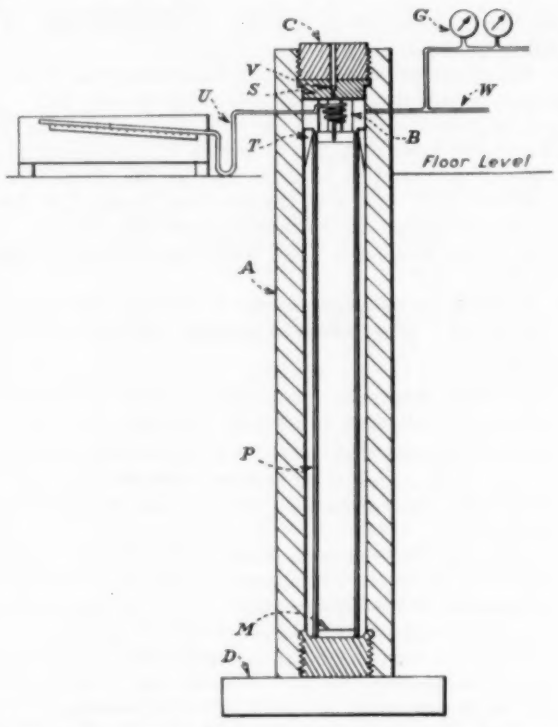


FIG. 25 APPARATUS FOR DETERMINING THE COLLAPSING PRESSURE OF PIPE

(*d*) *Description of Apparatus for Determining the Collapsing Pressure of Tubes.* The apparatus for determining the collapsing pressure of tubes is shown diagrammatically in Fig. 25. The pressure chamber *A* consists of an upright steel cylinder closed at the lower end, and resting on a concrete base *D* 13 ft. below the floor level. The chamber extends 3 ft. 3 in. above the floor level. Tubes 12 ft. long and up to 17 in. in outside diameter can be tested in this apparatus.

The screw cover *C* and secondary cover *S* are removable, so that the tube *P* may be lowered into place. Water under pressure is delivered to the chamber through the inlet *W*. Air between the tube and inner wall of the chamber is released through the valve *V*. The wooden blocks *B* and wedges *T* hold

the tube in place. The fittings required to make the apparatus watertight are not shown. Gages G indicate the pressure on the tube to the nearest five pounds per square inch for the lower ranges of pressure, and to the nearest ten pounds per square inch for the higher ranges.

Before a tube is tested, the ends are closed with the plugs M. A hole in the upper plug connects the inside of the tube with the right arm of the U-tube U. This tube is partially filled with water, and its left arm is open to the atmosphere. The function of the U-tube is to indicate the approach of collapse of the tube by revealing the rate at which the volume of expelled air changes with respect to pressure. At the approach of collapse, the amounts of air expelled increase for equal increments of pressure. The slightly inclined part of the left arm of the U-tube is adjustable, so that the expulsion of very small amounts of air can be detected. When collapse occurs, the water in the U-tube is forcibly ejected, and at the same time the pressure on the tube drops rapidly.

Appendix No. 2

WORK OF PREVIOUS INVESTIGATORS ON THE COLLAPSING STRENGTH OF STEEL TUBES

In the following discussion no attempt is made to evaluate the work of previous investigators. Their work is discussed only in relation to the development of the method of determining the collapsing strength of tubes which is presented in the paper.

(*a*) *Stewart.* In Stewart's investigation of the collapsing strength of commercial lap-welded steel tubes, the following factors were considered to have an influence on the collapsing strength of the tubes (1) (8).

1 Diameter of the tube
2 Length of tube between transverse joints or end connections tending to hold it to a circular form
3 Thickness of the wall
4 Deviation of the tube from perfect roundness
5 Physical properties of the material out of which the tube is made
6 Method of manufacturing the tube, in so far as it affects the uniformity of dimensions of the tube.

The empirical formulas for determining the collapsing strength of tubes that were developed in this investigation were based on the relationship between the collapsing pressure and the ratio of thickness to diameter (T/D). As a result of the investigation of the effect of the length of the tube upon the collapsing strength, it was concluded that the length has no practical influence on the collapsing pressure so long as it is not less than about six diameters. The extensive out-of-roundness measurements made in this investigation indicate that out-of-roundness was considered to have an influence on the collapsing strength of tubes. In the discussion of the physical properties of the steel in the tubes there were mentioned the yield point and ultimate strength. The method of manufacturing the tubes was considered to have an influence on the roundness and variation in thickness of the tubes.

Of the foregoing factors, the influence of the ratios L/D and T/D on the collapsing strength of lap-welded steel tubes were evaluated quantitatively.

(*b*) *Cook.* In Cook's investigation of the collapsing strength of "solid drawn" short, thin steel tubes, the ratios L/D and T/D and variation in thickness were considered to be important factors in determining the collapsing strength of steel tubes (9). On the basis of this investigation it was believed that the ratio L/D, above which the collapsing strength is unaffected by increase in length, probably varied from 13 to 18.

(*c*) *Carman.* In Carman's investigation of the collapsing strength of cold-drawn short steel tubes, the ratios L/D and T/D, out-of-roundness, and variations in thickness were considered to be important factors in determining the collapsing strength of steel tubes (2). From the results of this investigation it is evident that for steel tubes having an L/D ratio of 8 or more, the collapsing strength is practically unaffected by this ratio. In a previous investigation by Carman and Carr, the relationship between the collapsing strength of long, thin seamless tubes and the ratio T/D was indicated in the form of an Equation (11).

(*d*) *Southwell.* In Southwell's mathematical analysis of the collapse of short, thin tubes, an equation was developed for determining the collapsing pressure which took into consideration the dimensions of the tube, the modulus of elasticity, and Poisson's ratio (10).

(*e*) *Sturm.* Sturm's equation for determining the collapsing pressure of steel tubes has already been discussed in detail, so far as the dimensions of the tube, the modulus of elasticity, and Poisson's ratio are concerned.

Discussion

R. G. Sturm.[7] As Messrs. Jasper and Sullivan have pointed out, there are two distinct limitations upon the strength of thin shells under external pressure. One is that of purely elastic instability, and the other is that of instability resulting from the plastic yield of the material of which the shell is made. The first limiting condition is reached when the elastic distortions of the structure become relatively large under a slight increase in load. The other is reached when the strains in the material increase very rapidly with a slight increase in load. This is purely a case of plastic action, and the first signs of failure are indicated by the ordinary static-stress relationships. Between these two cases lies the range which cannot be figured by formulas for either case, because a combination of both actions really takes place.

Dr. H. M. Westergaard has divided the field of elastic action into three general classes:[8] orthostatic action, as in ordinary beams or tension members; astatic action, as in very long and thin columns or thin tubes; heterostatic action, as in crooked columns, columns with transverse loads, or thin tubes under similar conditions.

The field of heterostatic action in connection with the collapse of shells includes cases of partial hydrostatic pressure or concentrated loads resulting from supports and of concentrated or distributed loads resulting from appurtenances attached to the shell. Both papers show that their authors recognize this action.

Another classification of the actions of a shell subjected to external pressure may be made on the basis of the type of loads applied. In each general classification of loading just mentioned the loads may be applied in a number of ways. The buckling loads or pressures may be radial, axial, or a combination of both, thereby giving rise to biaxial buckling. The first type of loading is represented by that of a fire-tube boiler or a long pipe under partial vacuum. The shells referred to in the paper by Messrs. Saunders and Windenburg were subjected to both radial and axial pressure, uniform pressure being applied to both the sides and ends of the cylinders tested. This case is exemplified by the loading to which a submarine is subjected when it is submerged. A third case is that in which the end thrust is maintained and the radial pressure is removed. This case is exem-

[7] Research Engineer, Aluminum Company of America, New Kensington, Pa.

[8] "Buckling of Elastic Structures," by H. M. Westergaard, Trans. A.S.C.E., 1922, vol. 85, p. 577.

Appendix No. 3

TABLE 1 SHOWING SUMMARY OF DIMENSIONS, DIMENSION RATIOS, COLLAPSING PRESSURES, AND MAXIMUM STRESSES IN CARMAN'S LONG STEEL TUBES

Length, inches L	Outside diameter, inches D	Wall thickness, inches T	Ratio of length to outside diameter L/D	Ratio of outside diameter to thickness D/T	Collapsing pressure, lb. per sq. in. P	Maximum compressive stress at collapsing pressure, lb. per sq. in. S^1
7.5	0.988	0.0231	7.6	42.8	900	20,000
7.59	0.984	0.0212	7.7	46.5	710	17,200
7.44	0.970	0.0141	7.7	68.7	210	7,600
7.62	0.970	0.0141	7.9	68.7	190	6,900
7.62	0.978	0.0179	7.8	54.6	420	11,800
9.6	0.988	0.0231	9.7	42.8	900	20,000
9.56	0.984	0.0212	9.7	46.5	625	15,100
9.6	0.978	0.0179	9.8	54.6	440	12,400
9.56	0.970	0.0141	9.9	68.7	150	5,400
11.6	0.988	0.0231	11.7	42.8	750	16,600
11.63	0.984	0.0212	11.8	46.5	600	14,500
11.63	0.978	0.0179	11.9	54.6	375	10,500
11.6	0.970	0.0141	12.0	68.7	145	5,200
15.13	1.965	0.0459	7.7	42.8	960	21,600
15.13	1.947	0.0369	7.8	52.8	510	13,800
15.13	1.929	0.0281	7.8	68.7	220	7,900
18.94	1.965	0.0459	9.6	42.8	794	17,800
23.0	1.965	0.0459	11.7	42.8	780	17,500
27.8	1.965	0.0459	14.2	42.8	855	19,200
24.13	3.000	0.0651	8.0	46.1	684	16,200
24.25	3.00	0.0651	8.1	46.1	770	18,200

[1] Calculated by means of formula [C].

TABLE 2 SHOWING SUMMARY OF DIMENSIONS, DIMENSION RATIOS, COLLAPSING PRESSURES, AND MAXIMUM STRESSES IN STEWART'S LONG STEEL TUBES

No. of tubes represented by group averages	Average length of group, feet L	Average outside diameter of group, inches D	Average wall thickness of group, inches T	Ratio of average length to average diameter L/D	Ratio of average diameter to average thickness D/T	Average collapsing pressure of group, lb. per sq. in. P	Average maximum compressive stress at collapsing pressure of group, lb. per sq. in. S^1
9	20.0	2.997	0.112	80	26.8	1,860	25,400
2	20.0	2.992	0.143	80	20.9	2,963	32,400
5	20.1	2.995	0.188	80	16.0	4,095	35,300
5	20.2	3.993	0.119	60	33.5	964	14,300
5	20.2	4.014	0.175	60	22.9	2,280	23,600
5	20.2	4.026	0.212	60	19.0	3,170	31,700
5	20.1	4.014	0.327	60	12.3	5,560	36,800
10	20.0	6.017	0.130	40	46.3	524	12,600
18	20.0	6.028	0.167	40	36.1	928	16,500
13	20.0	6.024	0.186	40	32.4	1,251	20,200
7	20.0	6.011	0.222	40	27.1	1,779	24,600
20	20.1	6.026	0.266	40	22.7	2,441	28,100
6	20.1	6.660	0.155	36	43.0	592	13,200
4	20.0	6.654	0.166	36	40.1	670	13,500
5	20.0	6.684	0.200	36	33.4	1,184	20,300
9	20.0	6.677	0.251	36	26.7	1,879	26,300
5	20.1	6.655	0.268	36	24.8	2,224	29,000
10	20.1	7.028	0.160	34	43.9	592	13,200
5	20.0	7.050	0.242	34	29.1	1,680	30,000
5	20.2	6.987	0.279	35	25.0	2,147	27,800
5	20.0	8.643	0.185	28	46.7	536	12,500
8	20.0	8.660	0.268	28	32.3	1,419	23,200
11	17.3	8.653	0.309	24	28.0	1,708	25,500
5	15.0	8.654	0.308	21	28.1	1,791	26,800
5	15.0	8.653	0.184	21	47.0	548	13,700
5	10.0	8.656	0.178	14	48.7	548	13,700
5	14.2	8.642	0.215	20	40.2	847	17,600
4	11.8	8.663	0.219	16	39.6	835	17,400
5	10.0	8.669	0.214	14	40.5	845	17,700
5	15.0	8.652	0.272	21	31.8	1,540	25,000
5	10.0	8.668	0.267	14	32.5	1,533	26,000
5	15.0	8.659	0.275	21	31.5	1,421	24,200
5	10.0	8.671	0.271	14	32.0	1,541	25,200
5	20.2	10.041	0.165	24	63.0	225	7,090
5	20.1	10.026	0.194	24	53.1	383	10,100
5	20.2	10.001	0.316	24	32.6	1,319	22,000
5	19.5	10.779	0.512	22	21.0	2,585	28,400
5	20.0	12.790	0.511	19	25.0	2,196	28,200
5	20.0	13.036	0.244	18	53.4	463	12,300

[1] Calculated by means of formula [C].

TABLE 3 SHOWING SUMMARY OF DIMENSIONS, DIMENSION RATIOS, COLLAPSING PRESSURES, AND MAXIMUM STRESSES IN A. O. SMITH CORPORATION'S LONG STEEL TUBES

Test number	Length, feet L	Nominal outside diameter, inches D	Nominal wall thickness, inches T	Ratio of length to outside diameter L/D	Ratio of average outside diameter at collapsed section to average thickness at collapsed section D/T	Collapsing pressure, lb. per sq. in. P	Maximum compressive stress at collapsing pressure, lb. per sq. in. S^1
1	11.8	16	1/4	8.9	61.5	330	10,300
2	11.8	16	1/4	8.9	61.5	343	10,800
3	11.8	16	3/8	8.9	42.4	953	20,800
6	11.8	16	3/8	8.9	42.3	975	21,200
4	11.8	14	1/4	10	55.0	417	11,400
5	11.8	14	1/4	10	54.4	417	11,400
43	11.8	14	3/8	10	37.5	1,175	22,300
44	11.8	14	3/8	10	37.5	1,075	20,500
7	11.8	12 3/4	5/16	11	41.8	930	19,700
8	11.8	12 3/4	5/16	11	41.5	948	20,100
9	11.8	10 3/4	9/32	13	37.9	1,271	25,000
10	11.8	10 3/4	9/32	13	38.2	1,240	24,400
11	11.8	8 5/8	1/4	16	34.3	1,417	24,600
13	11.8	8 5/8	1/4	16	34.8	1,474	26,400

[1] Calculated by means of formula [C].

TABLE 4 SHOWING SUMMARY OF DIMENSIONS, DIMENSION RATIOS, COLLAPSING PRESSURES, AND MAXIMUM STRESSES IN "A" AND "B" BRANDS OF LONG STEEL TUBES

Test number	Length, feet L	Nominal outside diameter, inches D	Nominal wall thickness, inches T	Ratio of length to outside diameter L/D	Ratio of average outside diameter at collapsed section to average thickness at collapsed section D/T	Collapsing pressure, lb. per sq. in. P	Maximum compressive stress in tube at the collapsing pressure, lb. per sq. in. S^1
33-A	11.8	9	1/2	16	18.6	6,075	59,400
34-A	11.8	9	1/2	16	18.0	6,225	59,100
41-B	11.8	8 5/8	0.40	16	21.5	4,635	52,600
42-B	11.8	8 5/8	0.40	16	21.5	3,920	43,900

[1] Calculated by means of formula [C].

TABLE 5 SHOWING THE RELATION OF MAXIMUM COMPRESSIVE STRESS AT THE COLLAPSING PRESSURE TO THE YIELD POINT OF THE STEEL IN THE TUBES, FOR THE RATIO OF $D/T = 25$

Figure No.	Designation of tubes	Maximum compressive stress at collapsing pressure, lb. per sq. in.	Yield point of the steel in tubes, lb. per sq. in.
4	Stewart	28,500	37,000
1	A. O. Smith	37,000	52,000
5	B	43,500	66,600
3	A	47,400	77,300

TABLE 7 SHOWING THE VALUES OF MAXIMUM STRESS CORRESPONDING TO VARIOUS VALUES OF N (A TERM IN STURM'S EQUATION), FOR VARIOUS RATIOS OF LENGTH TO DIAMETER (L/D) AND DIAMETER TO THICKNESS (D/T)

Ratio of length to diameter, L/D	N	Ratio of diameter to thickness, D/T	Maximum compressive stress in the tube at the collapsing pressure S[1]
1.5	4.06	70	38,700
1.5	3.81	60	47,100
1.5	3.51	50	58,400
1.5	3.16	40	75,900
1.5	2.77	30	107,000
2	3.80	70	32,400
2	3.58	60	39,000
2	3.32	50	48,500
2	3.02	40	63,000
2	2.67	30	88,100
3	3.27	70	22,300
3	3.12	60	27,700
3	2.95	50	35,400
3	2.74	40	47,400
3	2.49	30	68,500
4	2.79	70	15,800
4	2.70	60	19,600
4	2.58	50	25,600
4	2.45	40	35,600
4	2.30	30	54,800
5	2.42	70	11,000
5	2.36	60	14,200
5	2.28	50	18,800
5	2.20	40	27,000
5	2.11	30	43,400
6	2.21	70	8,820
6	2.18	60	11,500
6	2.13	50	15,800
6	2.07	40	23,000
6	2.00	30	37,200
7	2.08	70	7,480

[1] S is calculated by means of the following formula:

$$S = \frac{PD}{2T}$$

$$P = \frac{2}{3}\frac{Et^3}{D^3}\left[\frac{N^4 - N^2 - k^2 + k(2N^2 - m)}{N^2(1 - m^2)}\right]$$

$$S = \frac{E}{3}\frac{t^2}{D^2}\left[\frac{N^4 - N^2 + k^2 + k(2N^2 - m)}{N^2(1 - m^2)}\right]$$

See Section IIIa for explanation of terms.

TABLE 6 SHOWING SUMMARY OF DIMENSION RATIOS, COLLAPSING PRESSURES, AND MAXIMUM STRESSES OF CARMAN'S SHORT STEEL TUBES

Length, inches L	Ratio of length to outside diameter L/D	Ratio of outside diameter to thickness D/T	Collapsing pressure, lb. per sq. in. P	Maximum compressive stress at collapsing pressure, lb. per sq. in. S
1.72	1.74	42.8	2,490	53,300
1.74	1.78	54.6	1,600	43,700
1.785	1.84	68.7	1,030	35,400
1.78	1.85	102	420	21,400
2.63	2.66	42.8	2,060	44,100
2.63	2.69	54.6	1,180	32,200
2.63	2.71	68.7	670	23,000
2.63	2.74	102	200	10,200
3.69	3.74	42.8	1,390	29,800
3.69	3.78	54.6	800	21,800
3.69	3.80	68.7	550	18,900
3.69	3.84	102	120	6,120
3.72	3.78	46.5	1,170	27,200
5.63	5.70	42.8	920	19,700
5.63	5.72	46.5	730	17,000
5.63	5.75	54.6	520	14,200
5.63	5.80	68.7	320	11,000
5.63	5.85	102	115	5,860
5.5	5.6	46.1	675	15,600
7.5	7.6	42.8	900	19,300
7.59	7.72	46.5	710	16,500
7.56	7.86	102	50	2,550
7.44	7.67	68.7	210	7,220
7.62	7.85	68.7	190	6,530
7.62	7.79	54.6	420	11,500
3.91	1.99	42.8	2,870	61,400
3.88	1.99	52.8	1,770	46,700
4	2.1	68.7	920	31,600
3.88	2.03	102	350	17,900
5.94	3.02	42.8	2,100	44,900
5.94	3.05	52.8	1,250	33,000
5.94	3.08	68.7	580	19,900
5.94	3.11	102	150	7,650
7.8	3.97	42.8	1,630	34,900
7.63	3.92	52.8	980	25,900
7.63	3.96	68.7	540	18,600
7.66	4.01	102	135	6,890
11.34	5.77	42.8	1,080	23,100
11.34	5.83	52.8	590	15,600
11.34	5.88	68.7	260	8,940
11.34	5.94	102	70	3,570
15.13	7.70	42.8	960	20,500
15.13	7.78	52.8	510	13,500
15.13	7.85	68.7	220	7,560
15.13	7.93	103	55	2,830
5.88	1.96	46.1	2,640	60,900
9.12	3.04	46.1	1,960	45,200
12.14	4.05	46.1	1,390	32,000
12.17	4.06	46.1	1,040	24,000
12.19	4.06	46.1	1,010	23,300
15.06	5.02	46.1	873	20,100
15.06	5.02	46.1	910	21,000
18.13	6.04	46.1	756	17,400
18.28	6.09	46.1	895	20,600
24.13	8.04	46.1	684	15,800
24.25	8.08	46.1	770	17,800

plified by the action of short, thin tubular struts subjected to end loads.

The effect of end conditions upon the elastic instability is of great importance for short tubes, but is practically negligible for long tubes. A separate classification of tubes according to end conditions is necessary for each of the subdivisions previously mentioned.

The tests made by Jasper and Sullivan covered the range of plastic action and extended into the range of elastic action of long, nearly hinged ended tubes, in which case the end thrust is not a significant factor. The tests made by Saunders and Windenburg were in the elastic range of short, practically fixed-ended tubes, in which case the end thrust is a controlling factor in collapse. It may be seen, then, that while the results presented in these two papers appear to be quite different, they each fit into the particular parts of the general behavior of tubes indicated by a general theory of the elastic and plastic behavior of tubes under external pressure.

D. S. JACOBUS.[9] This is a valuable paper, and the authors should be complimented on the complete way in which they have analyzed and correlated the experimental data and for making additional experiments where most needed. The results will be useful in rationalizing the tube sizes for use under external pressure.

REID T. STEWART.[10] This paper makes free use of the results

[9] Advisory Engineer, Babcock & Wilcox Company, New York, N. Y. Mem. A.S.M.E.

[10] Professor of Mechanical Engineering, University of Pittsburgh, Pittsburgh, Pa. Mem. A.S.M.E.

TABLE 8 SHOWING OUT-OF-ROUNDNESS DATA ON STEWART'S TUBES

Stewart's test number	Percentage out-of-roundness $100 \times \frac{(D_{max.} - D_{min.})}{D_{min.}}$	Ratio of average outside diameter to average thickness, D/T	Maximum compressive stress in perfect tubes, lb. per sq. in. S_1	Maximum compressive stress in Stewart's tubes, lb. per sq. in. S_2	Percentage reduction in collapsing strength due to out-of-roundness $\frac{(S_1 - S_2)\,100}{S_1}$
26	0.35	39.3	21,300	17,100	19.7
27	0.35	37.0	23,000	20,600	10.4
28	0.35	40.7	20,000	17,300	13.5
29	0.81	40.6	20,100	15,200	24.4
30	0.35	44.0	17,300	14,300	17.3
31	0.23	41.2	19,700	16,700	15.2
32	0.75	41.4	19,400	15,100	22.2
33	0.52	38.3	22,000	16,100	26.8
34	0.70	37.4	22,800	18,000	21.1
36	0.75	40.3	20,300	17,000	16.3
37	0.70	39.4	21,200	18,900	10.9
38	0.81	41.7	19,200	14,600	24.0
39	1.04	40.0	20,600	16,900	18.0
50	1.16	31.9	26,700	22,900	14.2
51	1.28	31.7	26,800	22,700	15.3
52	0.60	33.6	25,500	22,200	12.9
53	0.58	31.9	26,700	24,200	9.4
54	0.35	33.1	25,800	24,600	4.7
55	0.35	31.7	26,800	25,200	6.0
56	0.46	31.5	26,900	22,400	16.7
58	1.28	31.2	27,200	21,700	20.2
61	0.70	32.7	26,200	18,800	28.3
64	0.70	33.3	25,700	24,100	6.2
75	0.81	31.5	26,900	21,700	19.3
76	0.35	31.9	26,700	22,500	15.7
80	1.28	32.5	26,300	22,500	14.5
81	1.63	31.4	27,000	20,500	24.1
82	0.93	32.0	26,600	23,800	10.5
123	0.52	29.2	28,500	24,500	14.0
124	1.40	32.8	26,100	20,500	21.5
83	1.63	32.1	26,500	23,100	12.8
84	0.46	31.7	26,800	23,800	11.2
86	1.52	31.1	27,200	24,000	11.8
87	0.64	31.9	26,700	26,200	1.9
99	0.93	32.8	26,100	22,500	13.8
100	0.58	29.1	28,500	24,900	12.6
101	1.63	29.4	28,400	24,300	14.4
102	0.64	28.6	28,900	28,000	3.1
103	0.70	27.0	29,800	27,000	9.4
104	1.10	29.8	28,100	25,700	8.5
106	2.04	28.1	29,200	23,700	18.8
107	0.99	26.7	30,000	27,000	10.0
111	0.52	30.1	27,800	26,000	6.5
200	1.35	48.9	14,000	11,000	21.4
201	0.84	47.4	14,900	11,900	20.1
202	0.84	46.3	15,500	13,300	14.2
203	1.01	46.3	15,500	12,500	19.4
204	1.35	45.9	15,800	12,200	22.8
205	1.01	46.0	15,800	12,200	22.8
206	0.67	46.7	15,300	11,200	26.8
207	0.84	44.8	16,700	14,300	14.4
208	1.52	44.5	16,900	11,400	32.5
209	1.70	47.1	15,000	11,400	24.0
210	0.84	36.0	23,800	18,500	22.3
211	1.68	35.6	24,100	16,600	31.1
212	0.84	36.2	23,600	17,200	27.1
213	1.00	36.8	23,200	15,600	32.7
214	0.84	35.3	24,300	19,200	21.0

TABLE 8 (*Continued*)

Stewart's test number	Percentage out-of-roundness $100 \times \frac{(D_{max.} - D_{min.})}{D_{min.}}$	Ratio of average outside diameter to average thickness, D/T	Maximum compressive stress in perfect tubes, lb. per sq. in. S_1	Maximum compressive stress in Stewart's tubes, lb. per sq. in. S_2	Percentage reduction in collapsing strength, due to out-of-roundness $\frac{(S_1 - S_2)\,100}{S_1}$
215	0.84	35.9	23,900	18,100	24.3
216	1.00	38.9	21,600	14,800	31.5
217	1.34	34.5	24,900	17,400	30.1
218	1.34	35.8	23,900	15,400	35.6
219	1.00	36.3	23,600	17,800	24.6
221	1.50	36.3	23,700	13,800	41.8
224	1.00	36.3	23,700	20,200	14.8
223	0.67	35.7	23,900	19,100	20.1
225	1.34	36.4	23,500	13,000	44.7
238	1.17	36.1	23,700	14,000	40.9
291	1.68	35.9	23,800	18,900	20.6
240	1.17	33.7	25,400	18,500	27.2
241	1.76	27.5	29,500	22,000	25.4
242	0.84	32.6	26,200	20,700	21.0
243	1.50	35.4	24,200	15,500	35.9
245	1.34	31.1	27,200	21,400	21.3
246	1.35	26.1	30,400	24,800	18.4
247	1.68	33.9	25,200	17,100	32.1
249	1.17	26.5	30,000	15,900	47.0
250	0.67	31.3	27,100	22,700	16.2
251	1.17	28.1	29,200	24,600	15.8
252	1.51	27.1	29,800	28,100	5.7
253	1.85	32.7	26,200	15,500	40.8
254	2.02	27.3	29,700	21,200	28.6
256	1.00	32.5	26,300	23,600	10.3
257	1.68	31.9	26,700	19,900	25.5
258	1.18	33.0	25,900	22,300	13.9
259	1.68	33.1	25,800	18,200	29.5
261	0.83	24.0	31,500	21,000	33.3
263	1.68	21.5	32,700	28,000	14.4
264	1.33	23.1	31,900	28,300	11.3
273	1.51	22.3	32,300	25,300	21.7
276	1.00	23.3	31,800	28,800	9.4
279	1.67	23.9	31,500	28,200	10.5
282	1.51	22.5	32,200	25,300	21.4
285	1.17	22.3	32,300	28,400	12.1
301	1.29	40.4	20,300	14,600	28.1
302	2.05	39.6	21,000	12,500	40.5
303	1.67	40.3	20,400	14,700	27.9
304	1.29	40.5	20,200	12,200	39.6
305	1.82	34.1	25,100	18,800	25.1
306	1.67	33.5	25,500	20,200	20.8
307	2.13	33.1	25,800	21,100	18.2
308	1.44	32.6	26,200	20,600	21.4
310	0.45	25.6	30,600	29,100	4.9
311	0.76	26.3	30,200	28,400	6.0
312	0.99	26.5	30,100	26,200	13.0
314	0.83	26.6	30,000	28,900	3.7
315	1.00	44.6	16,800	12,700	24.4
316	1.14	44.5	16,900	12,200	27.8
317	1.58	45.6	16,000	13,100	18.1
318	0.86	42.3	18,900	12,300	34.9
319	1.14	43.3	17,900	11,700	34.6
320	1.36	28.8	28,700	25,600	10.8
321	1.29	29.3	28,500	23,100	19.0
322	1.43	30.3	27,700	23,100	16.6
323	1.58	28.8	28,700	24,100	16.0
324	1.29	28.4	29,000	26,300	9.3

TABLE 8 (*Continued*)

Stewart's test number	Percentage out-of-roundness $100 \times \frac{(D_{max.} - D_{min.})}{D_{min.}}$	Ratio of average outside diameter to average thickness, D/T	Maximum compressive stress in perfect tubes, lb. per sq. in. S_1	Maximum compressive stress in Stewart's tubes, lb. per sq. in. S_2	Percentage reduction in collapsing strength due to out-of-roundness $\frac{(S_1 - S_2)\,100}{S_1}$
401	2.28	42.4	18,700	12,400	33.7
402	2.28	43.9	17,300	12,800	26.0
403	2.13	43.3	17,900	12,100	32.4
404	2.13	43.5	17,700	11,300	36.1
405	1.67	25.4	30,700	27,100	11.7
406	0.75	25.4	30,700	25,100	18.2
407	1.36	23.9	31,500	30,600	2.9
408	0.91	25.5	30,600	26,300	14.1
409	1.36	23.4	31,800	27,400	13.8
410	2.12	27.7	29,400	23,300	20.7
411	1.06	26.5	30,100	25,700	14.6
412	1.82	26.1	30,400	23,200	23.7
413	1.51	26.9	29,900	23,000	23.1
414	1.52	26.9	29,900	21,700	27.4
415	1.16	24.3	31,300	27,000	13.7
416	1.00	23.1	31,900	29,500	7.5
417	1.33	22.3	32,300	26,800	17.0
418	1.00	21.9	32,500	31,600	2.8
419	0.83	22.3	32,300	28,700	11.1
420	1.05	27.9	29,300	25,200	14.0
421	0.70	28.6	28,900	27,700	4.2
422	1.51	28.3	29,100	23,100	20.6
423	1.16	28.5	28,900	26,100	9.7
424	1.87	28.7	28,800	22,600	21.5
430	1.01	24.1	31,400	29,500	6.1
431	1.16	24.7	31,100	28,400	8.7
432	1.15	26.2	30,300	24,000	20.8
433	1.16	24.3	31,300	26,500	15.3
434	1.15	25.9	30,500	25,600	16.1
435	2.03	43.6	17,600	14,500	17.6
436	1.29	41.8	19,200	13,500	29.7
437	1.29	45.2	16,400	13,700	16.5
438	1.43	43.3	17,900	14,600	18.4
440	1.34	21.9	32,500	30,400	6.5
441	2.19	22.3	32,300	24,000	25.7
442	1.01	22.3	32,300	28,900	10.5
443	1.85	22.1	32,400	27,200	16.1
444	1.34	21.9	32,500	26,900	17.2
446	2.22	60.1	9,200	6,760	26.5
447	2.01	60.5	9,000	7,260	19.3
451	2.02	51.7	12,400	10,100	18.6
454	1.92	51.5	12,500	10,300	17.6
455	2.02	31.3	27,100	20,000	26.2
456	1.82	32.0	26,600	21,600	18.8
458	1.82	31.5	26,900	20,600	23.4
459	2.02	31.9	26,700	22,100	17.2
460	1.26	33.5	25,500	15,500	39.2
461	2.03	32.7	26,200	15,900	39.3
462	1.01	32.7	26,200	16,800	35.9
463	0.75	33.3	25,700	17,200	33.1
464	2.28	35.0	24,500	15,100	38.4
465	1.00	23.2	31,800	23,800	25.1
466	1.50	22.5	32,200	25,000	22.4
467	2.27	23.2	31,800	28,100	11.6
468	1.00	21.9	32,500	27,800	14.5
470	2.26	18.7	34,000	29,200	14.1
471	0.50	18.9	33,900	29,500	13.0
472	1.50	19.2	33,800	30,300	10.4
473	0.50	18.6	34,000	31,400	7.6
474	1.25	19.6	33,600	30,100	10.4

TABLE 8 (*Continued*)

Stewart's test number	Percentage out-of-roundness $100 \times \frac{(D_{max.} - D_{min.})}{D_{min.}}$	Ratio of average outside diameter to average thickness, D/T	Maximum compressive stress in perfect tubes, lb. per sq. in. S_1	Maximum compressive stress in Stewart's tubes, lb. per sq. in. S_2	Percentage reduction in collapsing strength due to out-of-roundness $\frac{(S_1 - S_2)\,100}{S_1}$
480	1.34	27.3	29,700	21,200	28.6
483	1.69	27.3	29,700	23,600	20.5
484	1.35	26.9	29,900	27,200	9.0
485	1.35	26.7	30,000	24,000	20.0
486	1.68	26.8	30,000	24,800	17.3
487	1.34	26.8	30,000	26,300	12.3
488	1.68	26.3	30,200	26,600	11.9
489	1.35	26.5	30,100	28,800	4.3
491	1.35	21.5	32,700	27,700	15.3
495	1.35	15.7	35,100	33,000	6.0
496	1.34	15.7	35,100	33,000	6.0
497	2.03	15.8	35,000	33,000	5.7
498	2.03	16.5	34,800	30,500	12.4
499	0.67	15.8	35,000	33,200	5.1
500	0.84	21.2	32,900	26,000	21.0
501	1.88	21.1	32,900	27,200	17.3
503	1.31	21.3	32,800	26,500	19.2
504	0.65	20.5	33,200	28,600	13.9
505	0.79	25.3	30,800	30,200	1.9
506	0.87	24.7	31,100	28,200	9.3
507	1.02	25.3	30,800	26,400	14.3
508	0.78	24.8	30,100	24,800	17.6
509	0.79	24.9	30,000	27,600	8.0

TABLE 9 SHOWING CLASSIFICATION OF OUT-OF-ROUNDNESS DATA CONTAINED IN TABLE 1

Key to Classification of Group A:

Group	Range of ratio D/T of tubes	Range of percentage out-of-roundness of tubes
A-1	15–25	0.25–0.75
A-2	15–25	0.75–1.25
A-3	15–25	1.25–1.75
A-4	15–25	1.75–2.25

GROUP A-1

Stewart's test number	Per cent out-of-roundness	Ratio D/T	Per cent red. of max. stress due to out-of-roundness
471	0.50	18.9	13.0
473	0.50	18.6	7.6
499	0.67	15.8	5.1
504	0.65	20.5	13.9
Average	0.58	18.5	9.9
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..

GROUP A-2

Stewart's test number	Per cent out-of-roundness	Ratio D/T	Per cent red. of max. stress due to out-of-roundness
430	1.01	24.1	6.1
431	1.16	24.7	8.7
433	1.16	24.3	15.3
506	0.87	24.7	9.3
508	0.78	24.8	17.6
509	0.79	24.9	8.0
261	0.83	24.0	33.3
276	1.00	23.3	9.4
285	1.17	22.3	12.1
415	1.16	24.3	13.7
416	1.00	23.1	7.5
418	1.00	21.9	2.8
419	0.83	22.3	11.1
442	1.01	22.3	10.5
465	1.00	23.2	25.1
468	1.00	21.9	14.5
474	1.25	19.6	10.4
500	0.84	21.2	21.0
Average	0.99	23.2	13.1

TABLE 9 (*Continued*)

GROUP A-3

Stewart's test number	Per cent out-of-roundness	Ratio D/T	Per cent red. of max. stress due to out-of-roundness
263	1.68	21.5	14.4
264	1.33	23.1	11.3
273	1.51	22.3	21.7
279	1.67	23.9	10.5
282	1.51	22.5	21.4
407	1.36	23.9	2.9
409	1.36	23.4	13.8
417	1.33	22.3	17.0
440	1.34	21.9	6.5
444	1.34	21.9	17.2
466	1.50	22.5	22.4
472	1.50	19.2	10.4
491	1.35	21.5	15.3
495	1.35	15.7	6.0
496	1.34	15.7	6.0
503	1.31	21.3	19.2
...	...	...	...
Average	1.43	21.4	13.5

GROUP A-4

Stewart's test number	Per cent out-of-roundness	Ratio D/T	Per cent red. of max. stress due to out-of-roundness
441	2.19	22.3	25.7
443	1.85	22.1	16.1
467	2.27	23.2	11.6
470	2.26	18.7	14.1
497	2.03	15.8	5.7
498	2.03	16.5	12.4
501	1.88	21.1	17.3
Average	2.07	20.0	14.7
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..

Key to Classification of Group B:

Group	Range of ratio D/T of tubes	Range of percentage out-of-roundness of tubes
B-1	25–35	0.25–0.75
B-2	25–35	0.75–1.25
B-3	25–35	1.25–1.75
B-4	25–35	1.75–2.25

GROUP B-1

Stewart's test number	Per cent out-of-roundness	Ratio D/T	Per cent red. of max. stress due to out-of-roundness
52	0.69	33.6	12.9
53	0.58	31.9	9.4
54	0.35	33.1	4.7
421	0.70	28.6	4.2
463	0.75	33.3	33.1
55	0.35	31.7	6.0
56	0.46	31.5	16.7
61	0.70	32.7	28.3
64	0.70	33.3	6.2
76	0.35	31.9	15.7
123	0.52	29.2	14.0
84	0.46	31.7	11.2
87	0.64	31.9	1.9
100	0.58	29.1	12.6
102	0.64	28.6	3.1
103	0.70	27.0	9.4
111	0.52	30.1	6.5
310	0.45	25.6	4.9
250	0.67	31.3	16.2
406	0.75	25.4	18.2
Average	0.58	30.6	11.8
...	..	..	..
...	..	..	..
...	..	..	..
...	..	.	..

GROUP B-2

Stewart's test number	Per cent out-of-roundness	Ratio D/T	Per cent red. of max. stress due to out-of-roundness
432	1.15	26.2	20.8
434	1.15	25.9	16.1
50	1.16	31.9	14.2
420	1.05	27.9	14.0
423	1.16	28.5	9.7
505	0.79	25.3	1.9
507	1.02	25.3	14.3
462	1.01	32.7	35.9
75	0.81	31.5	19.3
82	0.93	32.0	10.5
99	0.93	32.8	13.8
104	1.10	29.8	8.5
107	0.99	26.7	10.0
311	0.76	26.3	6.0
312	0.99	26.5	13.0
314	0.83	26.6	3.7
240	1.17	33.7	27.2
242	0.84	32.6	21.0
256	1.00	32.5	10.3
258	1.18	33.0	13.9
249	1.17	26.5	47.0
251	1.17	28.1	15.8
408	0.91	25.5	14.1
411	1.06	26.5	14.6
Average	1.01	28.9	15.7

TABLE 9 (*Continued*)

GROUP B-3

Stewart's test number	Per cent out-of-roundness	Ratio D/T	Per cent red. of max. stress due to out-of-roundness
51	1.28	31.7	15.3
422	1.51	28.3	20.6
480	1.34	27.3	28.6
483	1.60	27.3	20.5
484	1.35	26.9	9.0
485	1.35	26.7	20.0
486	1.68	26.8	17.3
487	1.34	26.8	12.3
488	1.68	26.3	11.9
489	1.35	26.5	4.3
460	1.26	33.5	39.2
58	1.28	31.2	20.2
80	1.28	32.5	14.5
81	1.63	31.4	24.1
124	1.40	32.8	21.5
83	1.63	32.1	12.8
86	1.52	31.1	11.8
101	1.63	29.4	14.4
217	1.34	34.5	30.1
252	1.51	27.1	5.7
257	1.68	31.9	25.5
259	1.68	33.1	29.5
245	1.34	31.1	21.3
246	1.35	26.1	18.4
247	1.68	33.9	32.1
306	1.67	33.5	20.8
308	1.44	32.6	21.4
405	1.67	25.4	11.7
413	1.51	26.9	23.1
414	1.52	26.9	27.4
320	1.36	28.8	10.8
321	1.29	29.3	19.0
322	1.43	30.3	16.6
323	1.58	28.8	16.0
324	1.29	28.4	9.3
...	...	...	...
Average	1.47	29.6	18.8

GROUP B-4

Stewart's test number	Per cent out-of-roundness	Ratio D/T	Per cent red. of max. stress due to out-of-roundness
424	1.87	28.7	21.5
455	2.02	31.3	26.2
456	1.82	32.0	18.8
458	1.82	31.5	23.4
459	2.02	31.9	17.2
461	2.03	32.7	39.3
106	2.04	28.1	18.8
253	1.85	32.7	40.8
254	2.02	27.3	28.6
305	1.82	34.1	25.1
307	2.13	33.1	18.2
410	2.12	27.7	20.7
412	1.82	26.1	23.7
241	1.76	27.5	25.4
Average	1.94	30.3	24.9
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..

Key to Classification of Group C:

Group	Range of ratio D/T of tubes	Range of percentage out-of-roundness of tubes
C-1	35–45	0.25–0.75
C-2	35–45	0.75–1.25
C-3	35–45	1.25–1.75
C-4	35–45	1.75–2.25

GROUP C-1

Stewart's test number	Per cent out-of-roundness	Ratio D/T	Per cent red. of max. stress due to out-of-roundness
33	0.52	38.3	26.8
26	0.35	39.3	19.7
27	0.35	37.0	10.4
28	0.35	40.7	13.5
30	0.35	44.0	17.3
31	0.23	41.2	15.2
223	0.67	35.7	20.1
32	0.75	41.4	22.2
34	0.70	37.4	21.1
36	0.75	40.3	16.3
37	0.70	39.4	10.9
Average	0.52	39.5	17.6
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..
...	..	..	..

GROUP C-2

Stewart's test number	Per cent out-of-roundness	Ratio D/T	Per cent red. of max. stress due to out-of-roundness
207	0.84	44.8	14.4
29	0.81	40.6	24.4
38	0.81	41.7	24.0
39	1.04	40.0	18.0
224	1.00	36.3	14.8
238	1.17	36.1	40.9
210	0.84	36.0	22.3
212	0.84	36.1	27.1
315	1.00	44.6	24.4
316	1.14	44.5	27.8
318	0.86	42.3	34.9
319	1.14	43.3	34.6
213	1.00	36.8	32.7
214	0.84	35.3	21.0
215	0.84	35.9	24.3
216	1.00	38.9	31.5
219	1.00	36.3	24.6
...	..	..	..
Average	0.95	39.4	26.0

TABLE 9 (*Continued*)

GROUP C-3

Stewart's test number	Per cent out-of-roundness	Ratio D/T	Per cent red. of max. stress due to out-of-roundness
221	1.50	36.3	41.8
225	1.34	36.4	44.7
291	1.68	35.9	20.6
208	1.52	44.5	32.5
211	1.68	35.6	31.1
301	1.29	40.4	28.1
303	1.67	40.3	27.9
304	1.29	40.5	39.6
218	1.34	35.8	35.6
436	1.29	41.8	29.7
437	1.29	45.2	16.5
243	1.50	35.4	35.9
438	1.43	43.3	18.4
Average	1.45	39.3	30.9

GROUP C-4

Stewart's test number	Per cent out-of-roundness	Ratio D/T	Per cent red. of max. stress due to out-of-roundness
302	2.05	39.6	40.5
403	2.13	43.3	32.4
404	2.13	43.5	36.1
435	2.03	43.6	17.6
401	2.28	42.4	33.7
402	2.28	43.9	26.0
464	2.28	35.0	38.4
Average	2.17	41.6	32.1

Key to Classification of Groups D and E:

Group	Range of ratio D/T of tubes	Range of percentage out-of-roundness of tubes
D-2	45–55	0.75–1.25
D-3	45–55	1.25–1.75
D-4	45–55	1.75–2.25
E-4	55–65	1.75–2.25

GROUP D-2

Stewart's test number	Per cent out-of-roundness	Ratio D/T	Per cent red. of max. stress due to out-of-roundness
201	0.84	47.4	20.1
202	0.84	46.3	14.2
203	1.01	46.3	19.4
205	1.01	46.0	22.8
Average	0.93	46.5	19.1

GROUP D-3

209	1.70	47.1	24.0
317	1.58	45.6	18.1
437	1.29	45.2	16.5
200	1.35	48.9	21.4
204	1.35	45.9	22.8
Average	1.45	46.5	20.6

GROUP D-4

451	2.02	51.7	18.6
454	1.92	51.5	17.6
Average	1.97	51.6	18.1

GROUP E-4

Stewart's test number	Per cent out-of-roundness	Ratio D/T	Per cent red. of max. stress due to out-of-roundness
446	2.22	60.1	26.5
447	2.01	60.5	19.3
Average	2.12	60.3	22.9

of an extensive investigation of the collapsing properties of bessemer-steel lap-welded tubes published by the writer in 1906 Transactions of this Society. The purpose was to produce a dependable formula that could be readily used to determine the probable collapsing pressures of commercial tubes as then manufactured. The undertaking was meant to be of the most practical nature, but the writer spent much time in an effort to apply theory to the problem. In the end, however, he was obliged to resort to empiricism.

The authors of this paper seem to consider seriously the effects of the out-of-roundness of the tubes as reported by the writer. He is somewhat surprised at this, as he abandoned all idea of incorporating out-of-roundness in a formula after studying some 5000 experimental graphs made in this investigation. If the out-of-roundness had been elliptical in cross-section for a length of at least six diameters, the problem might have been comparatively simple. But most of the tubes were out of round in the most complex manner, not approximating even remotely to the theoretical elliptical section. Then, too, the characteristics of two adjacent sections are sometimes very different. The short diameter of one may roughly be at right angles to that of its neighbor, thus tending to lessen the weakness that would be caused by each alone. Many of the tubes did not fail at the place of greatest departure from roundness.

Even should a dependable formula be devised for the effects of out-of-roundness, which is not at all likely at present, of what possible use could it be in a commercial sense? The writer has spent many years in measuring, in a specially equipped laboratory, the out-of-roundness of tubes. With this experience in mind, it is inconceivable how any manufacturer of tubes could be persuaded to adopt a system of testing for out-of-roundness applicable to commercial tubes that are intended to resist collapsing pressures. This statement is made in view also of hundreds of comparatively recent tests on seamless-steel tubes manufactured for well casing. The results of these tests show, even to a more marked degree, the impracticability of considering out-of-roundness in the selling and buying of steel tubes. Of course, the writer is limiting his statements to out-of-roundness as found in legitimate manufacture and not to tubes that become unduly out of round due to subsequent rough handling.

Regarding the customary variations of thickness in the tube wall, as found in commerce, the less that engineers talk about the matter the better it will be for all concerned. Tubes of homogeneous material invariably fail at the thinnest part when subjected to internal fluid pressure. But this is not at all the case for similar tubes subjected to external fluid pressure. Both the results of tests reported by the writer in 1906 and those of recent tests on seamless-steel tubes, so far as he has worked over the results, show that the "axis of collapse" is not noticeably related to the thinnest portion of the tube wall. Its consideration has no place in engineering calculations.

Formulas based upon the theory of elasticity, such as Sturm's, given prominence in this paper, are generally useless for industrial calculations. The writer has been applying collapse formulas for about 25 years and does not recall a single instance in which such formulas were applicable, because of their limitation to the relatively thin walls, in which he was not concerned. Simple empirical formulas based directly upon actual tests of

commercial tubes, both welded and seamless, are entirely satisfactory for all cases of thickness divided by outside diameter greater than 0.023, which corresponds to the author's manner of expressing it of D/t less than 43, which covers the entire commercial range that the writer thus far has experienced.

In view of the foregoing, the writer cannot accept the authors' method of calculating the collapsing pressures of steel tubes, because he believes the procedure, in the light of present knowledge, is impracticable. It is not apparent just how the authors constructed the curves in Fig. 6 marked Y.P. 25,000 to 80,000, all of which are tangent to curve 1 calculated by Sturm's equation.

This chart is stated to apply to pipe that is perfectly round and uniformly thick. Presumably this chart is based upon Carman's tests on 21 cold-drawn steel tubes, 1 to 3 in. in diameter, which were machined in order to make their diameters and wall thickness uniform.

Unfortunately Carman did not test specimens cut from the same tubes, with the customary variations in diameter and wall thickness, in order to determine the direct relation of properties of the actual commercial to the machined perfect tubes. The authors therefore go to the writer's report of 1906 on an entirely different product, manufactured by an entirely different process—namely, bessemer-steel lap-welded tubes, 3 to 12 in. in diameter—for correction data on out-of-roundness to apply to calculations on the perfect tubes. In view of the numerous tests that the writer has made to date, both on welded and seamless steel tubes, the meagerness of Carman's data, and the inapplicability of the writer's out-of-round data, the authors' methods of calculating collapsing pressures seems to be ill-advised and unwarranted.

Near the bottom of the first column of the twelfth page there is the statement: "When the sizing press is closed, the tube is almost perfectly round. After the press is released, the tube may become slightly out of round on account of small changes in the elastic recovery, . . . If uniform external pressure is again applied (as is done in the collapsing tests), it is believed that the tube will regain its nearly complete roundness before it eventually collapses." The writer would like to ask the authors on what principles of mechanics this belief of theirs is based. Those who know their mechanics will agree, the writer feels sure, that this belief is entirely false, as uniform pressure was not applied by the sizing press in rounding up the tube, as inferred by the authors. In rounding up the tube the high portions would of necessity be subjected to great pressure, while the low portions at the same time would be subjected to very little or no pressure.

Only thus could the tube be squeezed into roundness. When subjected afterward to an external fluid pressure, which of course is uniformly distributed, any out-of-roundness in the tube will not decrease, as stated by the authors, but will increase. If the authors belief were correct, a tube could not possibly collapse. It would keep round all the while as the fluid pressure increased and would begin to fail only when the compression strength of the steel was reached, when the wall thickness would begin to increase and continue to increase with increase of fluid pressure.

GILBERT COOK.[11] A survey of the investigations which have been carried out in the subject of the collapse of tubes by external pressure reveals the comparatively small part which elastic theory has played in the framing of formulas and rules for practical design. The majority of experimenters in this field in former years have been content to devise empirical formulas to represent the results of their tests, a procedure which was, however, inevitable in the absence of an adequate theory. The disadvantage of an empirical formula is that its application must be limited to the conditions of the tests upon which it is based, and it gives no indication of the effect of any change in those conditions. While therefore it has been known that variations in thickness and imperfections of shape must affect the collapsing pressure of a tube, it has not been possible to predict the extent of their influence.

The mathematical theory of the stability of a cylindrical tube of uniform thickness was first worked out by Bryan in 1888, and he gave the formula [B] quoted by the authors. Previous to this, as long ago as 1875, Unwin had pointed out the analogy that existed between the collapse of a tube and the failure of a column under a compressive load, and obtained by this means a formula differing from [B] only in the numerical coefficient. The more complete theory given by Southwell covered the short as well as the long tube, and although the utility of Southwell's formula for the ordinary conditions of practice was diminished by the uncertainty regarding a numerical constant, it was the first successful attempt to indicate the manner in which the three variables of length, diameter, and thickness were concerned in the problem of collapse. The writer has demonstrated experimentally the accuracy of the formula for the particular end conditions for which the value of the constant was known.

It is interesting to observe that Southwell pointed out that the effect of the length of a tube, in so far as elastic stability was concerned, depended upon its relation to the quantity $\sqrt{(D^3/T)}$. It is not strictly correct to regard simply the L/D ratio as the criterion for a long or short tube. It is a convenience to do so, and may in normal circumstances be sufficiently accurate, but it is analogous to taking the slenderness ratio of a column as the ratio of length to transverse width, and ignoring the shape of the cross-section. It is probable that where elastic breakdown rather than instability is the cause of collapse, the effect of length is quite different, and should be expressed in terms of $\sqrt{(DT)}$. In either case the thickness cannot be entirely ignored in any true representation of the conditions.

The authors have made considerable use of a formula by Sturm. An account of the derivation of this formula does not appear to have been published, and it would be interesting to know in what way it is superior to that of Southwell.

Unfortunately the theory of elasticity has very little to say about the effect of variations in shape and thickness, which is the main theme of the author's paper. Experiment alone can, at present, give any reliable information in this important matter, and the authors have performed a most useful service in the careful analysis they have made of existing data. There are, however, one or two statements which call for comment. In Section II it is stated that "Variations in thickness below the average thickness tend to weaken the tube and adversely affect the collapsing strength; whereas variations above the average thickness, under special conditions, may strengthen the pipe and tend to increase the collapsing strength." The meaning of this paragraph is obscure, as the definition of "average" implies variations both above and below.

The explanation given for the effect described by the authors in Section VII (*b*) does not appear to be sound. For a given perimeter, the figure of greatest area is the circle. In order therefore that a tube initially out of round should become circular there must be an increase in area, and accordingly the work done by the external pressure would be negative, which is clearly impossible. Reverting to the column analogy, it is equivalent to the assertion that a column initially bent may become straight under the application of a compressive load. Is it not more likely that the yield point of the material has been underestimated?

[11] Professor, King's College, London, England.

CYRIL O. RHYS.[12] The general method followed in investigating pressures and stresses in tubes when about to collapse makes considerable use of the analogy between the failure of a column and the collapse of a tube. For a column, it is shown that Euler's equation,

$$\text{Max. unit load} = \frac{F}{A} = E\,\pi^2\,\frac{R^2}{L^2} \quad \ldots\ldots [1]$$

where

F = total load on column
A = cross-sectional area of column
E = Young's modulus
L = length of column
R = least radius of gyration of cross-section

agrees with experiment for long columns, but diverges from observed results for short ones. This is of course because Euler's equation is based on failure by bending, and short columns fail by direct crushing. Taking Tetmajer's experimental results for short columns, it is shown, in Fig. 2 of the paper, that these can be combined with Euler's equation so as to give a smooth curve, representing the relation between maximum unit load and slenderness ratio L/R. In the foregoing no allowance is made for eccentric loading.

Very much the same procedure is followed, in the paper, for tubes. Substituting the equation of Sturm or Timoshenko for that of Euler, and the results of pipe tests for Tetmajer's experiments, curves such as that of Fig. 2 are plotted showing the relation between maximum compressive stress and the ratio D/T or outside diameter ÷ wall thickness. The tubes are supposed to be straight, round, and of uniform thickness. Variations in these properties are allowed for by other charts such as Fig. 16 and Fig. 22. Equations [E] and [F] show how chart readings are combined to calculate collapsing pressures for various cases.

Returning to the column analogy, for short columns, the load which will produce rupture is fA, if f is the ultimate compressive stress, or

$$\frac{F}{A} = f \quad \ldots\ldots [2]$$

This can be combined with the Euler equation in the form

$$\frac{F}{A} = \frac{f}{1 + \dfrac{fL^2}{E\pi^2R^2}} \quad \ldots\ldots [3]$$

which is true for both long and short columns, since, when L/R is large, the 1 can be neglected and Euler's equation results, and when L/R is small the term containing it can be neglected, leaving [2].

Equation [3] is the well-known Gordon equation much used in column design. It is plotted on Fig. 26, accompanying this discussion, as well as the curve representing Euler's equation and Tetmajer's results. As will be seen, Gordon's equation is somewhat on the safe side, some allowance being thus made for unpredictable eccentricities in loading.

Using Gordon's equation as an analogy, an equation of the same general type as [3] can be applied to the collapsing of tubes. For very thick tubes, the collapsing pressure can be expressed by equation [C] of the paper in the form

$$P = \frac{K^2 - 1}{2K^2}\,S \quad \ldots\ldots [4]$$

[12] Mechanical Engineer, Standard Oil Development Company, Linden, N. J.

where $K = R_1/R_2$. Combining this with Timoshenko's equation [B] in the paper, to get a form like [3], there results,

$$P = \frac{\dfrac{K^2 - 1}{2K^2}\,S}{1 + \dfrac{2S(1 - m^2)}{E} \cdot \dfrac{K(K + 1)}{(K - 1)^2}} \quad \ldots\ldots [5]$$

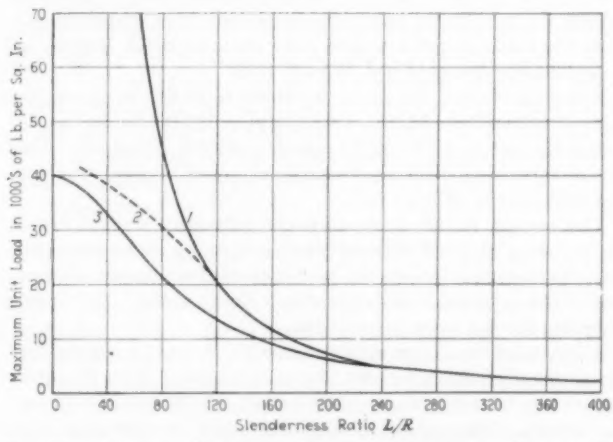


FIG. 26 MAXIMUM UNIT LOAD VERSUS SLENDERNESS RATIO L/R FOR STEEL COLUMNS WITH HINGED ENDS

(1, Euler's equation; 2, Tetmajer's experiments; 3, Gordon's equation.)

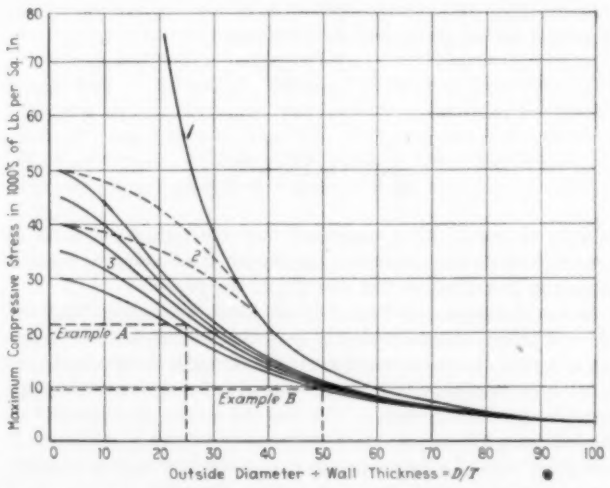


FIG. 27 RELATION BETWEEN MAXIMUM COMPRESSIVE STRESS AND RATIO D/T AT COLLAPSING PRESSURE

(1, from Timoshenko's equation; 2, from experimental results; 3, from analogy with Gordon's equation.)

This is true for thick or thin tubes. For a thick tube or K very large, $K\,\dfrac{(K + 1)}{(K - 1)^2}$ approaches unity. The quantity $\dfrac{2S(1 - m^2)}{E}$ is very small, and the equation reduces to [4]. For a very thin tube or K approaches 1, the second term in the denominator gets very large, the 1 can be neglected, and Timoshenko's equation results.

Fig. 27 of this discussion shows maximum compressive stress at collapsing pressure plotted against ratio D/T, for steel pipe having various yield-point values. As will be seen, the curves plotted from Equation [5] give safer values than the curves composed of the results of tests combined with Timoshenko's equation. This makes some allowance for the reduction due to out-of-roundness and other factors. The difference between the

solid and dotted curves meeting at (0,40), for instance, varies roughly with D/T, as does reduction in stress due to out-of-roundness shown in Fig. 22 of the paper.

As an illustration of this, if we take example *a* on the twelfth page of the paper, the maximum compressive stress given there is 28,800 lb. per sq. in. This has to be reduced by 13.9 per cent on account of 1 per cent out-of-roundness, leaving 25,000 lb. per sq. in. Using Fig. 27 of this discussion, there is obtained 21,500 lb. per sq. in. using curves plotted from Equation [5]. Had the out-of-roundness been 2 per cent, 21,500 lb. per sq. in. would have been obtained in each case.

For example *b* of the paper the stress is 13,400 lb. per sq. in. This is reduced by 24.2 per cent, giving 10,200 lb. per sq. in. Using the curve of Fig. 27, the stress is 9600 lb. per sq. in. This would correspond with a percentage of out-of-roundness of 1.25 per cent instead of 1 per cent.

Comparison is not made with the extremely short tubes of examples *c* and *d*. For these Sturm's equation no longer agrees with Timoshenko's equation, and time does not permit, at present, of the application of the Gordon formula analogy to Sturm's equation for the proper conditions.

From what has been shown, however, it would appear that for tubes of ordinary lengths, the curves plotted from Equation [5] on Fig. 27 of this discussion would be very suitable for practical design. The collapsing pressures could be computed from Equation [5]. Tables of $\frac{(K^2-1)}{K^2}$ and $K\frac{(K+1)}{(K-1)^2}$ would reduce the work to a minimum.

In checking over the examples *a*, *b*, *c*, and *d* of the paper, the tube radii are not given, and the relations

$$\frac{T}{R_1} = \frac{2T}{D}$$

$$\frac{R_1^2 - R_2^2}{2R_1^2} = \frac{2T}{D}\left(1 - \frac{T}{D}\right)$$

have to be used. It is suggested that the right-hand sides of these equations be substituted for the left-hand sides, at present appearing in Equations [E] and [F] of the paper.

In the discussion by Prof. Gilbert Cook it is stated that the effect of length should probably be stated in terms of the quantity $\sqrt{(DT)}$. In an investigation of the strength of vacuum drums it has been necessary to find an expression for the critical length between reinforcing rings. This has led to the development of the following theory, which is believed to be new. Although it may form part of a future paper, it is given here as it is of interest in connection with Professor Cook's discussion.

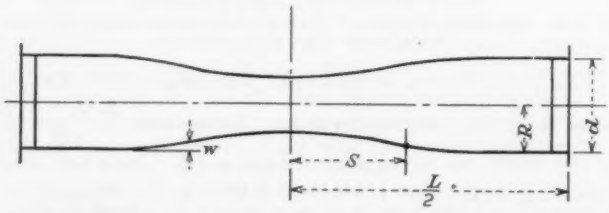


FIG. 28 COLLAPSE OF TUBE OR DRUM BETWEEN REINFORCING RINGS

Fig. 28 represents a length of tube or drum between reinforcing rings. Supposing the rings to be rigid and sufficiently strong to resist the external pressure, it is still possible for the tube to collapse, as indicated in the figure, by bending in the plane of the paper.

Using some well-known bending relations (see "Applied Elasticity," Timoshenko and Lessells, Chap. VI), we are led to the differential equation

$$\frac{d^4w}{ds^4} + 4n^4w = 0$$

and its solution $w = e^{-ns}(a \sin ns + b \cos ns)$ ..... [6]

where s = distance measured from a mid-section of the tube, L = length of the tube, w = deflection, and n = the quantity $\sqrt[4]{\frac{3(1-m^2)}{R_2t_2}}$. R = tube radius, t = thickness, and m = Poisson's ratio.

Differentiating [6] gives the slope anywhere, or,

$$\frac{dw}{ds} = ne^{-ns}[(a-b)\cos ns - (a+b)\sin ns]$$

This is 0 when $s = 0$, from which $a = b$, and it is also 0 when $s = L/2$ or $\sin Ln/2 = 0$. The least root of this is given by $Ln/2 = -\pi$, or $n = 2\pi/L$.

Substituting the expression for n, with Poisson's ratio taken as 0.3,

$$L = 4.9\sqrt{(Rt)} = 3.46\sqrt{(dt)}$$

which is of exactly the form indicated by Professor Cook. L is apparently the critical length below which the tube can only collapse as a ring collapses. Applications of the formula to a number of existing structures show the distance between reinforcing rings to be from $^1/_2$ to $^2/_3$ of L, as calculated above.

GEORGE C. PRIESTER.[13] The authors present a very practical method of determining the stress and the collapsing strength of steel tubes. The writer is pleased to note that the authors have regarded the collapsing strength of a steel tube to be due to two causes depending on the ratio of the diameter of the tube to its thickness. When this ratio is small, the failure is due to local deformations where the unit stress has exceeded the yield point of the material; when this ratio is large, the failure is an elastic failure similar to the failure of a long column. To make use of the analogous buckling phenomenon of columns in determining the collapsing strength of tubes is therefore both rational and logical.

The presentation of sufficient data to calculate the strength of a steel tube of any reasonable length, size, and thickness is a valuable contribution to engineering literature.

H. E. SAUNDERS[14] AND D. F. WINDENBURG.[15] Attention is called to an apparent inconsistency in the paper. Equation [A], due to Sturm, is used for obtaining the collapsing pressures of tubes shorter than the critical length.

For the shorter lengths, the number of lobes into which the circumference of the tube divides itself at the time of collapse is greater than two. Theoretically, the number of lobes N will be integral, and the tube will collapse into that number of lobes which makes the collapsing pressure a minimum. Experimentally, the lobes are not always of equal length, especially when these lengths are measured after collapse.

It is easily possible to obtain the value of N, which will make Equation [A] a minimum by differentiating with respect to N, equating the result to zero, and solving for N. Whence,

---

[13] Professor of Materials of Engineering, University of Minnesota, Minneapolis, Minn.

[14] Senior Assistant to Officer in Charge, U. S. Experimental Model Basin, Navy Yard, Washington, D. C.

[15] Assistant Physicist, U. S. Experimental Model Basin, Navy Yard, Washington, D. C.

$$N = \sqrt{\frac{\pi a}{\frac{L}{R}}} \times \sqrt[4]{\frac{\pi^2 a^2}{\frac{L^2}{R^2}} - m} \quad [7]$$

$$= \sqrt{\frac{\pi a}{2\frac{L}{D}}} \times \sqrt[4]{\frac{\pi^2 a^2}{4\frac{L^2}{D^2}} - m} \quad [7a]$$

In the paper the value of N is determined from existing experimental data. Quoting: "In order to evaluate this term [N], the foregoing data on the collapsing pressure of short tubes were substituted in Sturm's equation, and N was determined for several L/D ratios. The results of these determinations were plotted in a series of intermediate curves, of which Fig. 7 represents the final result. In Fig. 7 the value of N is plotted against the ratio L/D for various D/t ratios. On the basis of the value of N shown in Fig. 7, the collapsing strengths of short tubes having L/D ratios of $1^1/_2$ to 7 were calculated."

The Fig. 7 referred to shows distinctly separate curves for D/t ratios of 30, 40, 50, 60, and 70. From these curves, it is seen that N depends upon t/D, increasing as D/t increases for a constant value of the length-diameter ratio L/D. However, Equation [7a] shows that, in Sturm's equation, N is a function of L/D only and is independent of t/D. That is, according to this equation, two tubes of the same length and diameter will collapse in the same number of lobes, even though one tube is very thin and the other very thick. Evidently, then, Sturm's equation will not predict the correct number of lobes, and hence it cannot safely be used for calculating the collapsing pressures of short tubes. This is especially significant, since the equations developed by Southwell and von Mises give the correct number of lobes in practically every case.

Under Figs. 1, 3, 4, and 5, it is stated that curve 2 (broken line) is "plotted from experimental results." Actually the range of D/T values in the experiments quoted by no means covers the range for which curve 2 is drawn, with the exception of Fig. 4. It would be illuminating to have the experimental spots indicated in the figures. Furthermore, to be exact, there should be a series of curves No. 1 for various L/D ratios.

The choice of the term "collapsing strength," which has been used extensively throughout the paper to denote calculated unit stress at collapse, seems rather unfortunate. It is suggested that some more definite or descriptive term be substituted for it.

With regard to the last paragraph of Section IV and other similar paragraphs, it is felt that the limiting boundaries for the regions controlled by various factors should be more definitely given, if the paper is to be of the maximum benefit to the designer.

It would be instructive to have the reasons behind the statement in section V that "it is entirely logical that N should be represented by either a whole or a compound number." If N is calculated from a formula, it will rarely come out an integral value; on the other hand, why should a column collapse in a fractional number of bays?

Under section VI*a*, it should be pointed out that variation from circular section within the region of one lobe is the controlling feature in premature collapse. Any analysis based upon the behavior of an elliptical ring is almost sure to break down when the number of lobes is greater than two.

HENRY FOWLER.[16] In section VII of the paper, an average variation in wall thickness of 2.5 per cent (for tubes tested by A. O. Smith Corporation) is said to have no effect on the collapsing strength of the tubes; and in Section VI, an average variation in wall thickness of 12 per cent (Stewart's data) is assumed to be of considerably less account than an out-of-roundness figure varying from 0.52 per cent to 2.17 per cent.

While this assumption may be of value in determining the absolute maximum effect due to out-of-roundness, and in this way provide a factor of safety for design purposes, the writer does not think that it is justifiable to state that variation in wall thickness can be left out of account.

For example, using the method set out in the paper, a calculation for a tube with, say, 2 per cent out-of-roundness will indirectly cover any variation in wall thickness coming within ordinary manufacturing tolerances, but any greater variation will not be covered; and a calculation for a perfectly round tube will give an optimistic value for the pressure that can be withstood, as no allowance is made directly or indirectly for variation in wall thickness.

A. ONO.[17] As stated by the authors, the relation between tubes and struts is very close. This can be seen also in the calculation of the critical stress of a tube wall which fails by inelastic collapsing. As to the inelastic buckling of struts, Dr. Th. von Kármán showed clearly by theory and experiment that the stress in a strut may be found from the stress-strain relation observed in compressing the material of the strut under a stress beyond the elastic limit. A similar consideration may be applied to the case of a tube, and a formula deduced may serve for estimating the collapsing stress. As the deduction is very similar to the case of struts, a brief account of the calculation will be sufficient.

Let us denote the outside radius by r and the thickness of the wall by t, respectively. If the tube is truly circular, and the thickness is small as compared with the diameter, the stress is equal to pr/t, p being the intensity of the fluid pressure. If the pressure becomes high, the wall, which is assumed to be infinitely long for the present, undergoes collapsing with an additional bending stress. Denoting the distance from the neutral axis by η, the bending stress is approximately given by the following equation:

$$\sigma = H\eta\left(\frac{1}{\rho} - \frac{1}{r}\right)$$

where H stands for the constant $m^2E/m^2 - 1$ (E the modulus of elasticity and m the reciprocal of Poisson's ratio). Different values of H should be taken for the positive and the negative sides of η, as the stress-strain relation is not the same.

The bending moment per unit length of the tube is $M = \int \sigma\eta d\eta$ taken over a section of the wall. Introducing the foregoing expression for the stress, and using the condition that $\int \sigma d\eta = 0$, we obtain in the same way as in the case of a strut

$$M = \frac{Kt^3}{12}\left(\frac{1}{\rho} - \frac{1}{r}\right)$$

where

$$\frac{1}{\sqrt{K}} = \frac{1}{2}\left(\frac{1}{\sqrt{H}} + \frac{1}{\sqrt{H_1}}\right)$$

H and H_1 refer to the positive and the negative parts of η, respectively.

With this expression for the bending moment, the calculation of the critical pressure may be performed. (See Timoshenko, "Applied Elasticity.") The result of the calculation is, obviously, as follows:

$$p = \frac{K}{4}\frac{t^3}{r^3}$$

[16] Assistant to Vice-President (Research and Development), London Midland and Scottish Railway Company, Derby, England.

[17] Professor, Institute of Strength of Material, Kyushu Imperial University, Fukuoka, Japan.

Accordingly, with $S = pr/t$, the critical stress is given by the formula:

$$S = \frac{K}{4}\frac{t^2}{r^2}$$

This formula gives the critical stress in a long cylindrical tube under a right condition. Any defect such as out-of-roundness, local weakness of material, etc., may of course reduce the critical stress. So the deviation of a test result from the theoretical value may be taken as a sign of imperfection existing in some points.

Authors' Closure

The aim of the paper was twofold: (1) to gather theoretical and experimental material on the collapse of cylinders and endeavor to put it together in a manner which might be useful in the analysis of structures subjected to collapse; and (2) to differentiate in a practical manner between the influence of the physical properties and elastic properties of steel on the behavior of cylinders under external pressure. Had the latter not been aimed at, considerable difficulty would have resulted in dealing with tubes made of steels of widely varying yield points, so as to bring them under a rational analysis. In the use of available data there has been no desire to depreciate the work of other investigators nor to duplicate it except to coordinate it, but rather to recognize and use the careful work which has been done by others. The authors are therefore extremely grateful that such a mass of good experimental data was available for this purpose.

It is not presumed that this paper is anything but a stepping stone to a more complete analysis of the behavior of cylinders subjected to collapsing loads. Nor is it presumed that all of the ideas presented are free from controversy, but to the best of the authors' belief at the present time, the essential ones are. It might be appropriate here to state that the authors have applied the methods outlined to the design of large autoclaves under collapsing pressures and have followed the behavior of such equipment under test to the point of being exceedingly well gratified. The field of the very slender cylinder has been studied also from the experimental angle, and it is hoped that this will yield to an extension of the same general treatment which was applied to the less slender cylinder. This will be presented eventually as a supplement to this paper. There are many data available on slender cylinders which have not been used because they fall mainly outside of the range of D/T ratios presented in the paper. Particularly is this true of the valuable work done by Professor Cook of King's College, London,[18] which is being used for the extension of the method of analysis here applied.

It is believed by the authors that the mathematical treatment of a problem needs tests to supplement it; and that when tests of several investigators can be used which fall into line in a rational manner, a satisfaction is lent to the problem which otherwise could not be obtained. However, this particular problem in its entirety can be only partially analyzed by the available mathematical set-up; and to apply mathematics to the whole range of the problem would seem to be impracticable at the present time.

The derivation of Sturm's equation has not been produced because its author is preparing it for a Doctor's thesis to be presented at the University of Illinois. It is hoped that it will be available to those interested within the very near future. The development of this formula will therefore not be discussed.

Mr. Sturm has introduced the term "plastic yield" in his discussion. The authors have not been able to adjust themselves to such a term. It is believed that at or below the appropriate curves shown for a steel of definite physical properties and with definite dimensions of a tube, no plastic action is possible in the material. In the action of a tube under collapse the possibility of elastic buckling is freely admitted; and except for very stiff tubes the point at which buckling starts is well below the yield point of the material. It is believed that plastic yielding does not occur until the yield point of the material is reached. The authors are quite in agreement with Mr. Sturm's statement of Dr. Westergaard's conception of the division of the field of elastic action.

The authors are grateful to Professor Cook for drawing their attention to the fact that Bryan is responsible for the formula which was labeled as Timoshenko's. In 1913 Professor Cook pointed out that no universal formula had been found to cover the entire range of the behavior of tubes under collapse, and that it was unlikely that such a formula could be devised which would be sufficiently simple to be of practical value.[19] The authors are in entire agreement with him on this point, and the resort to curves has been made in recognition of this fact. Professor Cook also draws attention to the value of bringing mathematical studies into direct application by the use of actual tests. Such tests not only clarify the mathematical results, but also can be used (and have been used by the authors) to draw attention to the degree of perfection available in commercial tubes of today.

Referring to that part of Section II which is quoted in Professor Cook's discussion, "variations in thickness below the average thickness" concern the thin portions of a commercial tube, and "variations above the average thickness" concern the thick portions. In a uniformly round tube subjected to external pressure, the circumferential stress will attain a maximum at the thin portions, and the tube will collapse at a lower pressure than if it were both round and uniformly thick. In an out-of-round tube, the circumferential stress may or may not attain a maximum at the thin portions, depending upon the locations of these portions with respect to the greater diameter of the tube. If the thin portions are at the extremities of this diameter, the circumferential stress will attain a maximum at these points. However, if the thick portions of the tube are at the extremities of the greatest diameter, the circumferential stress may not always attain a maximum at these points, although the bending moment is greatest at these points, but rather at other points where the bending moment is less. Whether or not the circumferential stress will attain a maximum in the thick portions of the tube will depend upon the relative thickness of the thick and thin portions and the respective bending moments at both the thin and thick portions.

With reference to the first part of Professor Cook's closing paragraph, on the possibility of an out-of-round tube's becoming more round under collapsing loads, there can be a serious difference of opinion. This possibility was considered by the authors in conjunction with the test results obtained on tubes which were sized in a die that was as perfectly round as it could be made. In carrying out the sizing operations the portions of the pipe which were slightly thinner were strained a larger amount, and the yield point stress at those points raised above the value it would attain in the thicker portions. The greater amount of elastic recovery at the thinner portions therefore produced a slightly greater curvature at these portions. On subsequent reloading in the testing apparatus, the authors believe the following to be the explanation of the recovery of roundness. As the load was gradually applied, the bending moment at the thinner portions was slightly less than at the thicker portions; but the stress at the former was greater than at the latter, and therefore the curvature of the thinner portions tended to decrease and approach the roundness assumed by the tube when the sizing dies were closed. As the load continually increased there was of course no opportunity

[18] "The Collapse of Short Tubes by External Pressure," *Phil. Mag.*, vol. 28 (1914) pp. 51–56.

[19] "The Resistance of Tubes to Collapse," British Association Rep. (Birmingham) (1913), pp. 213–224.

for further elastic recovery, and the tube eventually collapsed. This seems to be confirmed by the test results as indicated in Fig. 1.

With reference to the last part of Professor Cook's closing paragraph, the tubes tested for collapse by the authors were tested for physical properties after they had been completely fabricated, so that the reported yield points represent the values in the tubes when tested in collapse.

Messrs. Saunders and Windenburg, in their discussion of lobes, state, "Theoretically, the number of lobes N will be integral, and the tube will collapse into that number of lobes which makes the collapsing pressure a minimum." The authors are in agreement with the latter part of this statement, but not with the former part.

Theoretically, the number of lobes will not necessarily be integral, as the lobing is not a discontinuous function of collapsing stress; if it were, the number of symmetrical lobes would abruptly change from one whole number to the next.

The tube will have a natural lobing tendency, and the maximum collapsing pressure that a tube can sustain will be controlled by this natural tendency. It is quite possible that the natural length of a lobe may not divide into the periphery of the pipe an even number of times, and at least there may result one hybrid lobe which represents a whole natural lobe and part of another. This is shown by the unsymmetrical forming of lobes.

The authors have taken the experimental data for short tubes and assumed for each set of dimensions that the value of N is an unknown quantity. Solving for this unknown quantity in Sturm's equation will result in the curves for N which are given in Fig. 7. Fig. 7 was interpolated for values of N, and these latter values were then substituted in Sturm's equation to provide stress values for Fig. 8. Fig. 8 is directly applicable to tube-strength analysis under collapsing pressure when the yield point of the steel, L/D, and D/T are known; and from this figure the collapsing stress can be determined.

It is not clear to the authors how Messrs. Saunders and Windenburg arrive at the conclusion that "Equation [7*a*] shows that, in Sturm's equation, N is a function of L/D only and is independent of t/D." Presumably, in the partial differentiation of Sturm's equation the ratio t^3/D^3 is considered to be constant. But if it is so considered, the deduction, made by Messrs. Saunders and Windenburg that "according to this equation, two tubes of the same length and diameter will collapse in the same number of lobes, even though one tube is very thin and the other very thick," is incorrect. In the partial differentiation t^3/D^3 was made a constant, and this implies that while this ratio is kept constant N will vary only with L/D according to the mathematics shown.

Concerning Sir Henry Fowler's discussion, the authors wish to state that the question of thickness is not considered to have no effect on the collapsing strength. It is believed, however, with the present methods of pipe making (which use fairly uniform thicknesses of steel), that this factor is very much less effective than the question of out-of-roundness, and can be neglected for all practical purposes. This conclusion was reached after a considerable study of tubes which were very round, but in which the thickness difference in the tubes was the maximum tolerance allowed in the plates purchased for tube making.

It is gratifying to read the kind remarks of Dr. Jacobus and Professor Priester.

Professor Ono and Mr. Rhys have both been very helpful in their discussions in clarifying the analogy which the authors have drawn between columns and tubes.

In his reference to Gordon's formula, Mr. Rhys has suggested a more conservative series of curves for tubes. The authors believe, however, that while Gordon's formula is a reliable consideration for columns because of the difficulty in avoiding eccentric loading in service, it does not strictly apply to tubes under collapse. In the very principles applied to tube making, this degree of eccentricity (which is represented by out-of-roundness) need not be present in modern methods of manufacture. The degree of out-of-roundness found in commercial tubes today is very small. In order to get information on out-of-roundness the authors had to study data on tubes made by old tube-manufacturing methods. These methods have been so much improved that today the importance of out-of-roundness is reduced to a minor position. Nevertheless, an evaluation of out-of-roundness in tubes has been made in this paper; and can be applied to processes of manufacture which demand it. The effect of out-of-roundness becomes increasingly important in studying large vessels under collapsing loads, such as autoclaves and the like.

Concerning Professor Stewart's discussion, the authors believe that his generalizations on commercial tubes and on the impracticability of evaluating the effect of out-of-roundness and variations in thickness would be more helpful if they were substantiated by specific test results on modern tubes.

In 1906 he made a splendid contribution to the study of the collapse of tubes and revealed the superiority of the prevailing methods of manufacturing tubes over those existing in 1858 when Professor William Fairbairn published his investigation on riveted tubes.[20] Professor Stewart's tests, however, were made on tubes which were more out-of-round and had greater variations in thickness than is commercially necessary at the present time. It is therefore logical to suppose that his results can be bettered by those on commercial tubes of today.

While industry is approaching nearer to the ideal in tube manufacturing, it is helpful to have some kind of calibration to reveal what this ideal is, and to demonstrate how near commercial tubes can come to this ideal. Professor Carman's tubes were machined to nearly perfect roundness and uniform thickness; and the test results are shown in Fig. 3. Although these tubes cannot be classed as commercial, reference is made to Fig. 1 for tubes selected at random which were made commercially to the extent of over 1,000,000 tons within the past few years. The correlation of results between Carman's almost perfect tubes (Fig. 3) and those shown in Fig. 1 will indicate the degree of perfection possible in commercial tubes at the present time. A close study of the experimental curves in Figs. 1 and 4 will show particularly in the elastic range of the curves the degree to which modern manufacturing methods have progressed toward the ideal tube.

It is believed that the technical points raised by Professor Stewart are covered in the author's remarks bearing on Professor Cook's, Sir Henry Fowler's, and Mr. Rhys's discussions.

---

[20] "The Resistance of Tubes to Collapse," *Philosophical Trans.* of the Royal Society of London, vol. 148 (1858), pp. 389–413.

APM-53-18

Spiral Springs

A New Theory Regarding Their Stress, Strain, and Energy Functions

By J. A. VAN DEN BROEK,[1] ANN ARBOR, MICH.

The paper presents a new theory relative to the stress, strain, and energy functions of spiral springs. This theory differs from the current one in that it takes into account the large deformations implied in the action of spiral springs. In the light of the theory, definite recommendations are made with reference to the design and mounting of spiral springs. It is suggested that springs may be loaded in such a manner as to eliminate all locally overstressed points as well as all friction between leaves. No grease or graphite, therefore, need be employed and the efficiency of spiral springs may thus be increased: (1) By the utilization of space otherwise occupied by a lubricant, from 10 to 20 per cent; (2) by the elimination of all friction between leaves, from 20 to 40 per cent; and (3) by the elimination of locally overstressed points, from 40 to 100 per cent.



FOR a number of years the author has been giving a discussion of spiral springs before his classes at the University of Michigan. He was aware that his analysis differed from those of Föppl, Morley, and Castigliano, and while writing a chapter on Resilience for his text "Elastic Energy Theory" (John Wiley & Sons, Inc., 1931) this disagreement became all the more apparent. Hence this paper is presented with a view to offering opportunity for discussion. The same arguments used in his class work and as given in the text just referred to, are more or less followed in the present discussion. The problem, however, is dealt with in more detail. Furthermore a rational application of the theory to practical problems in design is suggested.

A distinction is made between four essentially different conditions of loading of spiral springs, which include most of those of practical importance at the present time. These four loading conditions are discussed separately under cases I, II, III, and IV.

After having established what appears to be the most desirable condition of loading of the spiral spring, practical means of obtaining it are considered in the light of the theory.

---

[1] Professor of Engineering Mechanics, University of Michigan. Mr. Van den Broek received his B.S. degree from University of Kansas, in 1911, and Ph.D., University of Michigan, 1918. At first he was employed on railway location and railway construction. He was structural designer with the Boston Bridge Works and with the Bridge Department of the Canadian Pacific Railway. He is author of "Effects of Cold Working on the Elastic Properties of Steel," Carnegie Scholarship Memoirs, Vol. 9, reprinted in *Engineering*, July, 1918, and *Zeitschrift für Metallkunde*, Vol. 12, and of "Elastic Energy Theory," John Wiley & Sons, Inc., 1931.

Presented at the National Applied Mechanics Meeting, Purdue University, Lafayette, Ind., June 15 and 16, 1931, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Case I—Outer Extremity of Spring Free to Move in Direction of Radius, but Restrained Against Rotation and Against Motion at Right Angles to Radius.

Given: A beam in the shape of a spiral and loaded only through both its extremities (points A and E, Fig. 1). The beam is a coplanar structure, that is, the center line of the beam and the loads or reactions lie in a plane. The conditions of restraint are such that, as a couple M_1 is applied to the inner extremity, both extremities, points A and E, remain on the original line connecting them. Furthermore the beam at point A is assumed to be restrained against rotation, that is, the tangent at A remains permanently vertical.

To find: The reactions, Q and M_2; the maximum elastic energy the beam is capable of storing; the relation between applied couple M_1, the dimensions of the beam, the elastic constants of the beam, and the maximum stresses and deformations involved.

The problem being a coplanar one, the most general condition of loading is a horizontal and vertical force as well as a couple acting at both extremities, points A and E. The conditions of restraint are such as to insure zero rotation of the tangent at A and zero vertical displacement of one extremity relative to the other. There being no restraint in a horizontal direction, there are no horizontal forces or reactions acting at either point A or point E.

In his book "Elastic Energy Theory," the author follows a

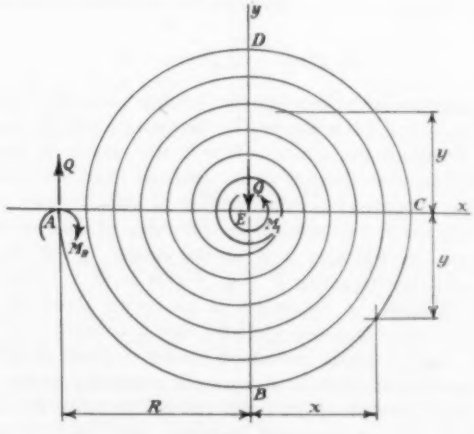


Fig. 1

different line of argument. There he proves that, if the extremities of the spring are restrained sidewise, the horizontal reaction is very small, practically zero, provided the spring may be considered as composed of a large number of concentric coils.

It should be kept in mind that no distinction can be maintained between loads and reactions. The structure shown in Fig. 1 is in equilibrium under the action of all the forces shown. Consideration of the equation $\Sigma F_y = 0$ leads to the conclusion that the vertical reactions at both points A and E are neces-

sarily equal and opposite. Assuming an x-y coordinate system imposed on the system as shown in Fig. 1, the actual bending moment M_q at any point in the structure is given by the expression $M_q = M_1 + Qx$. In this expression the signs are used in conformity with the conventions of analytic geometry. The effect of M_1 throughout the length of the beam is the same in intensity and sign. M_1 causes the curvature of the spring at all points to be increased, and the radius of curvature at all points to be decreased. The effect of Q is to cause change of curvature of the spring in the sense of M_1 for positive values of x, and bending in a sense opposite to M_1 for negative values of x.

The structure is once redundant. That is, the beam may be assumed held at A and loaded with M_1 at E. If the point E is not restrained, the structure would be statically determinate, that is, the moment at all points would be equal to M_1. The structure, however, is restrained at E, that is, the limiting conditions as given at the outset are to the effect that the vertical displacement of point E relative to point A is zero. This may be mathematically expressed as follows:

$$\Delta E_y = 0$$

The presumption is that reaction Q is necessary at points E and A to satisfy these conditions of restraint. In terms of the elastic-energy theory this is expressed as:[2]

$$\Delta E_y = \int \frac{M_p M_q ds}{PEI} = 0$$

P is an auxiliary load applied at E in the direction of ΔE_y. M_p is the moment caused by P. In this case $M_p = Px$, therefore

$$\int \frac{xM_q ds}{EI} = 0$$

E and I, being constant, may be canceled. Thus, $\int xM_q ds = 0$. Substituting the value of Mq from the preceding paragraph, we obtain:

$$M_1 \int x ds + Q \int x^2 ds = 0$$

The integration is to proceed over the entire length of the spring from E to A. $\int x ds$ is the sum of the moments of all elementary lengths ds about the y-axis. This is the length of the spring multiplied by the distance of its centroid from the y-axis. If the spring consists of a large number of coils, it may, without appreciable error, be regarded as a number of concentric rings, the centroid of which is at E on the y-axis. $\int x ds$, therefore, is zero, and consequently $0 + Q \int x^2 ds = 0$. $\int x^2 ds$ is the moment of inertia of the length of the spring about the y-axis. Since this cannot be zero, it follows that Q is zero.

Thus, a spiral spring restrained against vertical displacement as well as against rotation at its outer extremity, point A, and loaded with a couple at its inner extremity, point E, is loaded throughout with a constant moment, and with a constant moment only. The energy due to bending is

$$W = \frac{M^2 ds}{2EI} = \frac{M^2 L}{2EI} \quad \ldots\ldots [1]$$

where L = length of the spring.

To express the elastic energy stored in a spring in terms of stress, we substitute for M its equivalent SI/C. For a spring made of a flat leaf of rectangular cross-section bt,

$$W = \frac{M^2 L}{2EI} = \frac{S^2 btL}{6E} = \frac{S^2 V}{6E} \quad \ldots\ldots [2a]$$

For a spiral spring made of a solid circular wire,

$$W = \frac{M^2 L}{2EI} = \frac{S^2 V}{8E} \quad \ldots\ldots [2b]$$

where V = volume of the spring.

The angular displacement θ of the tangent to one extremity of the spring relative to the tangent to the other extremity is given by the expression

$$\theta = \int \frac{M ds}{EI}$$

M being constant,

$$\theta = \frac{ML}{EI} \quad \ldots\ldots [3]$$

Expressed in terms of stress S instead of M,

$$\theta = \frac{ML}{EI} = \frac{SIL}{CEI} = \frac{SL}{CE} \quad \ldots\ldots [4]$$

and for a spring of rectangular cross-section bt,

$$\theta = \frac{2SL}{Et} \quad \ldots\ldots [5]$$

The stress-strain analysis of spiral springs is interesting because, in the theory of the strength of materials, it exemplifies the elastic behavior of structures, which is at variance with the behavior generally encountered. Two major assumptions commonly underlie the analysis of most problems in strength of materials, assumptions which bear close watching:

1 Material is elastic
2 The principle of superposition holds.

Our analysis of the spiral spring is based on these two assumptions.

In the analysis of redundant structures, where the theory of elasticity finds its widest application, the assumption is generally made that the deformations of the structure are of a relatively small order of magnitude. Without this assumption it could generally not be maintained that the principle of superposition is valid.

In the analysis of spiral springs, large deformations are implied. In disregarding large deformations the analysis is not complete. Any theory we advance, in order to be valid, must hold true for the entire range of stress from zero to the elastic limit. Assuming the spring to be built of elastic material, we have still to consider the question whether or not our analysis holds true for the entire range of loading. In other words, is the second assumption valid in spite of the large deformations in the structure which are involved?

It is proposed to show that, provided certain conditions are satisfied, the foregoing theory is correct and the derived formulas are strictly applicable to spiral-spring analysis or design, regardless of the large deformations involved.

Formulas [1] to [5] give expression to elastic energy stored in and to deformation of spiral springs in terms of loads M, and stress S, volume of spring V, and modulus of elasticity E. It is a very significant fact that all five formulas are independent of the overall dimensions of the spring, particularly are they independent of the radius R (Fig. 1). That is, the formulas hold true in both close-coiled and loose-coiled springs, and in springs both heavily and lightly loaded, with equal validity.

[2] For a detailed proof of the equations here employed, the author refers to his text "Elastic Energy Theory," John Wiley & Sons, Inc., 1931, or to a pamphlet published by him in Ann Arbor entitled "Elementary Elastic Energy Theory."

This is true provided the limiting conditions of restraint as stated at the outset are satisfied.

Under case III it will be shown that this ideal condition of loading is approached fairly satisfactorily in spiral springs mounted in agreement with common practice. Later, under "Suggested Improvements," it will be shown how the ideal condition of loading, discussed under case I, may be practically realized.

CASE II—OUTER EXTREMITY OF SPRING FREE TO MOVE IN DIRECTION OF RADIUS AND TO ROTATE, BUT RESTRAINED AGAINST MOTION AT RIGHT ANGLES TO RADIUS.

Given: A beam in the shape of a spiral and loaded only through both its extremities (points A and E, Fig. 2). The beam is a coplanar structure, the center line of the beam and the loads or reactions lie in a plane. The conditions of restraint are such that, as a couple M is applied to the inner extremity, both extremities, points A and E, remain on the line AE.

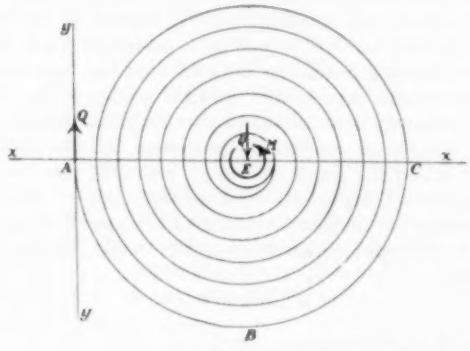


FIG. 2

To find: The maximum elastic energy the beam is capable of storing.

Case II differs from case I essentially in the mode of support of the outer extremity of the spring, point A. The end A of the beam is restrained to move on the line AE. The tangent to the beam at the point is free to rotate. Furthermore, point A is free to move in the direction AE.

In his book "Elastic Energy Theory," the author proves that, in case point A is restrained in a horizontal direction as well as in a vertical direction, the horizontal component of the reaction at either A or E is zero, provided the spring may be considered as composed of a large number of concentric rings. This proof is omitted here as it is not essential to the present argument.

Under the conditions of loading, as stipulated, the beam is statically determinate, or

$$M = QR$$

If we conceive of an x-y coordinate system imposed, with the origin at A and with the x-axis horizontal and passing through the center point E (Fig. 2), then the general expression for M_q, the bending moment at any point, is:

$$M_q = Qx$$

The elastic energy stored in the spring is:

$$W = \int \frac{M_q^2 ds}{2EI} = \frac{Q^2}{2EI} \int x^2 ds$$

The term $\int x^2 ds$ is the moment of inertia of the spring about the y-axis. This moment of inertia is equal to a moment about an axis parallel to the y-axis and passing through the centroid, plus the length times the square of the distance between the two axes. The moment of inertia of the spring about an axis through the center of gravity is equal to the length of the spring times the radius of gyration squared. Provided the spring consists of a large number of coils, its radius of gyration squared will, to all intents and purposes, be equal to the square of the radius of gyration of a circular area, and is thus equal to $R^2/4$. Therefore

$$\int x^2 ds = LR^2 + \frac{LR^2}{4} = \frac{5}{4} LR^2$$

[The square of the radius of gyration of twenty concentric rings varying in radius uniformly by one inch (from a ring with a radius of 1 in. to one with a radius of 20 in.), about a line tangent to the outer ring is 505 in.^{2} instead of 500 in.^{2} as the formula would give. The error here involved in this assumption is therefore of the order of magnitude of 1 per cent.]

The total elastic energy stored in the spring is:

$$W = \frac{5Q^2R^3L}{8EI} \quad \ldots\ldots\ldots\ldots\ldots\ldots [6]$$

The maximum bending moment occurs at C and is equal to $2QR$. The maximum stress in a spring of rectangular cross-section bt is:

$$S_{max.} = \frac{Mc}{I} = \frac{6M}{bt^2} = \frac{12QR}{bt^2}$$

Thus $QR = Sbt^2/12$.

The elastic energy stored in the spring, expressed in terms of this maximum stress at point C, is:

$$W = \frac{5btLS^2}{96E} = \frac{5VS^2}{96E} \quad \ldots\ldots\ldots\ldots\ldots\ldots [7]$$

Assuming the spring to be sufficiently stiff and rigid for the loading conditions to remain unaltered until the elastic-limit stress at point C is attained, the maximum elastic energy that can be stored in the spring is:

$$W_1 = \frac{5VS_1^2}{96E}$$

where S_1 is elastic-limit stress at point C. The maximum elastic energy that can be stored in a spring under the conditions of loading discussed under case I, was expressed by Formula [2], and is equal to

$$W = \frac{VS_1^2}{6E} = \frac{16VS_1^2}{96E}$$

The mounting of a spiral spring in agreement with the conditions set forth under case I is thus seen to be 16/5 times as effective (as far as the capacity for storing elastic energy is concerned) as mounting it in agreement with those set forth under case II.

NOTE: In the analysis of the problem discussed in this example various authorities commonly disagree in their conclusions, although their premises are essentially the same. Thus, August Föppl, in his "Vorlesungen über die Technische Mechanik," and Morley in his "Strength of Materials," give the value $\frac{4}{96} \frac{S_1^2 V}{E}$ instead of $\frac{5}{96} \frac{S_1^2 V}{E}$ for the elastic energy stored in a spiral spring under a condition of loading as here assumed. It would seem that Föppl made the error of equating the angular displacement at the arbor to the angular change between the two extremities, thus ignoring the fact that the tangent to the outer extremity also rotates. Morley in part of his analysis assumes "the spring at A to remain sensibly normal to CA," which is a contradiction of the condition that the tangent to the spring at A is free to rotate.

S. Timoshenko in his "Strength of Materials," vol. II, page 445, arrives at an expression for energy, $W = \frac{M^2 L}{2EI}$, which is the same

as that found in case I for a spring with the outer extremity restrained against rotation. He further concludes that the maximum moment, namely, $M_c = 2QR$, is at C. Certainly these two conditions cannot coexist.

Castigliano in "Theorie der Biegungs und Torsions-Federn," 1888, page 26, gives a formidable analysis without, however, clearly stating his premises or end conditions. He carefully includes the shear forces and direct compression, only to ignore them later on. His analysis applies to a condition of loading and support similar to but not identical with those of case I. His limiting conditions are not expressly stated. Various conditions of loading or support of spiral springs he does not discuss. In common with subsequent writers, he fails properly to evaluate or discuss changed conditions as the loading progresses and the outline of the spring changes.

Not only is the maximum elastic energy in a spiral spring stored under conditions of loading and support as here stipulated, at most less than one-third the amount possible under conditions as outlined in case I, but in the majority of springs the conditions of loading and support will be radically changed long before

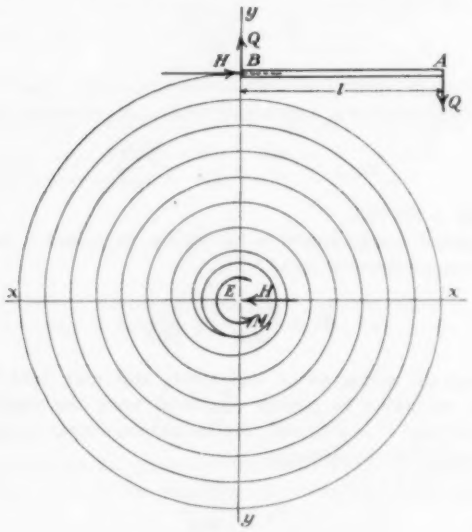


FIG. 3

the elastic-limit stress at point C is reached. Spiral springs such as mainsprings in clocks and phonographs are so flexible that, if loaded and supported in the manner indicated in Fig. 2, they readily deform, the leaves come in touch with each other, and finally all come to bear against the arbor (see Fig. 5). The instant two leaves are brought in contact the loading conditions as stipulated at the outset of this example are violated and the theory developed on the assumption that the spring is in the shape of a spiral with all leaves separated becomes invalid.

In his text on "Elastic Energy Theory," page 187, the author computes that in case of a clock spring of maximum radius $R = 5$ in., length $L = 160$ in., width $b = 1$ in., and thickness $d = 0.022$ in., consisting of 14 coils (see Fig. 8) the outer leaves of which are separated 1 in., a force of $^1/_{15}$ lb. is sufficient to bring the outer leaves in contact with each other. In a test of a spring of these dimensions, the spring being mounted on a drawing board, the friction factor involved made it impossible to check the value of $^1/_{15}$ lb. exactly. It was found, however, that the force necessary to bring the outer leaves in contact under a condition of loading as here assumed (Fig. 2) is of the order of magnitude of somewhat less than one ounce. With a load Q of $^1/_{15}$ lb. in the spring, the elastic energy stored is $^1/_8$ of 1 per cent of the energy that can be stored if the loading conditions were kept constant, or $^1/_{24}$ of 1 per cent of the energy that may be stored if the loading corresponds to that outlined in case I.

The ultimate condition, when the loading assumed in case II is maintained until all the leaves bear on the arbor, is probably the most important and representative condition of loading and support of spiral springs. This case is discussed in detail under case III (*d*).

CASE III—(*a*) OUTER EXTREMITY OF SPRING LOADED BY TWO EQUAL AND OPPOSITE FORCES APPLIED TO A RIGID BAR; (*b*) LOADED BY TWO EQUAL AND OPPOSITE FORCES APPLIED TO LAST QUARTER-BEND OF SPRING; (*c*) SPRING PINNED AT OUTER EXTREMITY AND SUPPORTED BY ROLLER $^1/_4$ TURN AWAY; (*d*) SPRING PIN-SUPPORTED AT EXTREMITY AND BEARING ON ARBOR.

Under case III it is proposed to discuss several analogous cases of loading.

(*a*) *Given:* A coplanar beam in the shape of a spiral. The outer end of the spiral beam (Fig. 3), point B, is clamped to a rigid bar BA. Compared with a flexible spring, the bar may be assumed of infinite rigidity. The spring is loaded with a moment M_1 and a horizontal force H at its inner extremity, point E, and with two equal and opposite forces Q applied to the two extremities of the bar AB as well as a horizontal force H along AB. The limiting conditions are such that the line connecting B and E remains permanently at right angles to the bar AB. Point E, however, is permitted to alter its linear distance BE.

If we assume an x-y coordinate system introduced with its origin at E and the y-axis coincident with BE, then the actual bending moment M_q is expressed as $M_q = M_1 - Hy$. If bar AB is assumed to be held permanently horizontal, then the fact that E is restrained to move along the y-axis BE may be expressed as

$$\Delta E_x = 0$$

In terms of elastic energy

$$\Delta E_x = \int \frac{M_p M_q ds}{PEI}$$

M_p is the moment in the beam caused by an auxiliary load P applied at E in the direction x. M_p in the above equation therefore is $M_p = Py$. $\Delta E_x = 0$ then becomes

$$\int \frac{PyM_q ds}{PEI} = \int \frac{yM_q ds}{EI} = 0$$

Substituting the expression for Mq and canceling EI, we obtain

$$M_1 \int y ds - H \int y^2 ds = 0$$

It has been previously shown, provided the spiral beam may be regarded as a number of concentric rings, that $\int x ds = 0$. Therefore

$$H \int y^2 ds = 0$$

Since $\int y^2 ds$ is the moment of inertia of the outline of the beam about the x-axis, and since this cannot be zero, it follows that

$$H = 0$$

Therefore, $M_1 = QL$. Thus the beam is loaded with equal and opposite moments at its two extremities. Therefore, the condition of loading here discussed is identical with that discussed under case I.

(*b*) *Given:* The same spiral beam discussed under case III (*a*) (Fig. 3) with the rigid bar AB replaced by a quarter-bend of the flexible beam (Fig. 4).

To find: The stress-strain relationship of the beam as a function of the loads.

In the analysis of the previous example, case III (*a*), we assumed bar AB of infinite rigidity as compared with the flexible

spiral beam. This, in turn, insured the x-y coordinate system to be permanently fixed with the y-axis vertical and the x-axis horizontal.

Suppose the bar AB (Fig. 3) is not rigid. For example, AB may be considered as the end of the spiral beam. Then, if points A and B are assumed held on a straight horizontal line,

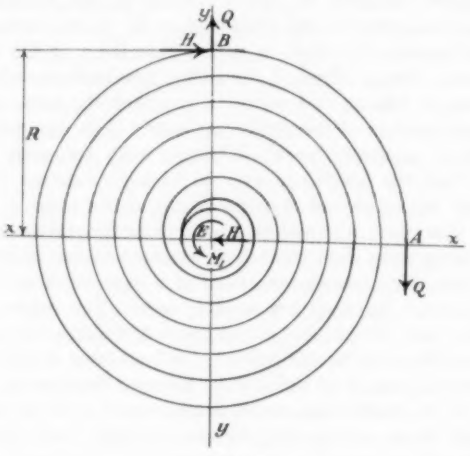


FIG. 4

the tangent to the beam at point B will rotate somewhat as the structure is loaded. If we define the y-axis as being permanently perpendicular to the tangent to the beam at point B and define the loads Q and H as permanently parallel to the y- and x-axes, respectively, then, for any stage of loading, the analysis given under case III (a) holds good.

If, finally, the outer end of the spiral beam be considered a quarter-bend of a circle and loaded with two equal and opposite forces at points A and B (Fig. 4) with a horizontal force at point B, and a moment and horizontal force at E, all the foregoing arguments of case III (b) apply here with the same force.

The assumption that the x-y coordinate system be allowed to rotate slightly and the direction of the forces be allowed to rotate with it so as to be always parallel to the x- and y-axes, is possibly an unusual condition, and in this case a hypothetical one. The arguments, however, are thought to be entirely sound. This condition of loading has been analyzed as a basis for the discussion of the following two conditions which have practical significance.

(c) *Given:* A spiral spring mounted on an arbor at E, pinned at A, and supported on a roller at B (Fig. 4). The reaction at B is in the direction BE. Point B is free to move on the line BE.

To find: The stress-strain relationship of the beam as a function of the loads.

In cases III (a) and III (b), points A and B were assumed fixed in space while point E was restrained to move on the y-axis. In case III (a) the y-axis is permanently vertical, while in case III (b) it is allowed to rotate slightly to compensate for the curvature induced in the loading end of the spring BA. In these cases the spiral spring is loaded with a constant moment, which is the same condition of loading discussed under case I. A constant moment means constant change of curvature. Therefore a spring which has the form of a number of essentially concentric rings to begin with maintains this form as the loading progresses. The rings become smaller in diameter and increase in number, but essentially they remain concentric rings throughout the loading process. Point E in cases III (a) and III (b) thus approaches point B along the y-axis. In the present case (with point E fixed in space) the same condition of loading will prevail, provided bar AB, Fig. 3, is permitted to approach E, remaining at all times parallel to its original position, or provided point B, Fig. 4, is permitted to approach E. If the portion BA, Fig. 4, is a flexible beam, the x-y coordinate system must in this instance be permitted to adjust itself to the change of curvature in the portion AB of the beam. The loading condition here discussed may be obtained by supplying a roller-bearing support at B and allowing it to move freely to the left as the spring is loaded. As the loading progresses, the direction of the reaction at A will change so as to be always parallel to BE, while B moves slightly to the left as well as toward E. With equal parallel, and opposite forces applied at A and E, the spring remains open as the loading progresses.

(d) *Given:* A flexible spiral spring mounted on an arbor at E and its outer extremity A attached to a string which is supported at E. (See Figs. 2 and 5.)

To find: The stress-strain relationships of the spring as a function of the loads and reactions.

In cases III (a) and III (b) the points A and B were assumed fixed in space and point E was free to approach B along the line EB.

In case III (c) the point E was fixed in space and point B was subjected to a reaction equal and opposite to the reaction at A.

In the case under discussion, the spring is loaded as given under case II. Case II shows that, with flexible springs, the leaves

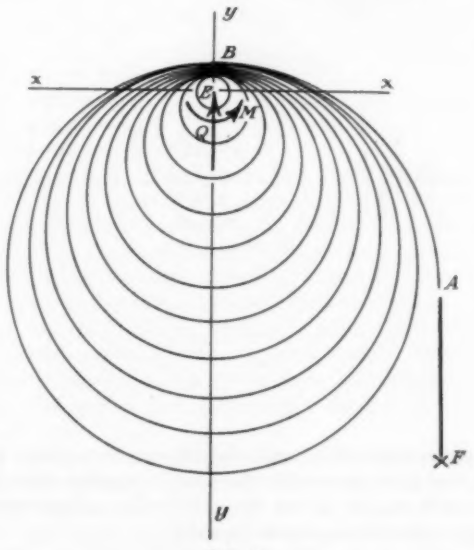


FIG. 5

soon come in contact with each other and the spring assumes the shape as shown in Fig. 5. The spring then consists of a number of tangential rings instead of a number of concentric rings. A spring loaded as illustrated in Fig. 5 presents essentially the same condition as the one discussed under case III (c). The roller-bearing reaction of Fig. 4 is replaced by a reaction supplied by the arbor in Fig. 5. Throughout the entire length of the spring, except for the quarter-bend BA and except for the friction between leaves on the arbor, the spring is loaded with a constant moment. As the tangent to the curved beam BA at point B is somewhat displaced (displacement due to the bending in the quarter-bend AB), point B shifts somewhat to the left and the axis of symmetry, shown as the y-axis in Fig. 5, turns somewhat in a counterclockwise direction. These changes, however, as discussed under case III (c), do not materially affect the theoretical analysis of the case under consideration. The friction

factor also, strange as it may appear, is more of a practical than a theoretical stumbling block. On any intermediate leaves, the pressure on the top and bottom of any one leaf is practically the same. The difference between the two pressures is of the order of magnitude computed under case II. There is thus on each individual leaf a frictional drag, tending to propel the leaf, practically equal to another drag tending to retard it. This holds true for all leaves except the outer one. This does not mean, however, that the friction is not objectionable. It does argue that the friction factor does not invalidate our conclusion to the effect that the loading (Fig. 5), in so far as storing of elastic energy is concerned, is essentially the same as that represented by Fig. 4, which, in turn, is substantially the equivalent of the loading represented by Fig. 1. (See also the concluding paragraph under case IV.)

August Föppl, in his "Vorlesungen über Technische Mechanik," 1919, Vol. III, page 227, writes: "Under a condition of a fixed-end support, the spring is more favorably loaded than is the case when the outer extremity is pin-connected. When there is doubt as to whether or not the condition of fixity of the support can be relied upon, our computations should be based on the formulas derived for the loading condition with pin-connected end. In the first four editions I have therefore discussed only the case of a pin-connected end; since this led to misunderstandings I have now also included the analysis of springs with fixed support. This is not done, however, with the intent to recommend an analysis differing from those formerly offered, but it is given with the implied warning that the conclusions will usually be too favorable when a fixed position of the outer end of the spring is assumed."

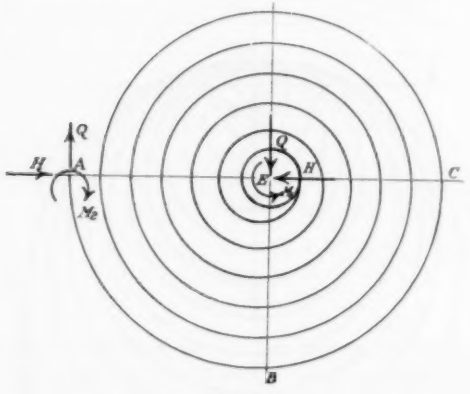


FIG. 6

The author feels that Föppl was one of the clearest and soundest writers on mechanics of all time, but in this instance he takes exception to Föppl's conclusions. Instead of being difficult of realization, the conditions of support analyzed under case III (*d*), with all leaves of the spring bearing on each other and on the arbor, constitute the most common kind of loading of spiral springs we appear to have. Except for a friction factor as the leaves rub over each other, this condition of loading and support is essentially equivalent to a condition of loading with fixed end.

CASE IV—OUTER EXTREMITY OF SPRING FIXED, COMPLETELY RESTRAINING IT AGAINST LINEAR OR ANGULAR MOTION.

Given: A beam in the shape of a spiral and loaded only through its extremities (points A and E, Fig. 6). The conditions of restraint are such that, as a couple M_1 is applied to the inner extremity, both extremities remain in the same relative position. Furthermore the outer extremity is restrained against rotation. Point A is fixed.

To find: The energy stored, and the stresses in the beam.

Case IV is identical with case I except for the restraint of point A in the direction AE. In this instance point A is what is technically known as completely fixed. Point E is fixed as to position. Under this condition of restraint, the structure has one degree of freedom less than it has under the loading and support discussed in case I. While in case I there is no horizontal reaction at either point A or E, in this instance we have to assume that there is such a reaction. In his treatise on "Elastic Energy Theory" the author proves that under these conditions of loading and restraint, provided the beam consists of a large number of essentially concentric coils, the horizontal reaction is practically zero. While in that argument it was implied that the conditions were to be kept constant in order that they might prevail throughout the entire range of stress, that is, that point A would be moved in as the spring is wound up, in the present case point A is assumed to be rigidly fixed. To be sure, for a spring composed of a large number of coils, the horizontal reactions are sensibly zero. The mathematical proof is a very simple and direct one. The experimental verification of the proof is still simpler. All one need do is to apply a horizontal force H at point A on a spring mounted as shown in Fig. 6, to realize how much displacement is produced with very little force, and by the reverse argument, how very little force would be required to overcome any tendency toward lateral displacement.

Under case I it has been demonstrated how, if point A is allowed to move laterally, the beam is subject to a constant moment throughout. As this moment increases in magnitude the beam is curved more acutely throughout its length and more concentric rings are formed, which, however, are closer together. The overall diameter of the spring decreases. If the spring is rigidly fixed at A this decrease of the overall diameter will induce a horizontal force at A. In other words, the decrease in the overall diameter will be accomplished by a double shift of point C toward E instead of equal shifts of points A and C.

At first glance it may appear that we are indulging in loose reasoning. In one case we argue that under certain conditions of restraint there is no horizontal reaction, and in the next argument that there is. The point is that the horizontal component of the reaction at A is in the nature of a differential of a higher order. So long as deformations of the structure are finite but small, the reaction H is of a differential magnitude. When we consider a spiral spring of, say, twelve coils, as being wound up and changed to a spring of thirty coils, the deformations are no longer small finite deformations but are finite and very large.

It is comparatively simple to develop with quite a high degree of accuracy an expression for the value of H under these conditions of loading. We shall forego that, however, and content ourselves with proving that a manner of support as here stipulated—complete restraining of the spring at point A—has no advantage, from the point of view of storing maximum elastic energy, over a pin-connected support as discussed under case III. In fact, it is easily conceivable that with a fixed support the loading conditions may be less favorable than they are with a pin support at A.

It was shown in case III (*d*) that with a pin support, the loading, except for friction between the leaves, is as efficient as the condition of loading comprising constant bending moment throughout the length of the structure except for the last quarter-bend. This same condition will ultimately prevail with the beam completely restrained at point A. The disadvantage in the latter mode of support lies in the fact that as the spring is wound with a pin support at A the entire spring from B to E is uniformly loaded, only the quarter-bend AB is subjected to a non-uniform loading varying from zero at A to a maximum at B. In the for-

mer case, however, with a fixed support at A, as the spring is wound, the entire spring from B to E, after all leaves are bearing on the arbor, is again loaded uniformly with the exception of the half-bend AC, which is subject to a reversal of bending moment. If the spring is wound tight, this reversed bending moment may well exceed the constant moment in the spring between points B and E.

We offer one more evidence to the effect that, from the point of view of maximum elastic energy stored, a pin support at A, as in case III (d), is as efficient as a restrained support at A, case I, and more effective than a fixed support at A, case IV, except for friction involved as the leaves of a spring slide over each other on the arbor and except for the final part AB of the spring.

Consider the winding of a spiral spring under the two conditions of loading and support as given in case I and case III (d), support restrained against rotation at A and pin support at A. The initial condition is one of zero stress throughout in both cases. The final condition is a spring wound tight on the arbor. The final conditions of the spring wound in the two ways will be practically identical. This being the case, the amounts of elastic energy stored in both cases must be practically identical. Furthermore, the number of turns required to reach both final conditions are identical. What we cannot prove by this argument is the rate of increase of stress under these two different ways of loading. This has been discussed under cases III (a) and III (b) which point to the fact that the rate of increase in both cases is essentially constant.

THEORY AND PRACTICE

The four cases here discussed may not cover the complete range of possibilities, but it is thought they cover the most essential conditions of loading and support of spiral springs. The author has quoted the German authority Föppl, with whose conclusions he disagrees. To quote one more authority, Arthur Morley, in his "Strength of Materials," 1923, page 399, says: "Proof resilience $= \frac{1}{24}\frac{f^2}{E} \times V$, which is only $^1/_4$ of that of a closely wound helical spring subject to axial twist, where all the material is subjected to the maximum bending moment, instead of a bending . . . etc."

The current theory of spiral springs appears to be one more case in which practice and theory disagree—with practice carrying off the honors. In spite of the current theory, practice has continued to develop and to use spiral springs for the purpose of storing mechanical energy in a great variety of mechanisms. It would seem that if helical springs subjected to torsion about the axis of the coil were four times as efficient as spiral springs, surely industry would have made more extensive use of helical springs thus loaded. Our theory points to the conclusion that a spiral spring, from the point of view of storing elastic energy, is exactly as efficient as a helical spring twisted about the axis of the coil. If we compare a spiral spring made of a flat leaf of rectangular cross-section and loaded in the usual manner with a helical spring made of a solid round wire and twisted about its axis, we shall find that the energy stored in the former is 33 per cent greater than that stored in the latter [see formulas 2(a) and 2(b)].

A helical spring loaded in the usual manner, that is, with two equal and opposite forces acting in line with the axis of the coil, functions essentially as a long, slender shaft compacted into a small space with each element of its length loaded with a constant torque. Similarly, a spiral spring is essentially a long, slender beam compacted into a small space with each element of its length loaded with a constant moment.

The difference between the elastic energy stored in a cantilever beam subject to constant bending and that of a cantilever beam with a concentrated load at its unsupported end is a qualitative illustration of the difference between the loading condition discussed under case I and that of case II.

It is argued that practice in regard to spiral springs is far better than current theory would lead to suppose. It does not follow, however, that there is no room for improvement in current practice.

In venturing to speak of practical problems the author wishes to make it clear that he does not lay claim to an extensive practical experience with spiral springs. Except for a study of his own watch, discussions with a watch repairer, and a phonograph repair man, and a few experiments with springs mounted on drawing boards for experimental purposes, he has no practical appreciation of the problem.

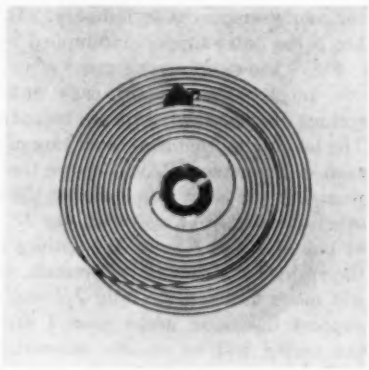


Fig. 7 Support of Hairspring in a Watch

Suggested Improvements in Mounting Spiral Springs

In discussing practical aspects, then, the author will limit his remarks to the following assumptions: (1) That the ideal loading of a spiral spring is such that the angular deformation is accurately proportional to the torque (hairspring in chronometer), and (2) that the spring must store a maximum amount of energy and must be free from discontinuous operation (mainspring in watch). It is proposed to discuss the best manner of satisfying these requirements.

One condition is essential, namely, the spring must operate without the leaves coming in contact with each other. The one and only way to obtain this result is to satisfy the limiting conditions of support and restraint as discussed under case I. The outer extremity of the spring is to be restrained against rotation. Its motion toward the center must, however, be unrestrained.

Föppl does not believe that complete fixity of the outer support can be attained. One of the largest American manufacturers of watches attempts it. It has been pointed out under case IV that in the author's opinion a pin-connected support is superior to a completely restrained support. The cheaper watches have the hairspring fixed at point A with the result that, as the hairspring is loaded and unloaded, it does not expand and contract about the center of support, but the leaves on the opposite side of the arbor crowd in on each other as the spring is loaded (see case IV). The better watches have the hairspring supported as shown in Fig. 7. This is a partially successful attempt to attain the limiting conditions of support as stated in case I. The ideal conditions of support may be attained in a number of ways.

Fig. 8 shows the outer extremity of a spiral spring clamped to a rigid bar. In order that the rigid bar may transmit a moment M to the spring at point A, so as to keep the tangent of the spring at the point unchanged, the bar must be loaded by a couple—two equal and opposite forces.

The rigid bar may be supported against the arbor at one point and against a bearing just beyond the support A as shown in Fig. 8. The bearings of the bar against these two supports

will supply the required couple, and the spring will be loaded and supported in agreement with conditions stipulated under case I. As the spring is being wound, more and more coils will be created. All coils will remain concentric rings, which, in a properly designed spring, will not touch each other until the spring is completely wound. The friction therefore between leaves is eliminated. There remains, however, a slight friction at the two points of contact of the bar. This may be eliminated or reduced to an inappreciable minimum in one of the following two ways.

It may be noted that the points of contact on the bar and the arbor move always in the same sense. By giving the arbor at the point of contact between itself and the bar the correct diameter, the two will move together and there will be no friction at that point. The bearing outside the spring, beyond point A, may of course be made quite free from friction by standard means commonly employed in industry. In the photograph shown in Fig. 8 the outer support is supplied by a string.

Fig. 9 shows an arrangement of two spiral springs which lend one another the support each individually needs. The two springs are identical, and are mounted on two separate arbors. The loading mechanism, consisting of two cog-wheels that mesh, each of diameter EH, will insure the fact that both springs are wound at the same rate. Either the springs are welded to each other at point A, or a straight bar JK connects the spring rigidly at points J and K. As the springs are wound, the tangent to the springs at J and K will remain unaltered. Points J and K will move in along the radii JH and KE, and the conditions of support discussed under case I are satisfied. Every inch of the spring will be equally stressed. The spring thus will be loaded throughout its length in the most efficient manner obtainable with structures subjected to bending. The operation will be very smooth as there will be no bearing on the arbor, nor will there be friction between the leaves of the springs, provided the latter are properly designed and manufactured. Springs mounted in this manner may be built into a tight box without the use of objectionable grease or graphite. There are no locally overstressed points. The possibility of breakage is thus reduced to a minimum.

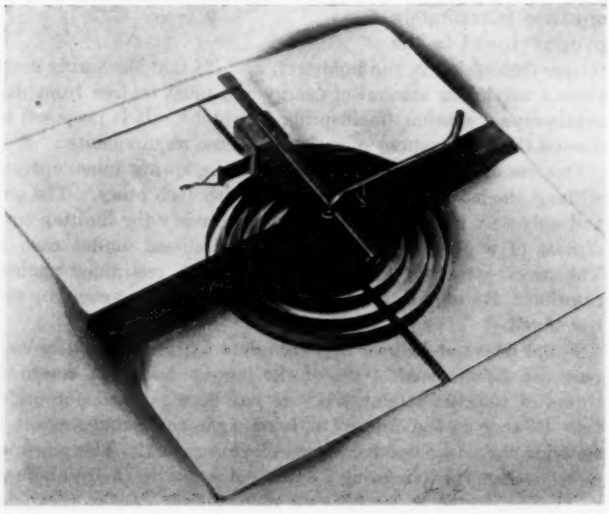


Fig. 8

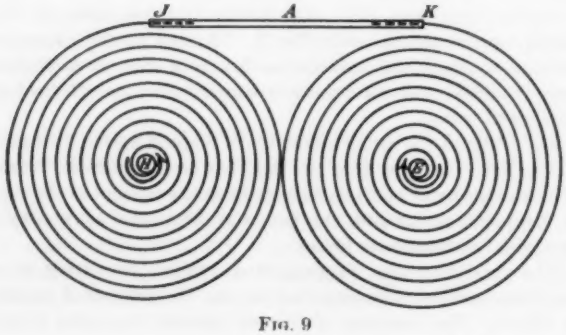
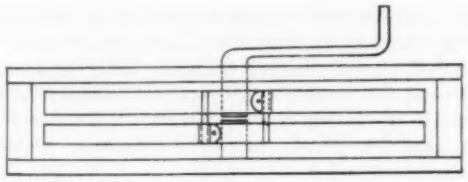


Fig. 9



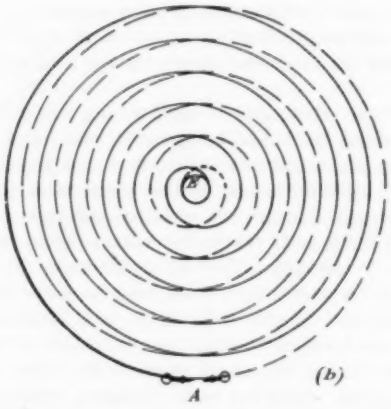


Fig. 10 Two Springs Mounted on Separate Arbors Having a Common Axis

[In the plan (*b*) the frame for mounting the springs is not shown; bottom spring is shown by broken line; top spring by solid line. A flat, loose strip of metal (not shown in either plan or elevation) may be necessary to prevent the springs from settling into each other.]

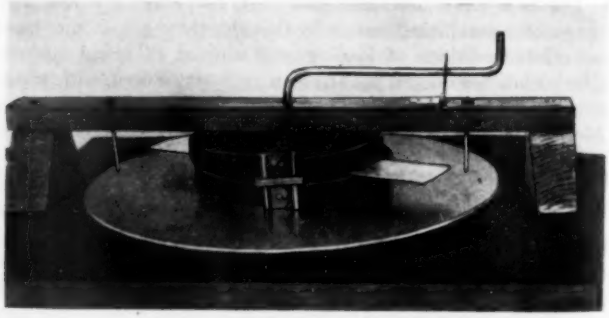


Fig. 11 View of Springs Shown in Fig. 10

The author can see only two possible objections to this mode of mounting power springs: (1) A little space, just below point A, is inevitably wasted. (2) The simple arrangement of supplying power through the arbor and of taking it off the drum uninterruptedly cannot in this case, in which the drum is other than circular in shape, be employed.

Figs. 10 and 11 show two springs mounted in a manner apparently to satisfy perfectly the conditions of support and loading discussed under case I. The springs are each mounted on a separate arbor. The two arbors turn on one and the same axis. The bottom spring is shown in Fig. 10(*b*) by a broken line and the top one by a solid line.

The winding of spiral springs is a matter of relative motion. The arbor at E (Fig. 1) may be turned and point A restrained, or else the arbor at E may be kept stationary and point A be made to move around the arbor, keeping the tangent to the spring at A at all times normal to the radius connecting A to E. In the device shown in Fig. 10, the top spring is loaded in the first-named manner, and the bottom one in the second manner. As the arbor of the top spring makes one complete revolution, point A makes one-half revolution. The top and bottom springs are loaded identically, and every element of the spring is subjected to the same stress conditions. There are no locally overstrained points. At the connections, at either one of the arbors, or at point A, the variations in stress are of such a nature as to reduce stresses for a very small distance. This device permits power being supplied to one arbor and uninterruptedly taken off the other. Grease and graphite can be entirely eliminated as there will be no rubbing of leaves over each other, provided the springs are properly designed and manufactured. This manner of mounting spiral springs suggests itself not only with regard to power springs, but also in reference to hairsprings in chronometers. A spiral spring takes the place of the one loop brought over as shown on Fig. 7. What the final coil in Fig. 7 is intended to accomplish, which, in Fig. 7, is attained with only partial success, is satisfactorily achieved by mounting spiral springs as indicated in Fig. 10. The device for regulating the period of oscillation in an arrangement of this kind would probably best be applied to the inner end of the bottom spring near the fixed arbor.

Fig. 11 is a photograph of two springs mounted in the manner here described. As the handle is turned two complete revolutions the connection between the springs makes one revolution. The loading mechanism may of course be designed to wind both arbors at the same rate in opposite directions. In that case the connection between the springs travels on a radius AE [Fig. 10(b)].

Fig. 11 shows a piece of tin slipped between the springs to act as a spacer, to prevent the leaves of the two springs from interlocking. In winding, the springs shown in Fig. 10(a) remain open and frictionless until completely wound.

DESIGN OF SPIRAL SPRINGS

Throughout the paper the author has occasionally qualified his remarks to the effect that the spring in question be properly mounted and designed. So far, the discussion has concerned itself only with the mounting and loading of springs. It remains now to consider their shaping and proportioning so that they may function most effectively.

To obtain proportionality between load and deformation (the hairspring in a watch, for example), it is important that the spring consist of a large number of coils approaching the equivalent of a number of concentric rings. For the best functioning of a hairspring in a balance wheel, the mounting should be in agreement with that discussed under case I and illustrated by Figs. 1, 8, and 10.

The design of a spiral spring with a view to storing a maximum amount of energy is not so simple a matter. A spiral spring is a self-locking device. To attain the highest degree of efficiency, two conditions must be satisfied. There should be no locally overstressed portions of the spring, and the stresses throughout the entire length of the spring should be as great as possible. Formula [2], $W = S^2V/6E$, expresses the energy in a spring in terms of the square of the stresses, the volume, and the modulus of elasticity. This formula applies to an open spring consisting essentially of a number of concentric rings.

If, when the spring is wound tight, the stress for any part of the spring is less than the safe working stress, the spring loses in efficiency. This is an important consideration, as the energy

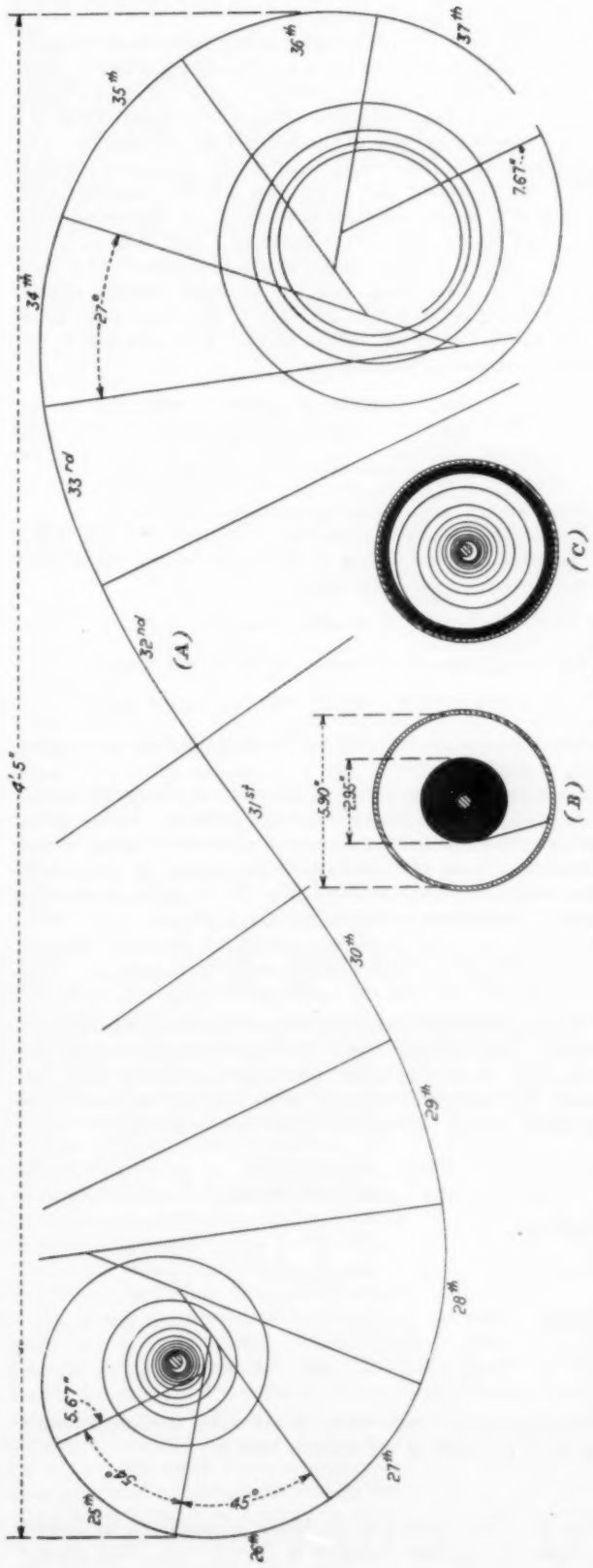


FIG. 12 DESIGN OF A SPIRAL SPRING

(Length, 267 in.; width, 1.0 in.; thickness, 0.025 in.; arbor diameter, 0.450 in.; barrel diameter, 5.900 in.; maximum stress, 375,000 lb. per sq. in.; modulus of elasticity, 30,000,000 lb. per sq. in. (*A*) shows spring entirely released; (*B*), wound tight on arbor; (*C*), partially released in barrel. Energy in spring: (*A*), 0.0 in-lb.; (*B*), 5215 in-lb.; (*C*), 925 in-lb.)

stored in the spring is a function of the square of the stresses involved. The highest stresses induced, consistent with safety, therefore, are the most effective.

Fig. 12 shows an outline of a spiral spring, theoretically designed to satisfy the following dimensions. The spring is 267 in. long. It is made of cold-drawn leaf steel of an elastic limit of 375,000 lb. per sq. in., and has a modulus of elasticity of 30,000,000 lb. per sq. in. The cross-section of the leaf is 1 in. $\times$ 0.025 in. The spring is shown in three stages, first, at (A), completely released, second, at (B), wound tight on an arbor of 0.45 in. diameter, and, third, at (C), partially released in a barrel. In the second stage the wound spring assumes the shape of a disk, 2.95 in. in diameter. The shape of the spring in the unstressed state is such that, when it is wound up tight, the maximum bending stresses throughout its entire length are equal to the elastic-limit stress of 375,000 lb. per sq. in. When this condition is satisfied, the spring has stored an elastic energy,

$$W = \frac{S^2V}{6E} = \frac{375{,}000^2 \times \pi(2.95^2 - 0.45^2) \times 1}{6 \times 4 \times 30{,}000{,}000}$$

$$= 5215 \text{ in-lb.} \quad \text{(See formula [2}a\text{])}$$

When wound tight, the spring consists of 50 coils.

The length of a tightly wound spring in terms of r_1, the radius of the arbor, t, the thickness of the spring, and n, the number of coils, is given by the expression

$$L = \pi n(2r_1 + nt)$$

The spring under consideration, therefore, is of a length

$$L = \pi \times 50(2 \times 0.225 + 50 \times 0.025) = 267 \text{ in.}$$

Let r represent the radius to a point on the tightly coiled spring ($r = r_1 + nt$).

Let R represent the radius of curvature of the spring at the same point when the spring is entirely released. As the spring undergoes the change from complete release to a tightly wound state, the change of curvature in the spring is $1/r - 1/R$. This change of curvature is M/EI. For a spring made of a flat leaf of rectangular cross-section bt,

$$\frac{1}{r} - \frac{1}{R} = \frac{M}{EI} = \frac{2S}{tE}$$

The conditions governing a spring of maximum efficiency are twofold. The bending stresses throughout must be within the elastic-limit stress, and must be as large as possible. If S represents the maximum working stress (in this instance S = 375,000 lb. per sq. in.), this condition is expressed as:

$$\frac{2S}{tE} = \frac{2 \times 375{,}000}{0.025 \times 30{,}000{,}000} = 1$$

Therefore

$$\frac{1}{r} - \frac{1}{R} = 1$$

Whence

$$\frac{1}{R} = \frac{1 - r}{r}, \quad \text{or} \quad R = \frac{r}{1 - r}$$

Let $\Delta\theta_1$ measure a segment of the nth coil of the tightly wound spring. The length of the segment then is:

$$\Delta L = r \times \Delta\theta_1$$

When the same element of the spring is completely unstressed its radius of curvature should be $R = r/(1 - r)$. The angular segment $\Delta\theta$, which this element extends in the unstressed state, is:

$$\Delta\theta = \frac{\Delta L}{R} = \frac{r\Delta\theta_1 \times (1 - r)}{r} = (1 - r)\Delta\theta_1$$

Table 1 lists the values of r, $1 - r$, R, $\Delta\theta$, and θ. For the first 11 coils of the tightly wound spring $\Delta\theta_1$ = 180 deg., while for the last 39 coils $\Delta\theta_1$ = 360 deg.

The construction lines on Fig. 12 indicate how the curve is drawn. For example: For the 25th coil in the tightly wound spring, the radius of curvature is:

$$r = r_1 + nt = 0.225 + 25 \times 0.025 = 0.850 \text{ in.}$$

The radius of curvature of the same element of spring when completely released, in order to satisfy the conditions stipulated, should be $R = r/(1 - r) = 0.850/0.150 = 5.67$ in. If we consider the element of length of spring under consideration to be the entire 25th loop of the closely wound spring, then $\Delta L = 2\pi r = 2\pi \times 0.85$. This same element of length, when it has a radius of curvature R in the unstressed state, extends through an angle $\Delta\theta = \Delta L/R = \Delta L(1 - r)/r = 2\pi(1 - r) = (1 - r) \times 360° = 0.15 \times 360° = 54°$. The 26th loop of the closely coiled spring has a radius of 0.875 in. Its radius of curvature in the unstressed state is 7.00 in. The angle it subtends is 45 deg. Fig. 12 shows how the curve is constructed by drawing successive angular segments with a common tangent, but with a different radius.

Only a few construction lines are drawn. The curve of Fig. 12 is known as an involute. The center of curvature continually shifts. The locus of the center of curvature of the curve is the evolute. For the beginning of the curve the centers of curvature for the various arcs lie very close together, too close to be accurately shown in Fig. 12.

It is seen that the radius of curvature for the 31st segment is infinity, and beyond that the curvature of the curve changes sign. To determine this point of inflection in general terms we equate R to infinity. Thus:

$$\frac{1}{r} - \frac{1}{R} = \frac{M}{EI} = \frac{2S}{tE}$$

$$\frac{1}{R} = \frac{1}{r} - \frac{2S}{tE} = \frac{1 - (2S/tE)r}{r}$$

$$R = \frac{r}{1 - (2S/tE)r}$$

When $1 - (2S/tE)r = 0$, or when $r = tE/2S$, then the radius of curvature of the unstressed spring is infinite.

$$r = r_1 + nt = tE/2S$$

Therefore, to locate in terms of n the point on the spring which in the unstressed state should have infinite radius of curvature, we solve for n.

$$n = \frac{(tE/2S) - r_1}{t}$$

The length of the spring, in terms of n, t, and r_1, is expressed as:

$$L = \pi n(2r_1 + nt)$$

Therefore the distance of the point of inflection from the arbor is:

$$L = \frac{\pi}{t}\left(\frac{tE}{2S} - r_1\right)\left(2r_1 + \frac{tE}{2S} - r_1\right) = \frac{\pi}{t}\left\{\left(\frac{tE}{2S}\right)^2 - r_1^2\right\}$$

$$= \pi\left(\frac{tE^2}{4S^2} - \frac{r_1^2}{t}\right) \qquad [8]$$

In terms of the data assumed in the design of the spring, represented by Fig. 12, the length of the spring from the arbor to the point of inflection is:

$$L = \pi\left(\frac{0.025 \times 30{,}000{,}000^2}{4 \times 375{,}000^2} - \frac{0.225^2}{0.025}\right)$$

$$= \pi(40 - 2.025) = 119.3 \text{ in.}$$

Fig. 12 shows a curve 267 in. long. A curve of such proportions may not be practical. However, any shorter portion of Fig. 12 satisfies the conditions of the problem equally well.

The spiral spring of Fig. 8 has a thickness t equal to 0.022 in. Assuming its elastic-limit stress as 375,000 lb. per sq. in. and its arbor of 0.225 in. radius, then its length up to its point of inflection should, in accordance with Formula [8], be:

$$L = \pi\left(\frac{0.022 \times 30{,}000{,}000^2}{4 \times 375{,}000^2} - \frac{0.225^2}{0.022}\right)$$

$$= \pi(35.20 - 2.30) = 103.4 \text{ in.}$$

As a matter of fact its length is 160 in., while the spring does not manifest a point of inflection. This argues that, when the spring is completely wound, it is not uniformly stressed throughout its length. The stresses in the spring near the arbor are considerably higher than those near the outer end of the spring.

The foregoing argument explains why, in commercial springs of proportions similar to Fig. 8, springs of considerable length but which do not show a point of inflection, breakages generally occur within the first three loops measured from the arbor.

The inner end of the spring is tangent to the arbor, the outer end approaches asymptotically to a circle of radius $R = tE/2S$.

In terms of the values assumed for the design illustrated by Fig. 12, the limiting radius of the circle which a spring of infinite length would approach as asymptote is:

$$R = \frac{0.025 \times 30{,}000{,}000}{2 \times 375{,}000} = 1 \text{ in.}$$

SPIRAL SPRINGS OF ANY THICKNESS

The radius of curvature of the completely released spring in terms of the thickness of the spring t, the working stress S, and the modulus of elasticity E, is given by the expression:

$$R = \frac{r}{1 - \frac{2Sr}{tE}} = \frac{r_1 + nt}{1 - \frac{2S}{tE}(r_1 + nt)}$$

Some other springs with an arbor radius $r_1' = wr_1$ and thickness $t' = wt$ would have a radius of curvature in the released state of

$$R' = \frac{r_1' + nt'}{1 - \frac{2S(r_1' + nt')}{t'E}} = \frac{w(r_1 + nt)}{1 - \frac{2S}{tE}(r_1 + nt)} = wR$$

As long as r_1 and t are maintained in a fixed ratio (in the design of Fig. 12, $r_1/t = 0.225/0.025 = 9$), therefore, Fig. 12 represents to some scale the outline of any spring designed to satisfy the condition of a working stress of 375,000 lb. per sq. in. and a modulus of elasticity of 30,000,000 lb. per sq. in. When the curve represented by Fig. 12 was plotted full size to represent a spring of thickness 0.025 in., its overall dimension was 53 in. In reducing the original for reproduction, the overall dimension proves to be $8^1/_2$ in. Any portion of the figure shown in Fig. 12 in this paper, measured from the arbor at the left, thus represents to full size the ideal shape of a spiral spring with a thickness $8^1/_2/53 \times 0.025 = 0.004$ in., designed for a working stress of 375,000 lb. per sq. in.

Fig. 12 may be photographically enlarged or reduced to represent to size the ideal outline (for the purpose of maximum storage of energy), for any spring with a ratio of arbor radius to thickness of spring equal to 9, a working stress of 375,000 lb. per sq. in., and a modulus of elasticity of 30,000,000 lb. per sq. in. In fact, nearly all of Fig. 12 represents the ideal outline of any spiral spring with a view to storing a maximum elastic energy, regardless of arbor dimension. For a spring with an arbor dimension larger than nine times the thickness of the spring, the point of beginning of the spring may be located on Fig. 12(*b*) and its corresponding point on Fig. 12(*a*), or may be obtained from values corresponding to these points listed in the table. For a spring with an arbor dimension smaller than nine times the thickness of the spring the curve should be plotted for smaller values than the initial values in the table.

Thus, Fig. 12, except possibly for the point of beginning, represents to some scale the ideal shape of any spiral spring of rectangular cross-section, designed for a stress of 375,000 lb. per sq. in. and a modulus of elasticity of 30,000,000 lb. per sq. in.

SPRINGS PARTIALLY RELEASED IN BARREL

Fig. 12 shows the outline which a spring must have in the unstressed state in order that it may be uniformly stressed throughout its length, and that it may store a maximum of elastic energy when completely wound. It is not likely that the total potential elastic energy in a spring can be utilized. A common practice is to restrain a spring against complete expansion. This practice finds its justification in the fact that the elastic energy stored is a function of the square of the stresses. For example, if the initial stresses in a partially released spring be $^1/_4$ of the maximum stresses throughout its entire length, and if, in the ultimate tightly wound condition [Fig. 12(*B*)], maximum stresses prevail throughout the entire length of the spring, then the spring will have stored in its partially released state $(^1/_4)^2$, or $^1/_{16}$ of the total potential energy which the spring is capable of storing. This would then leave $^{15}/_{16}$ of the spring's potential energy available for utilization.

The elastic energy in a spiral spring in any intermediate state is dependent upon the initial shape of the spring in the unstressed state. No convenient short-cut formula can be offered to express the energy of a spring in any state. It has been seen earlier in the section on design that commercial spiral springs at present available fail to satisfy the outline which our theory would dictate. Rather than make an analysis on the basis of an arbitrary shape of spring, it is proposed to evaluate the elastic energy stored in the spring, discussed in the section on design. For such analysis we have definite limiting conditions, when the spring is partially released in a barrel. A twofold purpose underlies this analysis: First, to illustrate how the energy of a spring in any shape may be evaluated, second, to obtain a numerical estimate of the statement previously made to the effect that partial restraint of a spring is not objectionable as long as opportunity remains of utilizing the highest increments of stress.

The spring represented by Fig. 12 is 267 in. long, 1.000 in. wide, and 0.025 in. thick. When completely wound, on an arbor of radius $r_1 = 0.225$ in. [Fig. 12(*B*)], it assumes approximately the shape of a cylinder consisting of 50 coils and having a final radius $r_2 = 1.475$ in. In this latter state it is uniformly stressed throughout its entire length to a stress of 375,000 lb. per sq. in., and stores a total elastic energy equal to 5215 in-lb. It is proposed to discuss the energy conditions of this same spring when released in a barrel having an inside radius twice as large as the radius of the spring in the tightly wound state [Fig. 12(*C*)]. The inside radius of the barrel then is $r_b = 2.95$ in.

Fig. 12(*C*) represents the spring as released in a barrel. Furthermore, all diagrams in Fig. 12 are drawn to scale, except that in (*B*) and (*C*), the closely wound portions of the spring, because of the reduction necessary, are represented as black rings instead of Archimedean spirals.

When the spring of Fig. 12(*B*) is released in a barrel of 2.95 in. radius, the outer end of the spring presses against the wall of the barrel in the shape of an Archimedean spiral. The inner end of the spring, however, is in the unstressed state as represented by Fig. 12(*A*). The point at which this unstressed portion of the spring [Fig. 12(*C*)] begins, is located in the following manner:

Suppose the entire length of the spring were crowded as closely as possible against the wall of the barrel. The entire spring then would be in the shape of an Archimedean spiral with a minimum radius of 2.565 in.

It may be seen from Table 1 that when the 20th coil [Fig. 12(*B*)] is released, it assumes a radius of 2.61 in., and at the release of the 21st coil the radius of the latter is increased to 3.00 in. The distance out to the end of the 20th coil [Fig. 12(*B*)] is 61.26 in.

As the spring represented by Fig. 12(*B*) is permitted to expand against the walls of a barrel having a radius of 2.95 in. [Fig. 12(*C*)] the last thirty coils [Fig. 12(*B*)] press against the barrel and against each other in the shape of an Archimedean spiral, while the first 20 coils assume the unstressed shape as represented in Fig. 12(*A*).

In the final shape [Fig. 12(*C*)] the inner 61.26 in. of the spring are unstressed, and therefore this portion has no elastic energy stored within it. The outer portion of the spring is in the shape of an Archimedean spiral. It is 267 — 61.26 = 205.74 in. long. Let r_4 represent the distance from the center to the point marking the end of the unstressed portion of the spring, which is the beginning of the Archimedean spiral of Fig. 12(*C*). The value of r_4 may then be found from the following equations:

$$r_4 = r_5 - n_4 t = 2.95 - n_4 t$$

$$L = \pi n_4(2r_5 - n_4 t) = 205.74 = \pi n_4(5.90 - n_4 \times 0.025)$$

Therefore $n_4 = 11.8$, and $r_4 = 2.95 - 11.8 \times 0.025 = 2.655$.

Our problem, then, is to find the elastic energy stored in the Archimedean spiral, consisting of 11.8 coils, having an outer radius $r_5 = 2.95$ in. and an inner radius $r_4 = 2.655$ in. In the coiled state this same spring is uniformly stressed from end to end to a maximum bending stress of 375,000 lb. per sq. in. [Fig. 12(*B*)] and consists of 50 coils having an outer radius $r_3 = 1.475$ in. and an inner radius $r_2 = 0.725$ in.

In changing from the closely coiled state to the expanded state any point on the spiral undergoes a change in curvature,

$$\frac{1}{r} - \frac{1}{R} = \frac{M_1}{EI} - \frac{M_2}{EI}$$

The radial distance to any point on the closely coiled spiral is, $r = r_2 + n_2 t$.

The radial distance to the same point on the expanded spiral is $R = r_4 + n_4 t$.

M_1 is the moment in the closely coiled spring [Fig. 12(*B*)]. We may express M_1/EI as K, and in this particular example $M_1/EI = K = 1$. M_2 is the moment in the expanded spiral.

The energy in the spiral closely pressed against the wall of the barrel is:

$$W = \int_0^L \frac{M^2_2 dL}{2EI}$$

$$\frac{M_2}{EI} = \frac{M_1}{EI} + \frac{1}{R} - \frac{1}{r} = K + \frac{1}{r_4 + n_4 t} - \frac{1}{r_2 + n_2 t}$$

From the equation $L = \pi n_4(2r_4 + n_4 t)$ we obtain:

$$r_4 + n_4 t = \sqrt{r_4^2 + \frac{Lt}{\pi}}$$

Similarly we may write:

$$r_2 + n_2 t = \sqrt{r_2^2 + \frac{Lt}{\pi}}$$

Thus:

$$\frac{M_2}{EI} = K + \frac{1}{\sqrt{r_4^2 + \frac{Lt}{\pi}}} - \frac{1}{\sqrt{r_2^2 + \frac{Lt}{\pi}}}$$

Hence, energy is expressed by

$$W = \frac{EI}{2}\int_0^L \left(\frac{M_2}{EI}\right)^2 dL$$

$$= \frac{EI}{2}\int_0^L \left(K + \frac{1}{\sqrt{r_4^2 + \frac{Lt}{\pi}}} - \frac{1}{\sqrt{r_2^2 + \frac{Lt}{\pi}}}\right)^2 dL$$

Integrating and substituting limits, the elastic energy stored in the spring represented by Fig. 12(*C*) is:

TABLE 1

No. of loops	r	$1 - r$	$R = \frac{r}{1-r}$	$\Delta\vartheta = (1-r) \times \Delta\varphi_1$	ϑ
1/2	0.2375	0.7625	0.3115	137° — 15′	137° — 15′
1	0.25	0.750	0.333	135° — 0′	272° — 15′
1 1/2	0.2625	0.7375	0.356	132° — 45′	360° + 45° — 0′
2	0.275	0.725	0.379	130° — 30′	360° + 175° — 30′
2 1/2	0.2875	0.7125	0.403	128° — 15′	360° + 303° — 45′
3	0.300	0.700	0.428	126° — 0′	2 × 360° + 69° — 45′
3 1/2	0.3125	0.6875	0.454	123° — 45′	2 × 360° + 193° — 30′
4	0.325	0.675	0.481	121° — 30′	2 × 360° + 315° — 0′
4 1/2	0.3375	0.6625	0.509	119° — 15′	3 × 360° + 74° — 15′
5	0.350	0.650	0.538	117° — 0′	3 × 360° + 191° — 15′
5 1/2	0.3625	0.6375	0.568	114° — 45′	3 × 360° + 308° — 0′
6	0.375	0.625	0.600	112° — 30′	4 × 360° + 58° — 30′
6 1/2	0.3875	0.6125	0.633	110° — 15′	4 × 360° + 168° — 45′
7	0.400	0.600	0.667	108° — 0′	4 × 360° + 276° — 45′
7 1/2	0.4125	0.5875	0.702	105° — 45′	5 × 360° + 22° — 30′
8	0.425	0.575	0.738	103° — 30′	5 × 360° + 126° — 0′
8 1/2	0.4375	0.5625	0.777	101° — 15′	5 × 360° + 227° — 15′
9	0.450	0.550	0.818	99° — 0′	5 × 360° + 326° — 15′
9 1/2	0.4625	0.5375	0.860	96° — 45′	6 × 360° + 63° — 0′
10	0.4750	0.525	0.905	94° — 30′	6 × 360° + 157° — 30′
10 1/2	0.4875	0.5125	0.952	92° — 15′	6 × 360° + 249° — 45′
11	0.500	0.500	1.000	90° — 0′	6 × 360° + 339° — 45′
12	0.525	0.475	1.105	171°	7 × 360° + 151′
13	0.550	0.450	1.223	162°	7 × 360° + 313′
14	0.575	0.425	1.353	153°	8 × 360° + 106′
15	0.600	0.400	1.500	144°	8 × 360° + 250′
16	0.625	0.375	1.656	135°	9 × 360° + 25′
17	0.650	0.350	1.856	126°	9 × 360° + 151′
18	0.675	0.325	2.090	117°	9 × 360° + 268′
19	0.700	0.300	2.333	108°	10 × 360° + 16′
20	0.725	0.275	2.610	99°	10 × 360° + 115′
21	0.750	0.250	3.000	90°	10 × 360° + 205′
22	0.775	0.225	3.445	81°	10 × 360° + 286′
23	0.800	0.200	4.000	72°	10 × 360° + 358′
24	0.825	0.175	4.720	63°	11 × 360° + 61′
25	0.850	0.150	5.67	54°	11 × 360° + 115′
26	0.875	0.125	7.00	45°	11 × 360° + 160′
27	0.900	0.100	9.00	36°	11 × 360° + 196′
28	0.925	0.075	12.33	27°	11 × 360° + 223′
29	0.950	0.050	19.00	18°	11 × 360° + 241′
30	0.975	0.025	39.00	9°	11 × 360° + 250′
31	1.000	0.000	∞	0°	11 × 360° + 250′
32	1.025	—0.025	—41.00	— 9°	11 × 360° + 241′
33	1.050	—0.050	—21.00	— 18°	11 × 360° + 223′
34	1.075	—0.075	—14.30	— 27°	11 × 360° + 196′
35	1.100	—0.100	—11.00	— 36°	11 × 360° + 160′
36	1.125	—0.125	—9.00	— 45°	11 × 360° + 115′
37	1.150	—0.150	—7.67	— 54°	11 × 360° + 61′
38	1.175	—0.175	—6.71	— 63°	10 × 360° + 358′
39	1.200	—0.200	—6.00	— 72°	10 × 360° + 286′
40	1.225	—0.225	—5.44	— 81°	10 × 360° + 205′
41	1.250	—0.250	—5.00	— 90°	10 × 360° + 115′
42	1.275	—0.275	—4.636	— 99°	10 × 360° + 16′
43	1.300	—0.300	—4.333	—108°	9 × 360° + 268′
44	1.325	—0.325	—4.077	—117°	9 × 360° + 151′
45	1.350	—0.350	—3.857	—126°	9 × 360° + 25′
46	1.375	—0.375	—3.667	—135°	8 × 360° + 250′
47	1.400	—0.400	—3.500	—144°	8 × 360° + 106′
48	1.425	—0.425	—0.353	—153°	7 × 360° + 313′
49	1.450	—0.450	—3.222	—162°	7 × 360° + 151′
50	1.475	—0.475	—3.100	—171°	6 × 360° + 340′

$$W = \frac{EI}{2}\left\{ K^2L + \frac{4\pi K}{t}\left(\sqrt{r_4^2 + \frac{Lt}{\pi}} - r_4\right) - \frac{4\pi K}{t}\left(\sqrt{r_2^2 + \frac{Lt}{\pi}} - r_2\right) + \frac{2\pi}{t}\log_e \frac{\sqrt{\left(r_2^2 + \frac{Lt}{\pi}\right)\left(r_4^2 + \frac{Lt}{\pi}\right)} \times \left(r_2r_4 + \frac{r_2^2 + r_4^2}{2}\right)}{r_2r_4\left(\sqrt{\left(r_2^2 + \frac{Lt}{\pi}\right)\left(r_4^2 + \frac{Lt}{\pi}\right)} + \frac{Lt}{\pi} + \frac{r_2^2 + r_4^2}{2}\right)}\right\}$$

In the example, $K = 1$, $r_2 = 0.725$, $r_4 = 2.655$, $L = 205.74$, and $t = 0.025$. Substituting these values in the foregoing formula we obtain for the elastic energy stored in the partially expanded spring [Fig. 12(*C*)]:

$$W = 925 \text{ in-lb.}$$

This is 17.7 per cent of the energy stored in the same spring when completely wound [Fig. 12(*B*)]. The spring, partially expanded in a barrel, may therefore be utilized to 83.3 per cent of its maximum potential energy.

EFFICIENCY OF SPIRAL SPRINGS

The only experimental evidence the author has to offer is of a qualitative nature. It consists of the photographs represented by Figs. 7, 8, and 11, which demonstrate the desirability of mounting springs in agreement with the theory expounded in case I and as represented by the loading and restraints shown in Fig. 1.

Since writing the foregoing, a valuable contribution to the spiral-spring theory has come to the author's notice, namely, "Spiral Springs for Spring-Driven Motors," by E. C. Wadlow, in *The Engineer*, October 31, 1930.

Mr. Wadlow comments on the inadequacy of the current theory of spiral springs. Although he offers no contribution toward improving this theory (such theory as he gives is practically identical with that found in Morley's "Strength of Materials"), he offers the results of some very interesting tests. In tests on four phonograph springs, with the inner end wound on an arbor and the outer end attached to a barrel, the springs being greased with a mixture of castor oil and graphite, he found the difference between the work supplied to the spring and the work given up amounted to a maximum of 35 per cent and to a minimum of 13 per cent. Furthermore, he states that the lubricating mixture between the leaves has a thickness of 0.003 in.

Aside from the undesirability of having to use a lubricating compound, which in time may lose its properties, its presence demands space and thus results in a loss of efficiency. In a spring of 0.025 in. thickness, assuming it otherwise to be most favorably stressed, this loss in efficiency amounts to 12 per cent.

On the basis of Mr. Wadlow's experiments, therefore, it would seem that the highest efficiency of a spring, having its outer end attached to a barrel and the leaves greased with a lubricating compound, is only 75 per cent that of a spring mounted in the manner shown in Figs. 8, 9, or 11.

To attain the highest possible efficiency in a spiral spring, one more point needs consideration, namely, the attachment of the spring to the arbor.

In the development of the theory the author has used the expression $r = r_1 + nt$, which is correct for an Archimedean spiral. He has also used the expression, $L = \pi n(2r_1 + nt)$, which expresses the length of a tightly wound spring, considered as equivalent to a number of concentric circles. The latter expression, although never strictly true, does not introduce any appreciable errors. The advantage of designing a spring in the manner shown in Fig. 12 and of mounting it in agreement with Figs. 8, 9, or 11, lies in the fact that under these conditions the spring is uniformly stressed from end to end with no localized points being subjected to overstress. If a spiral spring is wound on a cylindrical arbor, a bump is created when the second leaf adjusts itself over the first leaf at the point of attachment. For a very short distance the spring will be understressed and for a short distance, adjacent to the understressed portion, the leaf will be overstressed. This condition will make itself felt in a lesser degree throughout the coils at points approximately on a radius passing through the point of attachment of the spring to the arbor. The spring must be designed so that at this point of overstress the stress shall be within the safe working stress. This would be highly inefficient, as the condition in a fraction of 1 per cent of the spring would be the limiting condition for the entire spring. The other alternative is so to construct the arbor that its circumference will describe an Archimedean spiral. It may be difficult to realize the latter suggestion. On the other hand, the efficiency gained thereby is very considerable.

ACKNOWLEDGMENTS

The author is indebted to John Wiley & Sons, Inc., for permission to quote from his book, "The Elastic Energy Theory;" to Mr. Olaf Ohlson, of the Waltham Watch Co., for the use of Fig. 7; to Mr. Zimmerli, of Barnes, Raymond, Gibson & Co., for material supplied for testing purposes; to Messrs. Edw. A. Clark, watch repairer, and W. M. Simmons, phonograph repairer, both of Ann Arbor, for practical information; and to his colleagues, Professors H. R. Lloyd and T. H. Hildebrandt, of the University of Michigan, for valuable suggestions and criticisms.

Discussion

M. F. Sayre.[3] Certain criticisms in matters of detail might well be made in connection with this paper.

As dealt with by the author, case III (*b*) is identical with case III (*a*), and it in turn is identical with case I. In case III (*c*) there is an inconsistency in the statement that point *B* is free to move on the line *BE*, while at the same time assuming a reaction at *B* in the direction *BE*. As this case has been actually handled, the reaction has been ignored, so making it identical with case II.

In case II, the statement is made that "in case point *A* is restrained in a horizontal direction as well as in a vertical direction, the horizontal component of the reaction at either *A* or *E* is zero, provided the spring may be considered as composed of a large number of concentric rings." The derivation is then continued on the basis of zero horizontal restraint. This would appear to be a case of placing the cart before the horse. As shown by the author himself later on, even minute horizontal restraint acts to change radically the shape of the spring. This whole derivation hinges, therefore, not on the absence of a horizontal reaction at *A*, but on the limitation of angular distortions to extremely small amounts. It would seem better, therefore, to start with a different initial assumption.

From a practical standpoint, the effect of frictional resistance within the coils is extremely important, and usually well safeguarded against, in a hairspring. In the usual power spring it is of very much less importance, since in the wound-up position of the spring, in which friction is largest, the spring will usually be developing considerably more power than is needed. Any losses of torque in the run-down position of the spring are undesirable, as they act to decrease the number of working turns; but at this stage, friction is more likely to result from rubbing between the spring and the case than between successive coils of the spring.

[3] Associate Professor of Applied Mechanics, Union College, Schenectady, N. Y. Mem. A.S.M.E.

Attention might well be directed to recent work of the A.S.M.E. Spring Committee, which has shown that stress distribution in fibers in a cold coiled power spring is likely to be very far from directly proportional to distance from the neutral axis. The differences are great enough to call for attention in any attempts at exact analysis.

J. S. A. JOHNSON.[4] The writer should like to have information regarding the behavior of a closely wound spiral spring when the end *A*, Fig. 1, is fixed and a load whose line of action is perpendicular to the plane of the coil (or parallel to its axis) is applied at the inner end *E*.

E. C. WADLOW.[5] The first part of the paper demonstrates the importance of a clear statement and understanding of the problem under discussion, and the discrepancies between the results obtained in the paper and those of other writers appear to have arisen out of the incomplete treatment previously given to the problem.

The use of spiral springs instead of helical springs subject to axial twist is no doubt due to practical advantages which the spiral spring possesses, quite apart from any question of efficiency as the medium for storing elastic energy. The length of the path traversed by the free end of a spiral spring is very considerable, and owing to the ease with which this type of spring is able to drive mechanisms through gearing, it is possible to utilize this great deflection while still retaining the spring within the confines of a barrel a few inches in diameter. The best way to appreciate the merits and conveniences of the spiral spring is to attempt to substitute for it a spring of another form.

No matter what system of anchoring and loading is employed, so far as gramophone motors are concerned, the final condition is a spring wound tight on the arbor. This is the form in which the stress in the material is highest, and for the most efficient use of the material the stress should be uniform throughout and the highest allowable. Since the shape of the spring when fully wound is determined almost entirely by geometrical considerations and not by considerations of stress due to end fixing, and since this is the time when maximum energy is stored in the spring, it follows that for gramophone springs which are wound tightly, the end connections do not play such an important part as in other types of spring which do not undergo such considerable changes in shape.

Since S, r, t, and E are known in the equation $\frac{1}{r} - \frac{1}{R} = \frac{2S}{Et}$ (which was incorrectly quoted in the article in *The Engineer* referred to by the author), this equation enables us to determine the radius of curvature of every point of the spring in the unstressed state in the manner shown in the paper.

In the example chosen to illustrate the use of the formula, the author arrives at double involute curves shown in Fig. 12. These curves are calculated on the assumption that the elastic limit for the material is 375,000 lb. per sq. in., but judged by the material used in this country for springs of the dimensions given, this stress is much too high. The material used for mainsprings of gramophone motors has an ultimate stress of about 240,000 to 300,000 lb. per sq. in. and an elastic limit of about 180,000 to 240,000 lb. per sq. in. The spring also is exceptionally long. Springs employed in this country for gramophone motors usually are not longer than 200 in.

To avoid unnecessarily high stresses where the spring is bent round the arbor, the ratio of arbor radius to spring thickness is also much greater than the value of 9 assumed.

[4] Professor of Applied Mechanics and Experimental Engineering, Virginia Polytechnic Institute, Blacksburg, Va. Mem. A.S.M.E.

[5] Hayes End, Middlesex, England.

When these values are employed, the double involute form shown in Fig. 12 of the paper is never necessary—and fortunately so, for the coiling of a spring to this form would be much more difficult and costly than the simple coiling at present used.

In practice, the involute form is never accurately obtained by the coiling, as indeed is to be expected.

As a result of these departures from the ideal form when free, the stress when fully wound is rarely the same all along the length

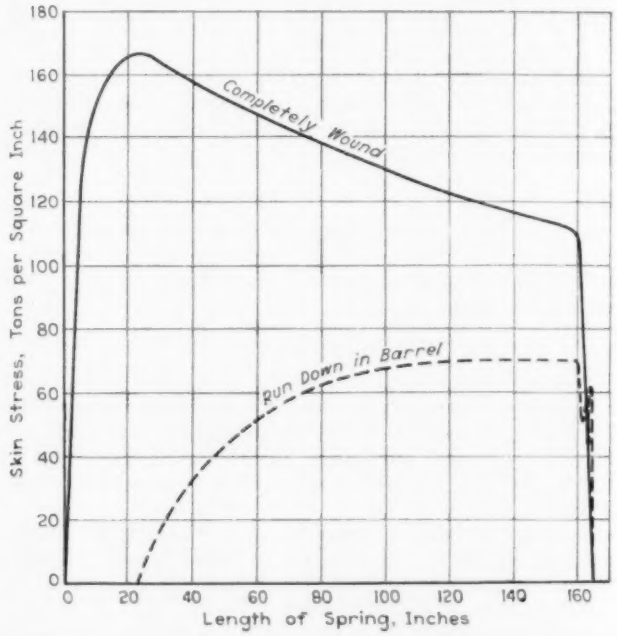


FIG. 13 CALCULATED SKIN STRESSES IN A GRAMOPHONE MOTOR SPRING

of the spring. Fig. 13 shows the calculated skin stress in a spring used in a portable gramophone motor, many thousands of which are giving a satisfactory performance. The stress when fully wound is very far from uniform, due to incorrect shape when free. It may also be observed that these calculated stresses are higher than the known breaking strength of the material, this resulting mainly through assuming that the material is perfectly elastic over the whole working range of stress shown. A certain amount of plastic deformation takes place in the most highly stressed layers every time the spring is wound up. This leads in time either to actual breakage of the spring or to sufficient permanent set to render the spring useless. Which of these happens, and the time elapsing before the failure, depend upon the properties of the material being used.

The initial deformation which takes place when a new spring is put into use is often considerable. Fig. 14 shows the results of measurements on two similar springs. The one was new, while the other had been in use for over a year. The change, however, usually takes place fairly quickly, and after a dozen winds is fairly stable. The condition that the spring must be able to withstand a number of winds without the number of coils, when free, increasing beyond a certain percentage, is sometimes included in a specification for spring material.

With regard to the lubrication of mainsprings the following figures may be of interest. They were obtained with a spring 176 in. long in a barrel $3^{11}/_{32}$ in. internal diameter with a sleeve $^{5}/_{8}$ in. in diameter. The spring was 0.022 in. thick and 1 in. wide.

The limiting conditions are shown in Fig. 15. The continuous line gives the torque when the spring was lubricated with a mix-

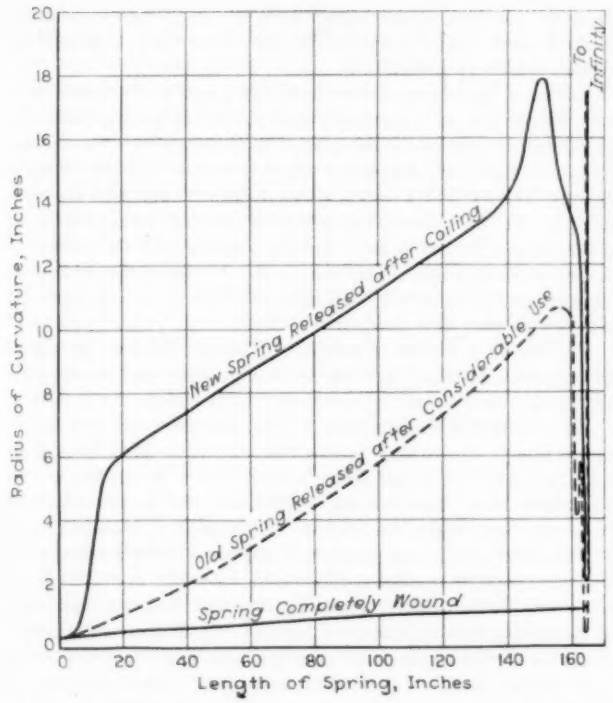


FIG. 14 RADII OF CURVATURE OF GRAMOPHONE MOTOR SPRINGS

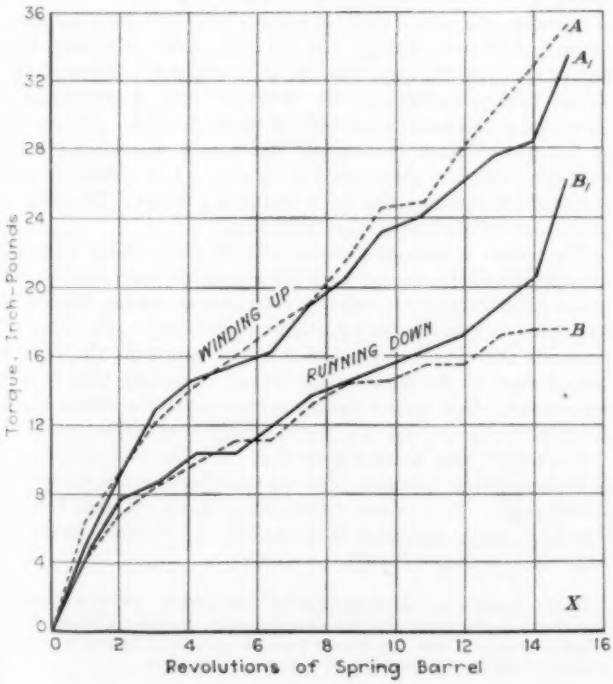


FIG. 15 TORQUE-ANGULAR DISPLACEMENT CURVES FOR SPIRAL SPRING

(Broken line, unlubricated; solid line, lubricated with castor oil and graphite.)

ture of castor oil and mineral graphite in approximately equal weights. The broken line shows the torque without any lubricant being present. The efficiencies on a number of cases were as follows, the efficiency being the ratio:

$$\frac{\text{Work recoverable while unwinding}}{\text{Work to wind}} = \frac{\text{Area } OBX}{\text{Area } OAX} = \frac{\text{Area } OB,X}{\text{Area } OA,X}$$

Lubricant	Efficiency, per cent
Castor oil and mineral graphite	70.4
Colloidal graphited castor oil	69.5
Mobiloil BB	68.6
Pure castor oil	66.1
No lubricant	61.2

These values are for single tests on the same spring carried out as nearly as possible under identical conditions.

With this design of spring and mounting, therefore, the increase in efficiency due to the lubricant more than outweighs the loss due to the space it occupies.

The conditions under which spiral springs have to perform in motors are particularly arduous. Few instances can be found where both the range of stress and the maximum stress are so great. This is possible, no doubt, because the allowable number of cycles of stress to cause failure is fairly small, being in the region of 30,000 to 50,000 cycles.

A. NÁDAI.[6] With reference to the principle of superposition employed in the paper and frequently in textbooks on strength of materials as one of the independent principles or assumptions on which the ordinary stress calculations are based, it seems to the writer that this principle could in many cases be considered merely as a consequence of *other* experimental and mathematical premises or fundamental assumptions also used in these computations, and on which certain parts of the theory of elasticity are ordinarily based: namely, that the three components of the displacement of a point in an elastic body are small (compared with the smallest dimension of the stressed body), and that stress and strain are connected by linear relations. As a consequence of these assumptions, differential equations are obtained which are linear, and therefore one solution or stress system can be added to another and the principle of superposition holds.

The superposition of particular integrals might sometimes even be applied when one of the displacement components starts to become greater; for example, when the plane elastic deflections of a slender straight steel strip clamped at one end and loaded by forces in one of its principal planes of inertia are considered at deflections which are *finite* and moderately large (for example, $^1/_{20}$ or $^1/_{10}$ of the span). The errors introduced by using the expressions $\epsilon_x = \frac{\partial u}{\partial x}, \ldots, \gamma_{xy} = \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}, \ldots$ where ϵ or γ are the unit strains, u, v, w the components of displacement and which are only valid for small values of u, v, w, might in such cases not be too serious, if one or the other of the components u, v, w becomes moderately larger. In other cases, however, of large deformation of beams (as in the case of the "Problem of Euler's Elastica"[7] or in the case of a beam with both ends rigidly clamped under moderately large deformations) these expressions cannot be used, or only in an infinitesimal region within the deflected beam; also in the case of the bending of flat, thin plates with moderately large deflections certain additional precautions have to be carefully considered, even if the deflections are still very small compared to the lateral dimensions of the plate.

One further important point might also be mentioned here, which has been clearly brought out by H. M. Westergaard.[8] There are cases in which, although stress and strain are still connected by linear equations and although the displacement components remain small, the deflections will not be linear func-

[6] Research Laboratories, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

[7] Cf. A. E. H. Love, "A Treatise on the Mathematical Theory of Elasticity," 4th edition, p. 401, Cambridge University Press.

[8] "Buckling of Elastic Structures," Proc. Am. Soc. C.E., vol. 47, no. 9, Nov. 1921, p. 455 (cf. especially p. 456.)

tions of the resultant forces under which they were produced. A well-known example is that of a beam bent by two equal axial loads acting eccentrically. In this case the greatest deflection of the beam is by no means proportional to the load, although the differential equation for the deflection is still a linear one. Further references to these and other similar points must be omitted here, as they are discussed clearly in detail in Westergaard's paper.[9] The writer thinks, therefore, that the statement: "The current theory of spiral springs appears to be one more case in which practice and theory disagree—with practice carrying off the honors," made by the author, may be a slightly exaggerated expression of the true conditions. The usual assumptions made in this special theory and in other parts of the mathematical theory of elasticity have often been verified for elastically isotropic materials such as steel by test. What is here important is that they have their definite limits of application as do all theories in physics. If this is borne in mind, if some errors apparently made in the boundary conditions in certain former computations on spiral springs as quoted by Professor Van den Broek are corrected, and if the conditions which define the limits for each assumption of the theory are carefully taken into consideration, no serious disagreement between "theory" and "practice" should remain.

In the second equation on page 2 of the paper and in the text below it, it would seem desirable to use for a displacement component not a capital letter and especially not the letter E_y, as E should be reserved for Young's modulus, which appears in the same equation.

It was instructive for the writer to see from one of the experimental demonstrations of Professor Van den Broek that the first winding of the spiral was mainly distorted, but the other windings remained practically circular. It would seem that this fact might be utilized for a more exact theory of the deformations of the spiral, including also large displacements in certain parts of the spring. How far Castigliano's theorem will still be valid in such a case of bending of a flat steel strip with parts of it having large lateral displacements seems to be a question which apparently has not yet been answered.

Author's Closure

The conclusions of the paper are to the effect that, except for friction between the coils, the principle of superposition holds and that the same theory and its resulting formulas are applicable throughout the entire loading range of spiral springs regardless of their condition of mounting.

Professor Sayre appears to argue that since the various cases in the last analysis are identical, they need not have been discussed. In the application of the theory of strength of materials we very often find that the principle of superposition is assumed without first verifying its applicability. Further, the analysis of structures involving material changes in dimensions while being loaded very rarely comes under discussion. In view of this the author considered it a necessary part of a rigorous proof to trace such changes of dimensions and loadings in detail.

In the preprint of the paper, in the middle of the fifth page left column, there appeared a typographical error; "case III (*a*)" was incorrectly printed as "case III (*b*)." Professor Sayre is correct in his claim that except for the shifting of the x-y coordinate system as the spring is loaded, case III (*b*) is the same as case III (*a*).

However, the author cannot agree with Professor Sayre when he charges that an inconsistency obtains in case III (*c*), (Fig. 4). True, Fig. 4 shows an upward force Q at B. Another downward force Q, however, is also shown at point A. Thus there is no vertical reaction at E, and point E is free to move in a vertical direction relatively to point B.

Professor Sayre's quotation from the paper with reference to his criticism of case II is correct, but entirely misleading when the paragraph in question is not quoted in its entirety. The author makes reference to a discussion which occurs in his book, "Elastic Energy Theory," of a spiral spring with its outer-end pin connected. In the present argument he omitted consideration of this case and instead chose to discuss a spring with the outer end A unrestrained in the direction A-E. If the outer end A is unrestrained in direction A-E, how can there be any component reaction in that direction?

The author is unable to enlighten Professor Johnson in regard to spiral springs fixed at their outer extremity and loaded concentrically with a force perpendicular to the plane of the spring.

Mr. Wadlow's contribution to the discussion is very much appreciated. He evidently has had extensive experience with spiral springs. It is not clear, however, how he arrives at the conclusion that, "the increase in efficiency due to the lubricant more than outweighs the loss due to the space it occupies."

The difference in percentages of efficiency between the best-lubricated spring and the unlubricated one taken from Mr. Wadlow's table is the difference between 70.4 and 61.2, or 9.2 per cent. If (according to Mr. Wadlow's paper in *The Engineer*) the space provided for the oil, amounting to 0.003 in., were filled with steel, there would be 3/22 more steel used, and thus an increase in power of $^3/_{22}$ of 61.2 or 8.35 per cent would be obtained. Even considering that there would be no deterioration of the lubricant with the passing of time, the lubricating of springs, though mounted in the old way, appears to have little to recommend it.

However, the author does not want to appear to be contending for more friction in springs. In the light of Mr. Wadlow's own curves (Fig. 15) showing the loss of energy due to friction, the author fails to understand Mr. Wadlow's lack of interest in a mounting which would practically eliminate friction. The spring in the motor-driven phonograph has a way of releasing itself suddenly while the phonograph is playing. This sudden release is invariably accompanied by a disturbing sound. This objection is met by the elimination of friction.

The author is inclined to agree with Professor Sayre when he asserts that in the wound state the spring has such an excess of power that, except for resulting intermittent action, friction is less objectionable than in a partially wound state. Mr. Wadlow contends that the energy stored in the spring in the tightly wound state is the determining factor. Granting this, it still remains true that in this tightly wound state, the spring being mounted in the old way, the stored energy is not all available.

The author very much regrets that Prof. Dr. Ing. Otto Föppl of Braunschweig, Germany, did not contribute to the discussion of this paper. In a review of the author's book, "Elastic Energy Theory," which appeared in *Zeitschrift des Vereines deutscher Ingenieure*, of July 11, 1931, he wrote:

> In the chapter on spiral springs he (the author) raises legitimate objections to the corresponding discussions in the older editions of A. Föppl's book. These objections have in part been taken into consideration in the last edition of Vol. 3 of this book.

By his reference, "the last edition," Föppl has in mind the book entitled, "Festigkeits Lehre von August und Otto Föppl," 1927. This is that celebrated book by the late August Föppl, the father, revised by Otto Föppl, the son.

In discussing the "principle of superposition," Dr. Nádai appears to interpret this principle as implying that the deformations are of a small order of magnitude relative to the smallest cross-section dimension of the structure. Has Dr. Nádai failed to

[9] With regard to these questions, compare also the corresponding chapters in Love's "Elasticity" quoted above, and S. Timoshenko's "Strength of Materials," 1930.

observe, what appears to the author one of the points of interest in his paper, that relatively very large deformations are taken into account without violating the principle of superposition? The case of the spiral spring is the only one within the author's experience in which this condition prevails. It may be true that relatively small deformations are generally a sufficient test for the applicability of the principle of superposition except in case of columns or examples supplied by Mr. Westergaard and referred to by Dr. Nádai. The author considers it as one of the points of interest of his analysis of spiral springs that it provides a clear illustration of an example in which the principle of superposition holds in spite of very large deformations. It thus provides an argument against a too ready conclusion that small deformations are a necessary and sufficient test of the applicability of the principle of superposition. It is not clear how, in the face of this evidence, Dr. Nádai still advances the requirement of small deformation as a necessary test of the linear relationship between stress and strain.

Dr. Nádai says that "the statement, 'the current theory of spiral springs appears to be one more case in which practice and theory disagree—with practice carrying off the honors,' made by the author of the paper, may be a slightly exaggerated expression of the true conditions." The author called attention to a disagreement between his own and August Föppl's conclusions, the difference amounting to several hundred per cent, and pointed out that his conclusions differed from those of Timoshenko by 100 per cent. As mentioned earlier in this closure, Otto Föppl, son of the late August Föppl, acknowledges the correctness of the author's conclusions. It is very much regretted that Dr. Timoshenko did not express his views on the merits of the author's contentions. If Dr. Nádai is of the opinion that the formulas quoted in the paper from other books correctly represent the elastic behavior of spiral springs, then why does he not defend them? If he believes they are incorrect and the author's conclusions correct, how then can he speak of an "exaggerated expression?"

As to Dr. Nádai's question relative to the applicability of Castigliano's theorem, it would appear that the answer is quite simple. Certainly, in cases involving friction between the leaves, no theorem based on the assumption of perfect elastic behavior can be considered as valid. In case 1, page 2, column 1, the energy due to bending is given as $W = \frac{M^2}{2}\frac{L}{EI}$. Thus $\frac{dW}{dM} = \frac{ML}{EI}$, which, by formula [3], page 2, column 2, is equal to θ. Thus the derivative of the energy due to bending with respect to M gives the angular displacement θ of the spring at the point of application of M. This is in perfect agreement with Castigliano's theorem, and independent of the degree of the deformation involved—provided the assumption of the existence of a relatively large number of concentric coils is satisfied.

Dr. Nádai further says, "If some errors apparently made in the boundary conditions in certain former computations on spiral springs as quoted by Professor Van den Broek are corrected, and if the conditions which define the limits for each assumption of the theory are carefully taken into consideration, no serious disagreement between 'theory' and 'practice' should remain." The author was naive enough to think he had done exactly that.

FSP-53-1

Stokers and Furnaces for New England Fuels

By R. A. FORESMAN[1] and DONALD J. MOSSHART,[2] PHILADELPHIA, PA.

In this paper the authors describe and analyze some of the troubles encountered in stoker installations and discuss the principal factors to be considered in making a successful stoker installation. The discussion is limited to those problems found in New England, where the fuel used creates problems peculiarly local. The principal fuel of New England is a semi-bituminous coal, known to the trade as Pocahontas and New River coal. Among the points discussed are, selection of type and size of stoker, preheated air, furnace volume, furnace walls, ash-disposal apparatus, waste fuel, operating methods, banking, boiler operation, and recent stoker installations. Many of the points discussed receive recognition in the majority of plants, but relatively few plants recognize them all. The authors bring these practical matters together in one group.

EVERY new boiler room is a potential trouble job. The various installations differ only in a degree governed by the design of the equipment, the fuel to be burned, and the manner in which the equipment is applied.

These statements concern both stoker and pulverized-fuel installations. The authors will make no attempt to compare stokers with pulverized-fuel equipment. Such comparison is better made by those who operate comparable installations of both classes of equipment. That such comparisons are being made is indicated by the continuously increasing sale of stokers and their repeated encroachment into the plants of users who have been, until recently, stanch supporters of pulverized fuel. The reason for the increasing favor accorded to stokers is that their troubles and advantages combine and react to produce steam more cheaply than the combined and reacting troubles and advantages of pulverized-fuel installations.

In this paper the authors will attempt to describe and analyze some of the troubles encountered in stoker installations and to discuss the principal factors to be considered in making a successful stoker installation. The discussion will be limited to those problems found in New England, where the fuel supply, which differs from that of other sections of the country, creates problems peculiarly local.

The principal fuel of New England is semi-bituminous coal, known to the trade as Pocahontas and New River coal. This fuel is used almost exclusively, for several reasons:

(*a*) It has a very high heat value
(*b*) The freight charges on coal delivered to New England are quite high, necessitating the purchase of coal of a high heat value, to keep the freight charges per B.t.u. as low as possible
(*c*) This coal can be safely stored in large quantities and for long periods of time, with a minimum of danger from spontaneous combustion
(*d*) Its low ash content reduces the charges for ash handling
(*e*) It is nearly smokeless, a quality desirable in those plants using hand firing or the less highly developed stokers.

In many respects Pocahontas and New River coal are highly prized for stoker use. They are more difficult to handle than the so-called "low grade" coals, when such factors as stoker maintenance, capacity, and flexibility are considered.

Some small quantities of waste fuels are available. The use of these fuels will be mentioned, but only in reference to burning them simultaneously with semi-bituminous coal.

It is a generally accepted fact that the multiple-retort underfeed stoker is the best stoker for semi-bituminous coal. This type of stoker has enjoyed wide popularity for many years, and has produced operating results of the highest order. The discussion will be confined to this type of stoker.

Selection of Stoker Size

The load requirements should be carefully estimated so that the proper size of stoker can be selected. Overloaded stokers are poor investments because of poor performance. Underloaded stokers are poor investments because money is unnecessarily tied up in them. The size of stoker to be selected is governed by:

(*a*) The amount and character of the load on the boiler unit
(*b*) The cost of coal and the evaluation of efficiency
(*c*) The capabilities of the stoker.

The limit of the capacity of the stoker is reached when the fuel bed is blown off the grates. This is a function of the velocity of air through the fuel bed, which, with a given percentage of excess air, is directly proportional to the combustion rate.

It is known from physical law that the force of impact of a current of fluid against a stationary object is proportional to the square of its velocity. Therefore, as the combustion rate increases, it might be expected that the lifting effect of the air current on the fuel bed would increase as the square of the combustion rate, with a consequent sudden and complete blowing of the fire from the stoker. However, certain changes take place in the character of the fuel bed, so that the actual area of air passage through the fuel is larger at the higher rates. A compensating reduction in actual velocity through the air passages somewhat modifies the expected action.

Stokers are rated with their 24-hour and 4-hour or 2-hour capacity. The 24-hour capacity is that combustion rate just short of where the ash-discharge section is smothered with fine coke blown from the fire or where the blowing is so severe that the fuel bed cannot be maintained in proper condition over a long period of time. The limits of capacity for the shorter periods are of the same nature, the more severe blowing at the maximum rates simply putting the fire out of condition more quickly and limiting the length of time that a given load can be carried.

In the past, stoker capacity has been measured in pounds of coal burned per square foot of grate surface.

The amount of air required to burn a pound of coal is a function of the combustible content of the coal, and therefore of the heat content of the coal. In other words, the factor which limits

[1] Chief Engineer, Stoker Department, Westinghouse Elec. & Mfg. Co. Mem. A.S.M.E.
[2] Assistant Chief Engineer, Westinghouse Elec. & Mfg. Co. Assoc-Mem. A.S.M.E.

Presented at a meeting May 15, 1929, under the auspices of the Boston Local Section of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

the capacity (the velocity of air through the fuel bed) is a function of the amount of heat being released per unit area and time, rather than of the weight of coal being burned per unit area and time.

This is an important consideration when comparing performance with different coals. The stoker which burns Illinois coal of 10,000 B.t.u. at a maximum rate of 80 lb. per sq. ft. could be expected to fail to burn New River coal of 14,000 B.t.u. at rates above 57 lb. per sq. ft. This comparison is based on equal heat-liberating rates.

New River or Pocahontas coal is relatively hard to ignite. Therefore, more of the grate area is inactive with these coals. Equally proficient stokers will be capable of a slightly lower heat-liberating rate when burning such coals than when burning high-volatile coals.

The conclusion is therefore that stoker tests should be compared on a basis of heat-liberating rate with similar coals.

Fig. 1 shows the characteristic performance of underfeed

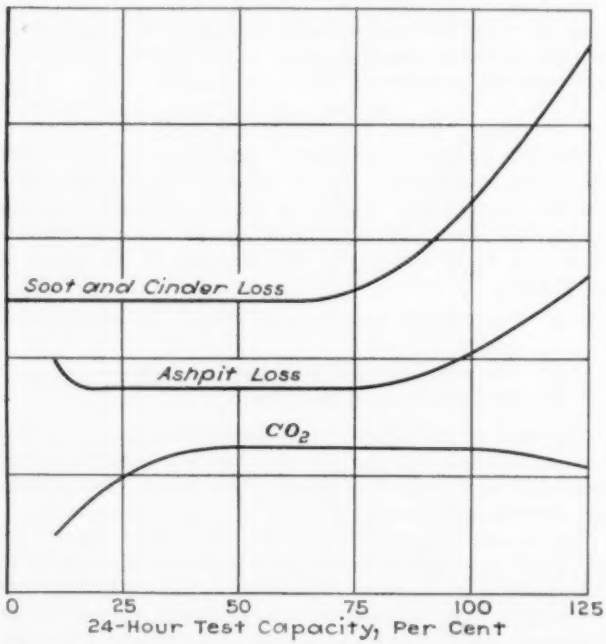


Fig. 1 Characteristic Performance of Underfeed Stokers

stokers. The abscissas represent percentages of the load that can be carried for 24 hours under test conditions. The ordinates are the three factors in the efficiency of the unit over which the stoker exercises control. Definite values are not assigned, but the different ordinates are of a similar scale of value in per cent overall efficiency of the unit.

Fig. 2 shows the typical load curves of three classes of industries. On these graphs the blocked figures show the load on the stoker (expressed in per cent of the 24-hour test capacity) for greatest dollar economy, considering investment cost, maintenance, and thermal efficiency, under average conditions. Analysis of local conditions, with particular regard to coal cost, may slightly modify these figures.

Knowing the capabilities of the stoker, the efficiency of the boiler unit, and the load conditions, the selection of the size of the stoker becomes a matter of arithmetic.

Selection of Type of Stoker

In this section there is covered the choice between clinker grinders and dump grates.

Where the plant carries a fairly heavy load 24 hours per day, a clinker-grinder stoker will ordinarily produce about 2 per cent more in efficiency than a dump-grate stoker.

For the load shown at *A*, Fig. 2, dump grates are the most economical, if the stoker is less than about 14 ft. long. Above

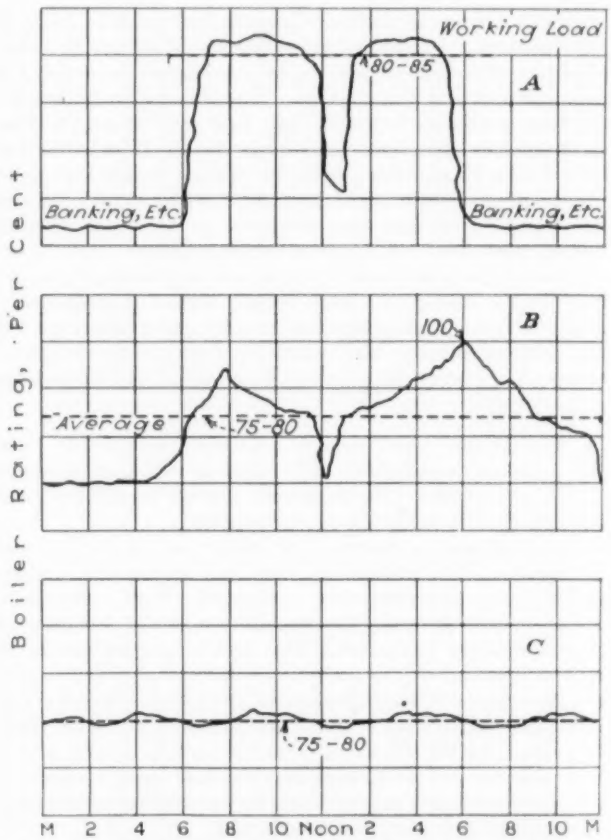


Fig. 2 Typical Load Curves of Three Classes of Industry (*A*, daily boiler-load curve, usual industrial plant; *B*, daily boiler-load curve, typical central station; *C*, daily boiler-load curve, base central station or industrial plant, as paper mill, using process steam continuously.)

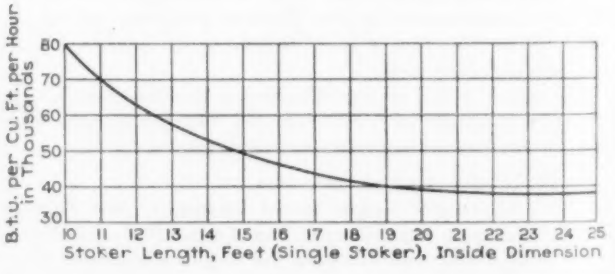


Fig. 3 Values of Heat Release With Different Lengths of Stokers

that length, clinker grinders should be used because of the large amount of ash to be handled per unit of stoker width. When using stokers of short or moderate length for such loads, the refuse is dumped during light loads or banked periods, and the performance closely approximates that obtained from clinker grinders under similar conditions.

For the loads shown at *B* and *C*, Fig. 2, clinker grinders should be used for all lengths of stoker, wherever space permits. If space limitations, particularly headroom prohibit the installation of clinker grinders, dump grates may be installed with the assur-

ance that they will give good results on any length of stoker yet developed.

Preheated Air

Preheated air has been in general commercial use for about four years. Extensive study during this time, of the problems created and of the results obtained by the use of preheated air, has permitted the following general conclusions to be drawn:

(1) Improvement in combustion, expected by the sponsors

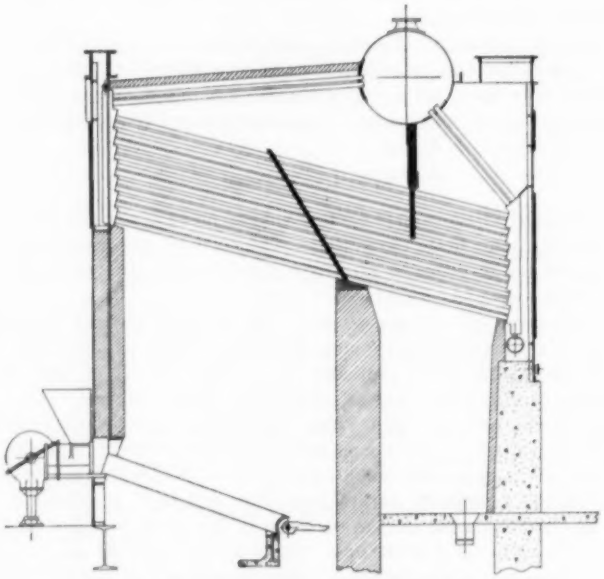


Fig. 4 Dead Areas in the Furnace Space

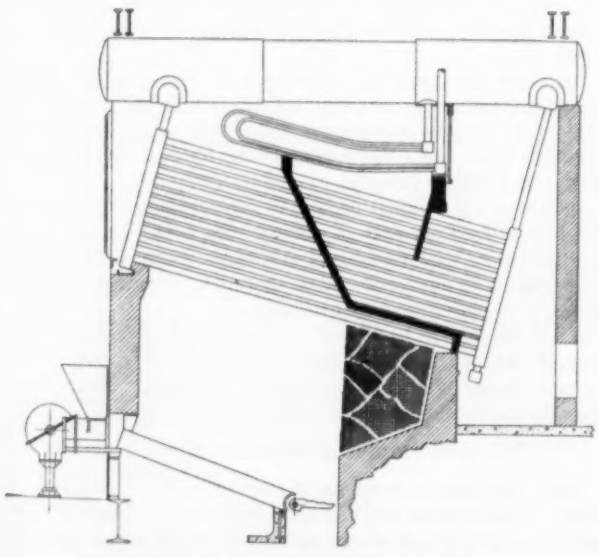


Fig. 5 Dead Areas in the Furnace Space

of preheated air and reported as being obtained by some users, has not materialized. Some changes do take place, but the beneficial effects, if any, can be duplicated with cold air by proper adjustment of the stoker.

(2) Preheated air does not measurably increase the capacity of the unit. The increased velocity of air through the fuel bed, caused by the greater specific volume of the heated air, increases the blowing of fuel and slightly reduces the coal-burning capacity of the stoker. A good rule for commercial practice is to figure a unit of grate area as having a fixed capacity in B.t.u. per hour, the amount of heat admitted to the furnace being

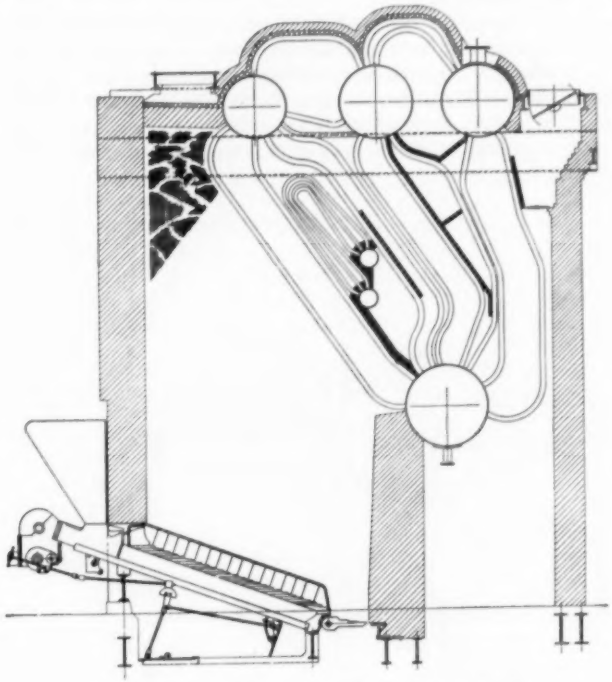


Fig. 6 Dead Areas in the Furnace Space

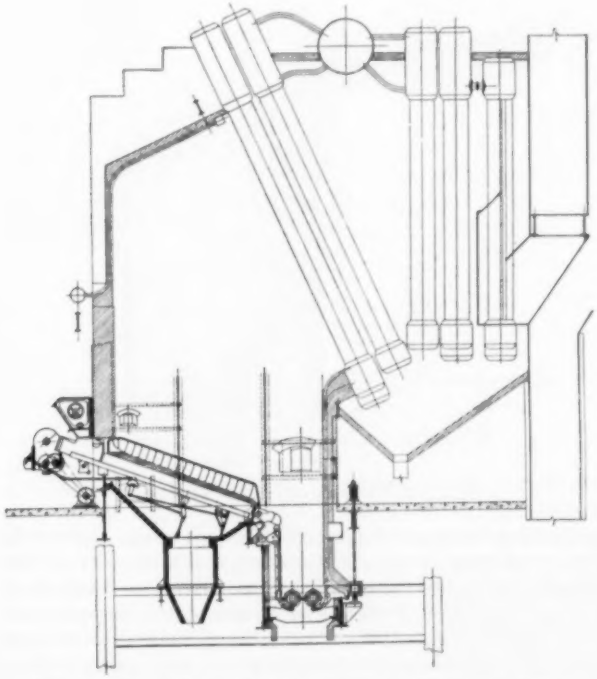


Fig. 7 Water-Wall Arrangement to Decrease Slag

the same whether or not the air is preheated. In the case of cold air, the heat released by combustion is the capacity. In the case of preheated air, the heat released by combustion plus the heat returned by the air is the capacity. In either case the total heat admitted to the furnace would be the same, but

with preheated air the amount of coal burned would be less by the coal equivalent of the heat in the air. This rule is not intended as a statement based on tests; it is for the commercial selection of stoker size.

(3) Preheated air does not measurably increase the efficiency of combustion. It may slightly decrease the ashpit loss, but this is offset by the increased loss in soot or fine carbon carried up the stack. The only value of preheated air lies in the amount of heat recovered and returned to the boiler.

(4) It is more difficult to operate an underfeed stoker with

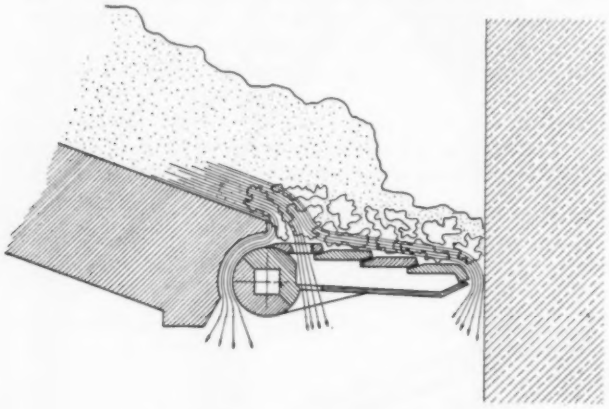


FIG. 8 DOWNWARD FLOW OF AIR

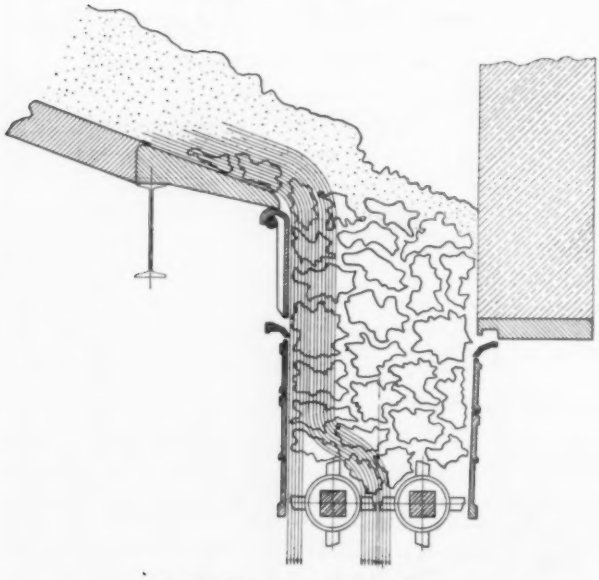


FIG. 9 DOWNWARD FLOW OF AIR

preheated air than with cold air. If the fuel in the retort swells slightly and tends to stick in the retort, as it may easily do with preheated air, any of the various designs of retort mechanisms yet produced lose part of their effectiveness in moving the coal. Readjustment of the motion of secondary pushers or other retort mechanism must be made to compensate for such conditions, and when the conditions disappear, further readjustment becomes necessary. Therefore the amount of operating attention required is considerably increased, particularly where a long retort is used. The longer the retort, the greater becomes the problem of handling fuel mechanically. Some manufacturers have limited the length of the retort and are applying a different type of grate surface to complete the stoker to the desired length.

(5) Present-day knowledge limits air temperatures to 350 deg. fahr. for satisfactory operation of a stoker of average length—that is, a stoker with retorts 9 or 10 ft. long, with semi-bituminous coal. It probably will be necessary to make rather radical changes in the basic method of handling fuel if this limit is to be raised.

(6) There is bound to be some increase of stoker-maintenance costs when preheated air is used. Present-day knowledge indicates that this increase is small and of lesser magnitude than the variation in costs between different plants operating under very similar conditions.

The economics of the local plant layout, the load factor of the plant, and the price of fuel must be evaluated to determine if the heat recovered by the preheater will pay adequate return on the required investment. Such matters are beyond the scope of this paper.

FURNACE VOLUME

The part that adequate furnace volume plays in efficient operation has long been recognized. The authors are mentioning the subject in order to present a more specific analysis of the problems than usually is made.

A number of variant values of the permissible heat release per cubic foot of furnace volume have been advanced. The authors recently assembled for analysis a large number of tests of stoker-fired boilers. It was found that, while the short stokers produced excellent results with surprisingly high heat release per unit of volume, the longer stokers suffered losses from incomplete combustion when operated with the same high heat releases.

This is a perfectly natural condition and is to be expected. The operator is less able to see flaws in the fire of a long stoker than in that of a short one. Areas of fuel bed of too great thickness may go undetected. The control of fuel distribution over a long stoker is less readily obtained than on a short stoker. All these factors render it necessary to increase the furnace volume in order to balance and compensate for imperfections in the fire.

Fig. 3 shows a graph of values of heat release per cubic foot of furnace volume for maximum efficiency with different lengths of stokers. Greater heat-release values may be employed where available space limits the volume of the furnace, but a lowered efficiency should be expected.

In comparing different designs of furnaces and in calculating the heat release per unit of volume, due consideration should be given to the relative value of different sections of the furnace. It is possible to put much useless space into a furnace and show a deceptively low rate of heat release.

If a boiler furnace be laid out with enough of the boiler surface to show the first baffle, one can readily detect dead areas in the furnace space. Figs. 4, 5, and 6 are examples. Heat-release rates should be compared on a basis of useful volume.

FURNACE WALLS

Much has been claimed for the use of water-cooled furnace walls. These claims have included reduction of furnace maintenance, reduction of stoker maintenance, increased heat absorption with consequent lower flue-gas temperatures, reduction of clinker trouble, reduction of radiation losses, and increase of stoker capacity. The authors have not seen conclusive and authoritative figures which place definite values on these various claims. Observation of various water-wall applications prompts the following statements:

(1) Water walls are a valuable aid to the stoker when they provide a clinker-proof wall at the edges of the fuel bed. Aside from this, they are of no value to the stoker burning semi-bituminous coal.

(2) The rear wall of the furnace should receive first consideration in the application of water walls. Insurance against freezing-over of the ash-discharge section is the most valuable function that a water wall can perform. Should more extended use of walls be justified, they should be installed so as to clinker-proof the sides of the stokers. The authors see no justification for the use of a water-cooled front wall.

(3) Semi-bituminous coal burns best if the furnace temperature is high. Low furnace temperatures tend to increase losses from incomplete combustion of gases and unburned solid particles of fuel carried up the stack. A true measure of this effect has not yet been made available, but there is ample evidence that the tendency exists. Naturally, such effects can be offset by increasing the furnace volume. The authors do not believe that there is any justification for the statement that water-cooled furnace walls permit an increase of the rate of heat liberation per unit of furnace volume; rather that the reverse is true.

(4) For present-day practice, therefore, in the opinion of the authors, water walls should be designed and applied in a manner that will not unduly lower the temperature of the furnace.

(5) Some users have found that the use of water walls, particularly in applications where the furnace temperature was materially lowered, resulted in decreasing the amount of slag deposited on the lower tubes of the boiler. The authors believe that it is quite possible to do this without seriously affecting combustion, particularly by arranging the water walls so that they exert a maximum cooling effect on the space just below the boiler and a minimum effect on the space just above the fire. Fig. 7 shows such an application at the plant of the New Haven Pulp and Board Company.

Ash-Disposal Apparatus

The authors will not attempt to compare the advantages of the various methods of disposing of the refuse, such as car haulage, sluicing, the use of chain or belt conveyors, or steam jets. Any method can be satisfactory so far as the stoker is concerned. There is only one requirement, and that is that the ashpit below the stoker must be effectively sealed to hold internal pressure at all times, except for the brief interval necessary for removal of the refuse.

If the ashpit is not sealed, there frequently may be a downward flow of air from the lower end of the stoker through the dump grates or clinker grinder, as shown by Figs. 8 and 9. In its passage the air may burn enough fuel to raise itself to a temperature high enough to destroy the parts of the ash-discharge section. Many cases of high maintenance of dump grates or clinker grinders have been traced to this cause.

Proper sealing of the ashpit, confining the air and gas until a slight pressure builds up in the ashpit, stops this action.

Practically all manufacturers of ashpit gates, sluicing systems, etc., are now building such apparatus in a manner that provides satisfactory sealing.

Waste Fuel

Many plants find themselves in possession of quantities of waste fuels, such as pulpwood bark, sawdust, scrap wood, domestic-size coke, coke breeze, anthracite waste, etc. Generally the available amount of such fuel is not sufficient to pay a good return on the investment in a separate boiler and furnace designed especially for its utilization. Such fuels can be burned quite efficiently and economically with an underfeed stoker.

If the fuel is of a fibrous nature, liable to clog the rams, it can be burned by dropping it on top of the fuel bed, through chutes in the front wall. Such a scheme will work well with hogged wood refuse or similar material.

Fig. 10 shows such an installation at a new plant in the South. In making such an application the following should be observed:

(1) The chutes in the front wall should center on equal portions of the stoker width. They should be located from 7 to 10 ft. above the grate surface and should be spaced not more than 7 ft. apart. Such precautions are necessary to assure fairly even distribution of waste fuel over the stoker fire.

(2) Provision should be made to admit the waste fuel into the furnace as nearly continuously as possible, so that uniform combustion conditions can be maintained. Blowing the fuel into the furnace with a current of air is unsatisfactory, unless

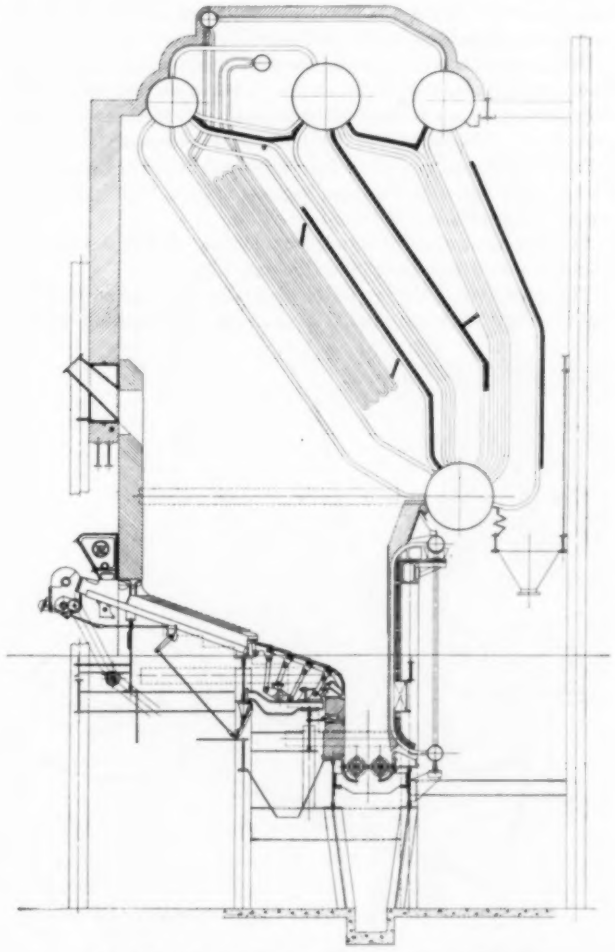


Fig. 10 Design for Burning Waste Fuels

the fuel is actually pulverized. Even very small chips of wood will drop to the bottom of the furnace, leaving the conveying air current to pass upward from the chutes as useless excess air. The better method of supplying the fuel is to have some sort of feeder, such as a paddlewheel, which drops the fuel into the chute at a controllable rate, relying on gravity to feed the fuel through the chute. By maintaining proper draft conditions in the furnace, the amount of air infiltration through the open chute can be made negligible.

(3) The chute should be divergent; it should increase in area as it approaches the furnace, in order to prevent clogging. It is preferable to water-cool the edges of the chutes next to the furnace. The amount of heat thus absorbed can be made so small that the cooling water can be discharged to waste without undue loss.

(4) Generally speaking, provision should be made for the injection of considerable quantities of free air over the fuel bed, under the control of the operator. This is particularly necessary if the waste fuel contains resin, oil, or similar substances.

If the waste fuel is granular or brittle in character, as coke breeze, anthracite waste, etc., it is burned to best advantage when mixed with the coal.

Such mixtures are hard to make and maintain in the bunkers. Coke, particularly, has a pronounced tendency to separate from the coal, so that the stoker receives alternate batches of coal and coke from a bunker mixture.

Figure 11 shows a mixing device which has been quite successfully used. In this device, the stoker ram feeds, with each charge, definite amounts each of coal and other fuel.

The hopper is divided, above the ram, into two compartments. The compartments are separated by a movable partition. The front compartment is supplied with coal, and the rear one with other fuel. The position of the partition determines the relative proportions of coal and other fuel which will be fed in by the ram. By the time the retort mechanism has fed the fuel up into the fire, the two fuels are well mixed.

The ram is always surrounded by coal, so that it is protected from the abrasive action of such fuels as coke.

The capacity of the unit will generally be reduced if waste fuel is mixed with the coal. By making the partition quickly

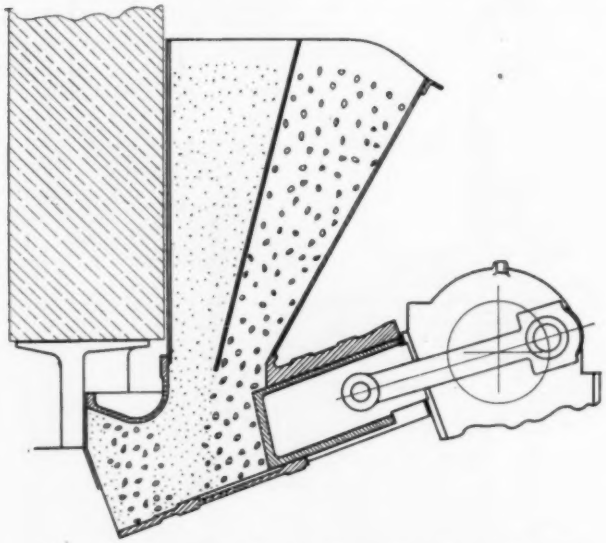


FIG. 11 DEVICE FOR MIXING COAL WITH OTHER FUEL

adjustable, the peak capacity of the stoker can be retained. The waste fuel can be cut off and the stoker operated on coal alone during the periods of heavy load.

OPERATING METHODS

STOKER REGULATION. A few simple rules for the best operation of underfeed stokers should be used as a basis for organizing and training the boiler-room personnel. They are:

(1) The rate of coal feed should be proportional to the rate of burning, so that uniform combustion conditions are maintained. The operator should be provided with some sort of fuel meter which he can compare and correlate with the steam-flow meter. This may be a table of speeds for different stoker motor controller points, an indicating device operated by the stoker engine or stoker turbine governor, or some form of tachometer. Any automatic controls of stoker speed should be frequently and carefully checked to assure that they maintain this relation.

(2) The stoker should be protected from the heat of the fuel bed by a layer of ash. This condition is sometimes hard to maintain with New England fuels, because of their low ash content, so that stoker maintenance is sometimes higher in New England plants than elsewhere. The operators should be trained

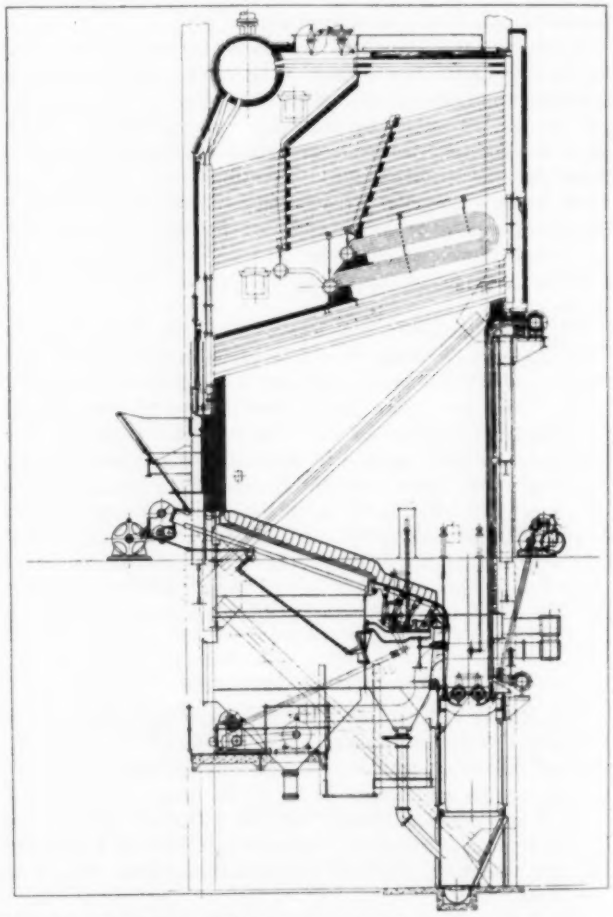


FIG. 12 STOKERS IN HUDSON AVENUE STATION OF BROOKLYN EDISON COMPANY

to watch this factor, and it is not improbable that the majority of supervisors could well give the matter study.

The principal things to watch are:

(*a*) With clinker grinder stokers, keep the level of the refuse in the pit at a sufficient height to hold an adequate layer of ash on the lower end of the stoker

(*b*) With dump-grate stokers, care should be exercised that the fire be moved back not too rapidly after dumping the refuse. If the refuse on the lower end of the stoker is suddenly shoved over to the dump grate, the lower end of the stoker and the dump grates themselves may be unduly exposed to the hot fuel, with resultant burning of parts.

(3) The underfeed stoker is essentially a machine for producing complete combustion of the fuel in all parts of the furnace. In other words, there should not be a stratum of gas over one section of the stoker containing an excessively high percentage of CO_2 and some CO, and another stratum over another section containing an excessively high percentage of oxygen. Losses from both incomplete combustion and excess air occur under this condition. Also the section of stoker having too little excess

air may be overheated. The distribution of fuel over the grate area should be such that combustion conditions are substantially the same in all parts of the furnace, except over the ash-discharge section.

Banking

It has long been a custom, accepted as being best practice, to bank boilers when the load drops below a certain point, say 100 per cent of rating, leaving other boilers to operate at higher loads.

Some years ago this was probably the best thing to do, because there were no instruments in the boiler room to guide the fireman. The only way to make him fire properly was to throw a heavy

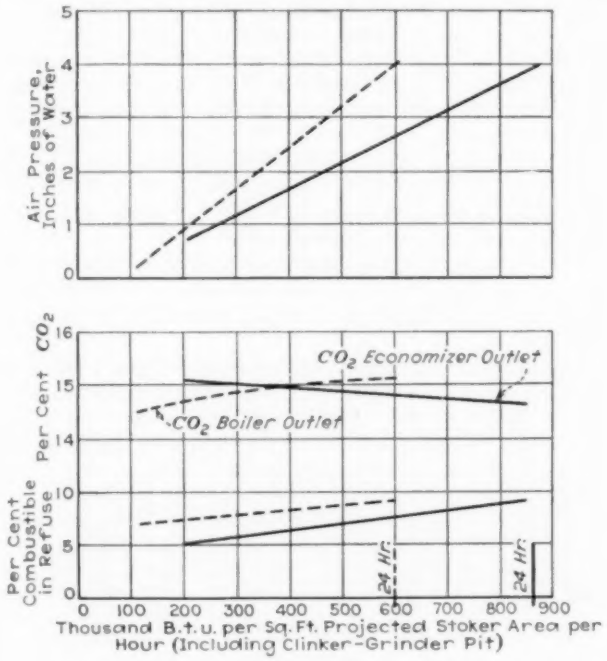


Fig. 13 Comparison of 1928 and 1923 Designs
(Broken line, 1923 design; solid line, 1928 design.)

load on his boiler so that he had to have a good fire to hold the steam pressure.

Basically and practically, this system is wrong. Modern stokers are capable of operation over a range in load of 6 or 7 to 1, with substantially the same combustion efficiency at all loads. Instead of banking some boilers and operating others at fairly heavy loads, all boilers should be operated at the same load. The operators should be provided with proper instruments and be held responsible for maintaining efficient combustion at all loads. Such practice will result in material economies, for several reasons:

(1) Due to highly efficient heat absorption by the boiler at light loads, the boiler-room efficiency is higher at such loads than at heavy loads.

(2) Banking losses are eliminated.

(3) The stoker parts are cooled by air at all times, instead of being exposed to the heat of a banked fire without a supply of cooling air.

(4) All fires are instantly ready to take a sudden increase of load.

The authors know of several plants burning Pocahontas coal which operate at 50 per cent of boiler rating, 24 hours per day, for weeks at a time with an efficiency of boiler, superheater, and stoker of 83 per cent. At other times these plants carry 250 per cent of rating on the same equipment at an efficiency of 78 per cent.

Boiler Operation

The authors believe that this paper would be incomplete unless mention were made of the difficulties experienced from the formation of slag masses on the lower tubes of the boiler.

Much theorizing about the causes and reasons for this slag formation has been indulged in by engineers. The various boiler manufacturers have produced slag screens which, in most cases have effected marked improvements. Nevertheless, the problem remains, even though it be in a modified form. Slagging of tubes is of course due to the air current lifting ash particles out of the fuel bed and subsequently depositing these particles on the tubes of the boiler. Remembering again the laws of physics, it would be expected that the slag formation would vary approximately as the square of the velocity of air through the fuel bed,

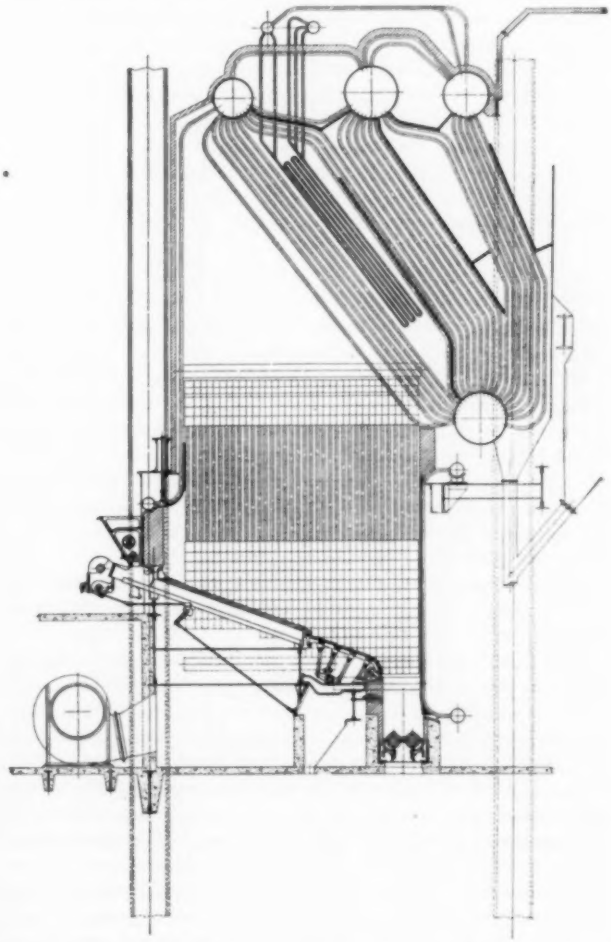


Fig. 14 Stoker Installation of Duquesne Light Company

or, in other words, as the square of the combustion rate. Therefore the only control that the stoker can exercise over slagging of the tubes is to operate with a maximum activity of fuel bed so that the air velocities through the fuel bed are as low as possible.

Semi-bituminous coals are bad tube slaggers for two reasons:

(1) The slowness of these coals to ignite reduces the active burning area of the stoker and therefore increases air velocities through the fuel bed.

(2) The ash in the majority of these coals is relatively light and easily carried away by the air current.

Therefore it is the authors' opinion that all boilers should be equipped with means for removing the slag from the lower tubes while the boiler is in operation at light loads or is banked. By far the most effective means of slag removal is to lance the tubes through doors located at the top of the furnace just below the boiler tubes. Such doors are readily installed with either brick walls or water walls.

A walkway should be provided so that the men can easily reach all parts of the lower tubes with a lance. Convenient connections for the lancing medium should be provided at both sides of the boiler so that two or more men can work simultaneously and keep the lancing period as short as possible.

Eastern operators favor compressed air or steam for lancing the tubes. Either of these mediums is rather slow in action. Western operators use a water lance with a remarkable degree of effectiveness. This water lance consists of a pipe turned up at the end and supplied with water so regulated that a tiny stream issues from the end of the lance. The stream of water need not be over 6 in. long for maximum effectiveness. The only precaution necessary is to regulate the water so that it is not sprayed against the joints of the tubes in the headers.

By such means the lower tubes of the boiler can be kept relatively clean at all times, so that the draft loss of the boiler will remain reasonable and there will be a minimum of slag formation on the upper tubes of the first pass of the boiler.

Recent Stoker Installations

Fig. 12 shows an installation in 1928 of stokers in the Hudson Avenue Station of the Brooklyn Edison Company. These stokers are 15 retorts wide and have an inside length of 19 ft. 9 in. They burn semi-bituminous coal with cold air in a furnace having refractory-covered water walls for side and rear walls and a solid refractory front wall. These stokers have developed in excess of 300,000 lb. of steam for 24 hours and in excess of 360,000 lb. of steam for 2 hours, on official tests. The heat-liberating rate for the 24-hour period is in excess of 825,000 B.t.u. per sq. ft. of grate area and is approximately 53,000 B.t.u. per cu. ft. of useful furnace volume.

Fig. 13 shows the results of official tests of one of these 1928 units compared with the results of similar tests on stokers designed in 1923.

Both equipments were tested (in different plants) by the ascending-descending series method. That is, individual tests were run at different loads, but were run as a continuous series, without gaps between tests, starting with a test at light load and running succeeding tests at successively increasing loads until the maximum load was reached, then reversing the process and running tests at successively decreasing loads until the last test is run at the same load as the first. This method eliminates the errors due to storage of coal on the stoker. It also eliminates the inaccuracies and deceptive results which occur under the common practice of burning down and cleaning the fire between tests.

The heat-absorbing equipment used with the two different stoker equipments is not comparable, so the comparison is made on a basis of combustion efficiency.

The soot and cinder loss was not measured on the 1923 design, but the heat balances indicate that it was about the same at the 24-hour capacity of that stoker as it was at the 24-hour capacity of the 1928 design—that is, about 3.5 per cent.

Air pressure is not necessarily a measure of air velocity through the fuel bed, but it is interesting to note that the same air pressure is used at the 24-hour capacity of both equipments.

However, it is interesting to note the progress of the art in increased capacity per unit of stoker area. The 24-hour capacity of the 1928 design is 1.4 times that of the 1923 design, and approximately the same combustion efficiency is obtained at the increased capacity.

Fig. 10 shows a stoker installation now being made at the Mobile Steam Plant of the Alabama Power Company. This installation is being equipped to burn wood refuse with high-volatile bituminous coal and preheated air.

Fig. 14 shows a stoker installation soon to be made at the new Reed Station of the Duquesne Light Company, Pittsburgh. These stokers are 18 retorts wide and have an inside length of 19 ft. 9 in.

These three recent installations are equipped with a new type of stoker which is an underfeed stoker with an undulating link-grate overfeed section. This design permits the construction of extremely long stokers, and at the same time eliminates the problems of handling fuel through extremely long retorts.

Fig. 7 shows an installation recently placed in service by the New Haven Pulp and Board Company, New Haven, Conn. This is an eight-retort stoker of the straight underfeed type with an inside length of 14 ft. 8 in. New River coal is burned with preheated air. One unit now carries the entire load that formerly was carried on several smaller stoker-fired boilers.

Conclusion

The authors have endeavored to present and to discuss the factors that are important in making a successful stoker installation. Many of the points discussed receive recognition in the majority of plants, but it has been their observation that relatively few plants adequately recognize all of them. It was with the idea of bringing all of these practical matters together in one group that this paper was written.

While the paper has been presented by individuals, it does not represent the individual thought or effort of one person; rather it is the accumulation of the experience of the authors and their associates.

Soot Particles in New York City Air

By E. E. FREE,[1] NEW YORK, N. Y.

In this paper the author tells of the methods used and the results obtained in a series of counts of soot and dust particles in the air of New York City. The counts were made by an impact dust counter. Two variables affect the results. One is the wind and the second the degree of clumping or flocculation of the soot particles in the air. The number of soot particles counted showed a high degree of correlation with wind direction. Nothing was discovered to show the causes for clumping of the soot particles.

IN APRIL of this year there was started a series of approximately daily counts of soot and dust particles in the air of New York City. During April and May counts were made at a number of different stations on the Island of Manhattan, to get some idea of the geographical distribution of air particles. Subsequently, most of the counts have been made from the seventeenth floor of an office building at Twenty-Third Street and Fifth Avenue. The earlier counts at other localities show that this station is not unrepresentative of average air in the central part of Manhattan.

All the counts have been made by an impact dust counter, slightly modified from the instrument devised by Dr. J. S. Owens, of England. In these instruments a definite volume of air is first moistened by being allowed to remain for a few moments inside a tube lined with wet blotting paper. A definite volume of this air is then drawn by a suction syringe through a small orifice. The expansion of the air thus produced cools it, and condenses moisture on any particles which are present. These moistened particles impinge on the surface of a glass microscope slide, a short distance beyond the orifice. Being moist the particles stick to this glass surface. The slide is then removed and examined under the microscope, the numbers of particles being counted in the usual manner.

The modification of the instrument used in these counts, as compared with the standard Owens counter consists in the use of a circular orifice instead of a narrow slit. This is in order to obtain, it is believed, a better estimate of the aggregates of soot particles. The slit instrument breaks up these aggregates more generally than the instrument with a round hole. Counts of particle numbers obtained with the slit instrument are larger than those obtained with the round-hole instrument for the reason that the latter instrument counts aggregates as one particle each, while the slit instrument counts each individual particle into which the aggregates are divided.

It is believed that for practical purposes it is desirable to count the aggregates as one particle each rather than dividing them into their constituents.

In the instrument used in these counts the volume of air tested was in some instances 5.9 cu. in., in others 2.95 cu. in., depending upon the amount of dust in the air. In either case, a proper factor is applied to convert the measured counts into numbers of particles per cubic foot of air.

One hundred and forty-two counts have been made, many in duplicate. In general, these have been taken approximately at midday, although the hour is of little importance as the 24-hour variation is entirely obscured by other and greater variations.

Data obtained are chiefly characterized by very high variability, it being not unusual for the numbers of particles in the air to change by a factor of as much as ten, either way, within twenty-four hours. Instances have been observed of such extreme changes within two or three hours, usually correlated with changes in wind direction or velocity.

The maximum count obtained was on July 30, 1928, equaling 880,000 particles per cu. ft. of air. The minimum count was on July 2, 1928, and was 35,000 particles per cu. ft. The average of all counts during the period specified was 470,000 particles per cu. ft. of air. No great numerical accuracy is attachable to these figures, even to the averages. It is possible to say, however, that approximately a half-million dust and soot particles are to be found, on the average, in each cubic foot of New York City air.

The character of the particles observed, as judged by qualitative tests under the microscope, is remarkably uniform. Counted slides have shown never less than 96 per cent of particles which are clearly ordinary soot, that is, unburned carbon. This is for the level of the seventeenth floor. A slightly higher percentage of other particles is usually found at the street level. It is believed, however, that at least 90 per cent of all solid particles ordinarily encountered in New York City air are soot.

Other materials observed in individual instances include particles of rubber from automobile tires, of asphalt, of brick, of rock from excavations, of horse excrement and other organic materials, and slivers of glass at first believed to be distant volcanic dust, but later found to be too numerous for this. These are now considered to be minute flakes scaled off from glazed tile used on buildings. A few particles of leather have been found, a few cotton fibers, an occasional hair or skin scale apparently from a human being, and the usual relatively small number of what seem to be spores of living molds.

No identifiable soil particles have been found, no diatoms, and none of the so-called Tissandier spheres which are apparently tiny globules of metallic iron. The apparent absence of these usual constituents of air dusts from New York counts may be explained, however, as due to the coating of these objects with the omnipresent soot particles.

The sizes of these soot particles vary from approximately one ten-thousandth of an inch to approximately one fifty-thousandth of an inch. As stated, a relatively large percentage of these soot particles is observed to be gathered together in clumps or aggregates, like grains of popcorn stuck together to make a popcorn ball. These aggregates measure between one fifteen-hundredth of an inch and one five-thousandth of an inch. The percentage of the soot particles present in the air as aggregates has been found to be exceedingly variable. This variability is not believed to be due to any variation in method of collection or counting but to represent a real variability in the clumping together, or flocculation of soot particles in the air. This observation is believed to be new, and may have some importance in practical problems of city soot and city fog.

The effect of elevation above the street level is to decrease the number of soot particles in the air. The percentage of decrease is extremely variable but approximately 100 ft. of rise above the street level means between 25 per cent and 50 per cent reduction in the number of soot particles. The above figures for the particle counts should be multiplied, therefore, by between 1.5 and 2.0 to give the average number of particles at the street level. This conclusion applies to the Fifth Avenue and Twenty-Third Street

[1] School of Commerce, Accounts, and Finance, New York University.

Contributed by the Fuels Division and presented at the Annual Meeting, New York, N. Y., Dec. 2 to 6, 1929, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

district only; no high-level counts having been made at other points. As might be expected, there is some evidence that days on which the ultimate soot particles show much aggregation into clumps are also days of greater difference between the dust counts at the street level and at higher levels, indicating the relative rapidity with which the soot aggregates settle out of the air as compared with the settling rate of the ultimate particles.

Rain appears to have less effect on the particle counts than would be expected. Also, the effect is erratic. Some rains apparently produce virtually no decrease in air soot. Other rains produce relatively large decreases. It has not been possible to correlate these variations with rain intensity or with any other characteristic of the rainy days.

All of the data are consistent with the idea of two chief variables; the wind and the degree of clumping of the particles. The wind is undoubtedly the more important. Geographical counts make this evident. When the wind is blowing, for example, across Manhattan Island from west to east, the air of the west shore of the Island is relatively low in soot particles. At the same time the air of the east shore is relatively high. Similarly, the number of soot particles counted at Twenty-Third Street and Fifth Avenue shows a high degree of correlation with wind direction and wind velocity. Winds from smoky areas mean high soot counts. Winds from directions in which little smoke is being discharged usually mean low soot counts.

Nothing has been discovered, as yet, concerning the causes of the variation in the clumping or flocculation of the soot particles in the air. Nor is it possible to say, as yet, how important this factor is in removing soot particles from the air, as compared with the velocity and direction of wind. There is some suggestive evidence that the city experiences more haze and fog when the degree of flocculation of the soot particles is low. If true, this would be highly important in aiding the prediction of fog, now possible only with considerable inaccuracy on the basis of wind direction. However, until more has been learned of the reasons for the variation in soot flocculation it will be impossible to make use of these ideas with much assurance. Several theories have suggested themselves concerning the cause of these flocculation variations such as differences in humidity, differences in sunlight reaching the particles, differences in air ionization, etc. None is supported, as yet, by direct evidence. Obviously, the problem deserves further investigation.

Little practical importance attaches to the calculation of the total amount of suspended soot in city air. It may be said, however, that the average over New York City, on the basis of these counts appears to be approximately one ounce, Avoirdupois, for each 10,000,000 cu. ft. of air. For the air over the entire city the amount is from four to five short tons. Ninety per cent or more of this is soot. The total probably varies, with the wind and other conditions, from about one-tenth of this value to about three times the value. Discrepancy with previous much larger estimates which have been published is believed to be due to the fact that these previous estimates, made on the basis of solid materials caught by air-washing plants, include a considerable fraction of roof-fall of cinders and similar particles which remain only a brief time in air suspension. These are not really a part of the normal air dust of the city.

As to the ultimate sources of city soot, the relative absence of ash confirms, so far as it goes, the opinion easily formed from direct observation; namely, that most of the city's soot and air dust comes from many small fires in apartment houses, office buildings and homes, not from the smoke stacks of the large power houses or other industrial plants. This conclusion is supported, as is well known, by much other evidence. The present investigation while agreeing with this, provides few additional facts.

This is a preliminary report. The soot counts will be continued during the winter and probably for a considerable portion of the coming year. It is planned, also, to compare these counts more definitely than hitherto with weather conditions, with the receipt of ultraviolet radiation at the building level in the city, and with other similar factors. It is a pleasure to acknowledge the assistance of C. A. Johnson and Bernard Stoll of my laboratory. Mr. Stoll has made the majority of the counts.

Discussion

DAVID R. MORRIS.[2] The Weather Bureau has been making dust counts at 12 noon daily for nearly a year at 13 of its stations with the Hill dust counter. It is thought that this is the first concerted count using identical technique to be taken in this country. The instrument used is similar to the one used by Dr. Free, but under the Hill method the count is made with a magnification of 80 instead of 1000. The work is in cooperation with the American Society of Heating and Ventilating Engineers, which is also receiving health statistics from the cities where the counts are made.

In New York City the count is made at the Central Park Weather Bureau, and the results are published in the Monthly Bulletin of the observatory. Numerous weather data are printed alongside the dust observation at the time of the count and for some hours previous to the count.

The importance of dust in producing various weather conditions has been mentioned in meteorological textbooks for a long time. This was mostly the dust from wide country areas or volcanic erruptions. The generic name "dust" still holds today, but the character of the particles and their source are now quite different, especially in our large cities. It is in the latter regard that the most interest exists today and which may be expected in time to yield improved apparatus and the adoption of methods which will put the entire work on a more comparable basis.

HENRY KREISINGER.[3] It is stated that 96 per cent of the dust particles in the New York City air are soot coming from small fires in apartment houses, office buildings, and homes, and not from the smokestacks of large power plants and other industrial plants. It would be interesting to know to what extent the automobile vehicles contribute toward loading the New York air with soot. There are thousands of these vehicles running over the streets of the city under conditions very adverse to good combustion. The automobiles have to run slowly and they also frequently stop and start up again, conditions which make it necessary to operate the engines with excess fuel. Even under the best operating condition the combustion in an automobile engine is far from complete. Numerous tests made by the Bureau of Mines and other laboratories show that normally one half of the carbon of the fuel supplied to the automobile engines burns to CO. It is to be expected that when CO is formed there is also a large amount of soot formed and exhausted into the air. Let it be assumed that about a hundred thousand automobiles are running over the streets of New York during the busy hours. Under the unfavorable condition of operation, each automobile consumes about two gallons of gas per hour. This would make the gasoline consumption about two hundred thousand gallons, or approximately one and a half million pounds. This amount of gasoline is equivalent to about two million pounds of coal, or about one thousand tons of coal per hour. It takes a good-sized power plant to consume one thousand tons of coal

[2] New York Meterological Observatory, Department of Parks, Central Park, New York, N. Y.

[3] Research Engineer, Combustion Engineering Corporation, New York, N. Y. Mem. A.S.M.E.

per day. These figures may not be correct, but they indicate the extent of soot contribution by the running automobiles toward the sooty atmosphere of New York City. Moreover, the soot made by automobiles is discharged into the atmosphere at the street level and is more apt to find its way into the offices and dwellings. Soot from chimneys is discharged at higher elevations and is more likely to be carried away by the wind.

The paper states that there is no appreciable soot reduction in the atmosphere after rain and that the main factor tending to clear the atmosphere is wind. It is a common experience that soot and all very small particles floating in gases are difficult to wet and wash out. If such washing is to be successful, the dust particles must be wetted by the condensation of water vapor. If the gases containing the dust particles are first saturated with water vapor and then cooled below the condensation point, the water vapor will condense on the surfaces of the dust particles and wet them, thus increasing their agglomerating property and their weight. When in such state they can be more easily washed out or separated from the gases. The principle of condensation of water vapor on the dust or soot particles is used by the author of the paper for collecting the dust in the air samples.

PHILIP DRINKER.[6] The author discusses the presence of aggregates or clumps of soot particles in the air sampled and mentions the high correlation between counts and wind velocity. We have found in laboratory studies[8] that the settlement of fine particles, of the order of 0.5 to 5 microns, especially particles of carbon, is accomplished mainly by aggregation and that turbulent air motion or ionization will accelerate this aggregation. Whether or not this fact, well known in chemical warfare,[9] has a direct bearing on Dr. Free's problem, is not known.

In our experimental work, we have had abundant experience in the use of Owens' standard slit instrument, but have not used the modified Owens counter that the author describes. Our observations[7] indicate a rather low partial correlation coefficient between dust counts and wind velocity. In calculating these coefficients, we used routine counts taken twice daily for 16 months. Kimball and Hand[8] and Owens,[9] using data of a more selected nature, showed that a distinct inverse, but non-linear, relationship existed between counts and wind velocity. Their data would seem to agree with those in this paper.

It is hoped that, in his final paper, Dr. Free will work out and publish the partial correlation coefficients between counts, wind velocity, visible and ultra-violet solar radiation, and per cent relative humidity, and that he will be able to publish his final results in c.g.s. (and microscopist's) units.

C. HAROLD BERRY.[10] To the engineer it may seem that the presentation of data on a quantity that varies from 5 to 100 in no very systematic fashion is of little value, but it should be borne in mind that we are on the threshold of an important and difficult study. Its importance to the public is evident. No less is its importance to industry. Information is needed to refute false charges and to serve as a basis for the removal of grounds for well-founded charges. Air pollution is too great a price for the public to pay for the carelessness or indifference of industry, if these are the causes. On the other hand, if air pollution is largely caused by small furnaces of which the domestic heating plant is typical, the fact should be proved beyond doubt. In either case, it is the task of the engineer to remedy the situation.

The wide variability of results doubtless reflects the fact that we are dealing with a quantity that actually does vary widely in place and in time. This, combined with the difficulty of securing a small representative sample, makes precise measurements exceptionally difficult.

It is to be hoped that much more work will be done along this line in the near future.

LOWELL ANDREW.[11] The paper adds valuable information to the very limited knowledge of the amount and character of the microscopic particles present in the air in large cities. In some respects, however, the paper might have contributed more tangible information had it described in greater detail the means used of identifying under the microscope the particles observed and had presented further information on the size of the orifice used and its effect on the number of particles counted as compared with the Owens machine using the slit. Observation and experience with this type of instrument and the Hill instrument has been that the degree of magnification used in counting must be specified before the figures presented for dust counts can assume any relative value.

Extreme variability from day to day seems to characterize observations with jet dust counters, but as yet we have been unable to observe any great seasonal variations such as one might expect—for instance, between summer and winter.

We have not been able to get satisfactory comparisons between counts with the Owens type counters using high magnification of 1000 diameters and the Hill type using a magnification of only 80 diameters. This is no doubt due to some extent to the variable characteristic of aggregation in large particles.

One particle per cubic centimeter is equivalent to approximately 28,000 particles per cubic foot, so that the minimum count of 35,000 particles, as mentioned in the paper, represents only about $1^1/_4$ particles per cubic centimeter. The widely quoted figure for a certain city is 17,600 particles per cubic foot or $^2/_3$ particle per cubic centimeter. However, with an Owens counter and a magnification of 1000 diameters, counts as high as 152,000,000 particles per cubic foot have been obtained.

Dust counts are being made this winter with the Owens type of instrument using a slit and the particles counted under a magnification of 1000 diameters. It is to be hoped that further, better, and more reliable methods of determining the amount and character of the particles in the air will soon be developed.

LAURA A. CAUBLE.[12] The National Conference Board on Sanitation is made up of men and women, scientists, physicists, department of health officials, physicians, organized bodies of womens' clubs, and home makers. We are making a determined inquiry into the question of the increasing pollution of the air of cities with the consequent loss of sunlight and the vital ultra-violet.

---

[6] Assistant Professor of Ventilation and Illumination, Harvard School of Public Health, Boston, Mass.

[8] "The Effect of Turbulent Air Motion and of Humidity on the Stability of Dust, Fume, and Smoke Clouds," by P. Drinker, R. M. Thomson, and J. L. Finn, *Journal of Industrial Hygiene*, 1926, vol. 8, p. 307.

[9] "The Disappearance of Smoke in a Confined Space," by R. C. Tolman, E. B. Vliet, W. McG. Peirce, and R. H. Dougherty, *Journal of the American Chemical Society*, 1919, vol. 41, p. 304.

[7] Paper in manuscript, presented at Third National Fuels Congress.

[8] "Investigation of the Dust Content of the Atmosphere," by H. H. Kimball and I. F. Hand, *Monthly Weather Review*, 1925, vol. 53, p. 243.

[9] Eleventh Report of the Committee for the Investigation of Atmospheric Pollution, p. 42. H. M. Stationery Office, London, 1925.

[10] Professor of Mechanical Engineering, Harvard University, Cambridge, Mass. Mem. A.S.M.E.

[11] Assistant to Victor J. Azbe, Consulting Engineer, St. Louis, Mo. Jun. A.S.M.E.

[12] Chairman National Conference Board on Sanitation, New York, N. Y.

This body is very much interested in the study of dust that the author has been making. It has been measuring visible light for the last three years. We have used the Case photo-electric cell with a Leeds & Northrup potentiometer. Our records show that there is a constant average loss of 30 per cent of light in the city caused by the smoke pollution as compared with the light recorded in the country at a point not subjected to the same waste by smoke.

On the longest sunny day in June, in which there was 15.1 hours of sunshine, the records showed a loss of 42 per cent of visible light. The following day was continuously sunny, with a loss of 36 per cent of light. The difference between the available sunlight in the country and in the city is the measure of pollution by smoke which is being endured in the city. It is an intolerable burden for which relief must be found, and no body in the country is so capable of solving this problem as the mechanical engineers.

Our measurement of light is a continuous record from daybreak until sunset. There has been no such continuous measurement of dust made at any point so far as can be learned. These measurements are made momentarily over short periods of time, and they do not give any accurate measure of air pollution.

FSP-53-3

Boiler Settings for Burning Refuse Wood

By C. S. GLADDEN,[1] DETROIT, MICH.

This paper includes a discussion of: (1) Some recent developments in the design of boiler furnaces for burning green or wet refuse wood; (2) the experiences influencing the development of furnaces for burning dry refuse wood; and (3) factors influencing the design of wood-burning furnaces.

THE economic utilization of wood refuse from lumbering and woodworking operations has become a problem of increasing interest. Formerly, when fuels were comparatively cheap and lumber-cutting operations produced more waste wood than could be consumed, there was little concern given to efficient methods of burning it in steam-boiler furnaces. In lumber-sawing operations, the sawdust produced has in most cases been sufficient to generate all the steam required for operating the mill, and the slabs have been disposed of either by burning in large incinerators or working up into lath, kindling wood, or other commercial products. Later, as lumber and fuel became more expensive and woodworking plants producing finished parts from kiln-dried lumber became more numerous, improvement in the utilization of the wood refuse for the generation of steam became more imperative. Certain modern mills for the production of finished kiln-dried lumber from the log have arranged their plants so that the steam requirements for power and kiln-drying operations will quite nearly consume all of the mill refuse in its generation. By-product processes, such as the evaporation of salt, and the production of power to the extent that all of the refuse may be utilized in the generation of steam have been given increasing consideration.

The refuse resulting from the milling of air-dried or kiln-dried lumber produces a fuel which is especially well adapted for burning under steam boilers in properly designed furnaces, and there are now many installations which indicate the increasing efficiency that is being attained in the burning of this material.

In the plants of furniture and automobile manufacturers the demand for power and steam is such that the problem of properly burning the kiln-dried wood refuse is one that is worthy of serious consideration of the most efficient methods.

The wood refuse available for fuel may be considered under three general classifications:

1 The sawdust and slabs from sawmills containing from 30 to 50 per cent of moisture, where logs are transported to the mill dry, to perhaps 70 per cent moisture if the logs are of softwood and have been floated to the mill through rivers and streams or have been held in storage in mill ponds. The bark and shavings from pulpwood held in water storage are also very wet and contain over 70 per cent of moisture. Such refuse will be classified as wet refuse

2 The refuse of planing mills and finished-lumber operations resulting from the working of air-dried lumber containing from 15 to 25 per cent moisture

3 The wood refuse of furniture factories and automobile-body plants resulting from cutting and finishing of parts from kiln-dried lumber, in which case the refuse usually contains from 5 to 7 per cent moisture.

Combustion Temperatures and Calorific Values of Wood

We shall now consider the heating values and theoretical combustion temperatures of this material. It is apparent that in burning wet wood the fuel must first be raised to the temperature required for the evaporation of the moisture, then the latent heat for the evaporation must be supplied, and the resulting steam must be superheated to the final temperature of the products of combustion. A study of the combustion temperatures has been made of wood under different conditions of moisture content and of excess air for combustion. The results are shown in Fig. 1. The combustion temperatures are the temperatures which would be produced if there were no absorption of heat from the flame as the processes of combustion take place,

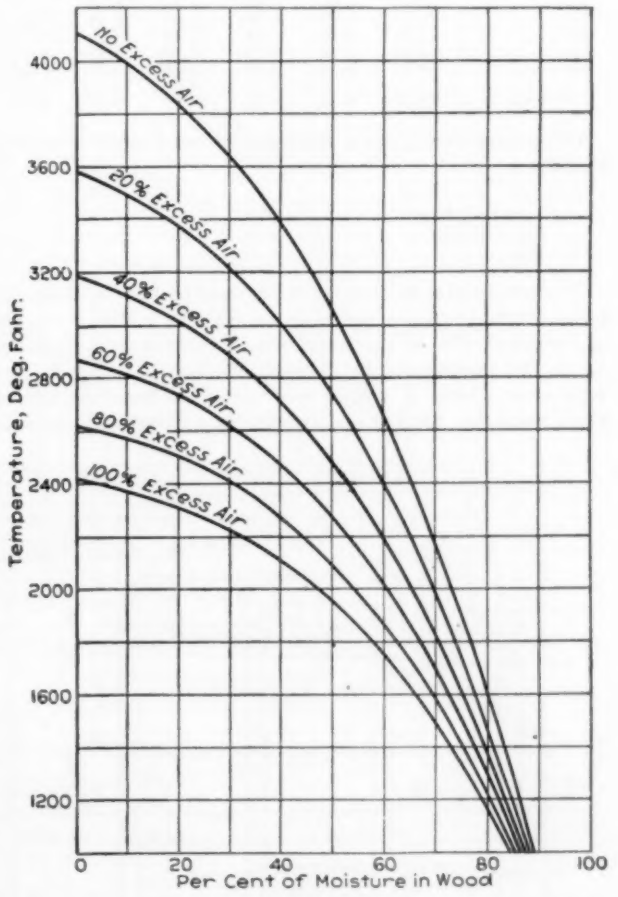


Fig. 1 Combustion Temperatures of Wood With Varying Moisture Content

[1] Director, Power and Maintenance Section, General Motors Corporation. Mr. Gladden was formerly associated with E. I. du Pont de Nemours & Co. at Wilmington Del., and with the J. G. White & Co., at New York, in steam power-plant design and construction work. Mem. A.S.M.E.

Contributed by the Power Division and presented at the Semi-Annual Meeting, Detroit, Mich., June 9 to 12, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

and the curves indicate the relative heat produced by the different classes of wood fuel with respect to moisture content.

The method used in determining the combustion temperatures is that outlined by Arthur D. Pratt in his "Principles of Combustion in the Steam-Boiler Furnace" in which the mean specific heats of the different constituents of the products of combustion vary, depending on the final temperatures attained. This method gives somewhat higher results with increasing moisture content than the formula frequently used in which the specific heat of the products of combustion is assumed to be constant regardless of the amount of excess air, moisture content, or of the final temperature.

The percentage chemical composition and calorific value of moisture-free wood have been taken as the averages of the data (by Gottlieb) given in Table 1.

TABLE 1 ULTIMATE ANALYSIS AND CALORIFIC VALUE OF DRY WOOD

Kind of wood	C	H	N	O	Ash	B.t.u. per lb.
Oak	50.16	6.02	0.09	43.36	0.37	8316
Ash	49.18	6.27	0.07	43.91	0.57	8480
Elm	48.99	6.20	0.06	44.25	0.50	8510
Beech	49.06	6.11	0.09	44.17	0.57	8591
Birch	48.88	6.06	0.10	44.67	0.29	8586
Fir	50.36	5.92	0.05	43.39	0.28	9063
Pine	50.31	6.20	0.04	43.08	0.37	9153
Average	49.56	6.11	0.07	43.83	0.42	8671

The proximate analysis of moisture-free wood refuse is roughly as follows:

	Per cent
Volatile matter	81.5
Fixed carbon	18.0
Ash	0.5

The calculated calorific value of dry wood by Dulong's formula, based on the above average ultimate analysis, is 7626 B.t.u. per lb. compared with an average of the calorimeter determinations for all the woods given by Gottlieb of 8671 B.t.u. per lb. and with several tests of hogged wood from an automobile-body plant averaging 8413 B.t.u. per lb. on a moisture-free basis. It has been demonstrated by various investigators that Dulong's formula does not apply accurately to wood, due to the presence of large percentages of hydrocarbons. In the case cited above the actual calorific value as determined by bomb calorimeter is 13.7 per cent higher than the calorific value calculated by the Dulong formula. Accordingly in computing the combustion temperatures of wood under different conditions of moisture content as given in the curves, the heating values obtained by the Dulong method have been increased by 13.7 per cent.

The actual combustion temperature in a boiler-furnace setting is lower than that indicated by these curves since the absorption of radiant heat by the boiler and the dissipation of heat by

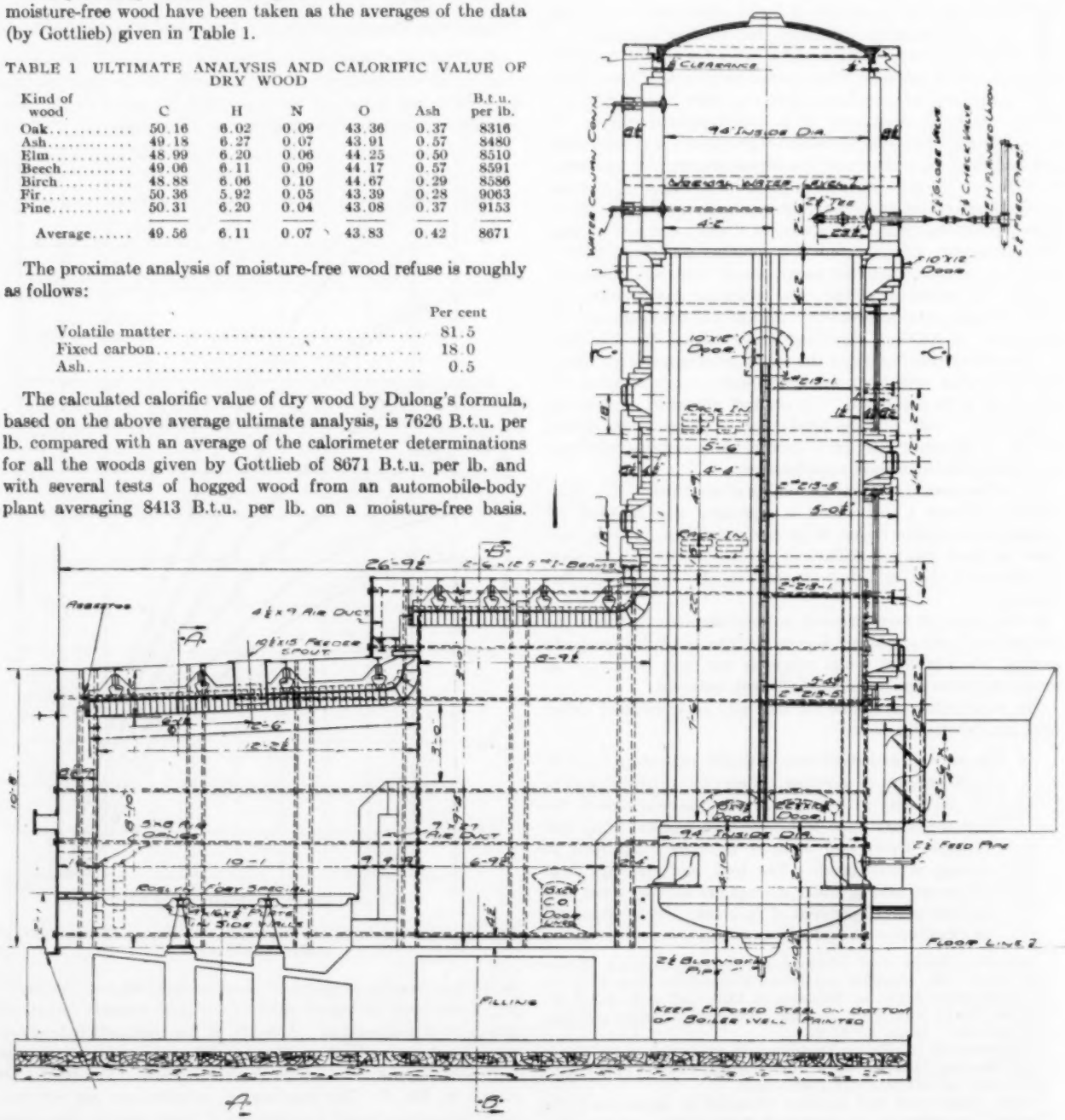


FIG. 2 FURNACE FOR SAWMILL REFUSE WITH VERTICAL WICKES BOILER

radiation and conduction through the furnace walls will prevent the maximum temperature of combustion from being attained.

On account of the high flame temperature of burning dry wood with little excess air, the problem of suitable refractories for wood-burning furnaces is of great importance. Although the ash of the wood is not of serious consequence, due to its small quantity, its silica and alkaline constituents combine to form a low-temperature-fusion slag which readily fluxes with the silica of ordinary firebrick. Some improvement has been obtained in the maintenance of wood-burning furnaces operating at high temperatures by the use of brick containing high percentages of alumina; but the most satisfactory results have been obtained both by the use of air-cooled refractory walls in which the temperatures of the refractories were maintained below the point of heat liberation. The air-cooled wall has greatly improved the life of the refractories in wood-burning furnaces, and the water-cooled wall has also been used with success. The temperature of the furnace should be maintained as high as possible to produce smokeless combustion, but not so high that the refractories suffer from fusion. It appears at the present time that a certain amount of water cooling together with air cooling provides the best type of furnace for air-dried and kiln-dried refuse wood.

The burning of wet refuse does not involve the question of life of refractories and the dissipation of heat from the furnace so much as the conservation of the heat of combustion and utilization of its radiant heat by means of reflecting arches and side walls to heat the refuse wood and evaporate the moisture in the fuel as it is delivered to the furnace.

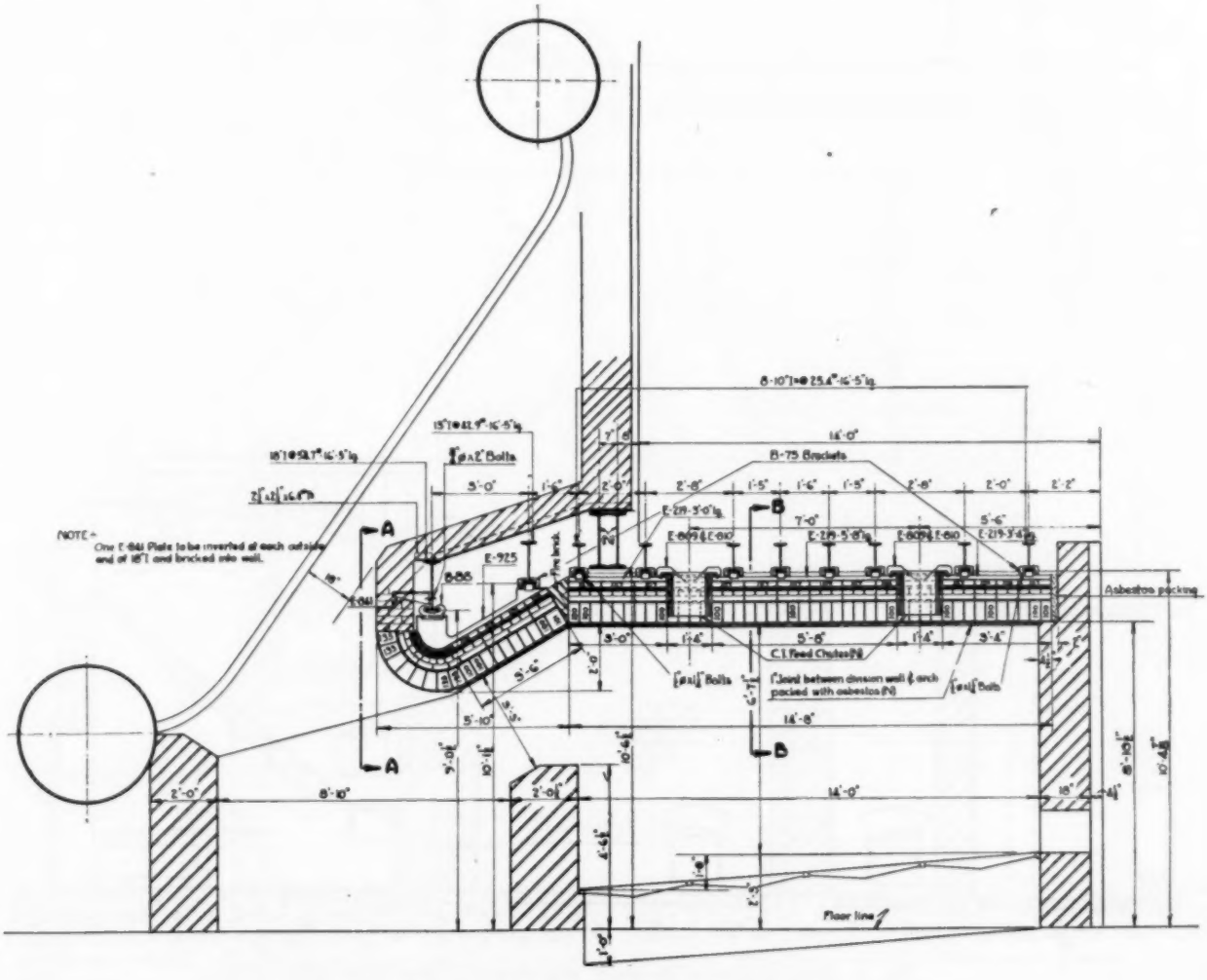


FIG. 3 FURNACE FOR SAWMILL REFUSE
(Louisiana Central Lumber Co., Clarksburg, La.)

at which the silica of the firebrick will fuse with the ash and cause erosion and by the introduction of water-cooling surface in the side walls of the furnace. Substantially the same development has taken place in the design of the side walls of dry-wood-burning furnaces as has occurred in recent years in the design of furnaces for burning pulverized coal. That is, the solid refractory walls were found to require high maintenance and frequent reconstruction, especially when the excess air is reduced to a minimum or the wood is burned in a dry state and at high rates

The early designs of boiler furnace for burning wood gave no consideration to efficiency of combustion or to the smoke nuisance. However, the present smoke-elimination ordinances of municipalities in which furniture, automobile-body, and other woodworking plants are located require the most careful consideration of the smoke problem.

The question of setting singly or in battery is also one worthy of serious consideration, depending on the extent of the operation, the single setting being considerably more flexible and easier

to maintain and repair than the battery setting with its troublesome common center wall.

Designs for Wet Wood Refuse

For refuse wood consisting of sawdust, shavings, and hogged wood, and slabs direct from the log or green lumber extended furnaces having reverberatory arches over the top should be provided. In the burning of wood as contrasted with the burning of coal, the combustion takes place from the surface of the pile of fuel in the furnace, while with coal the combustion is maintained throughout the fuel bed. This means that for burning green refuse wood the furnace should be provided with side walls and arches arranged to reflect the radiant heat back to the burning wood and thus promote the evaporation of moisture before combustion takes place.

A typical example of a green-wood-burning furnace installation is shown in Fig. 2. This furnace consists essentially of two combustion chambers. The fore oven is arranged with a suspended arch above the grates, on which the green fuel falls through the feeder spout. This pile of fuel is burned by air admitted above the grates by air openings near the front of the furnace side walls and through the portions of the grate around the pile of wood which is not covered too deeply. The products of combustion passing through the 3-ft. space above the bridgewall are mixed with additional air through the rear of the bridgewall, and further combustion of the gases and cinders takes place in the secondary rear oven. The wood is dried and combustion takes place from the surface of the pile, assisted by the heat reflected from the arch above and from the side walls. This typical setting for burning green wood is shown in conjunction with a vertical water-tube boiler. There are many of these boilers throughout the lumber-producing areas of the South and Northwest, primarily because this particular type of boiler was first introduced in these regions for steam production, and, secondly, it is a self-contained unit which can be shipped and erected intact and is therefore a cheaper installation than the modern type of bent-tube or horizontal straight-tube boiler which requires additional assembly and erection cost on the job.

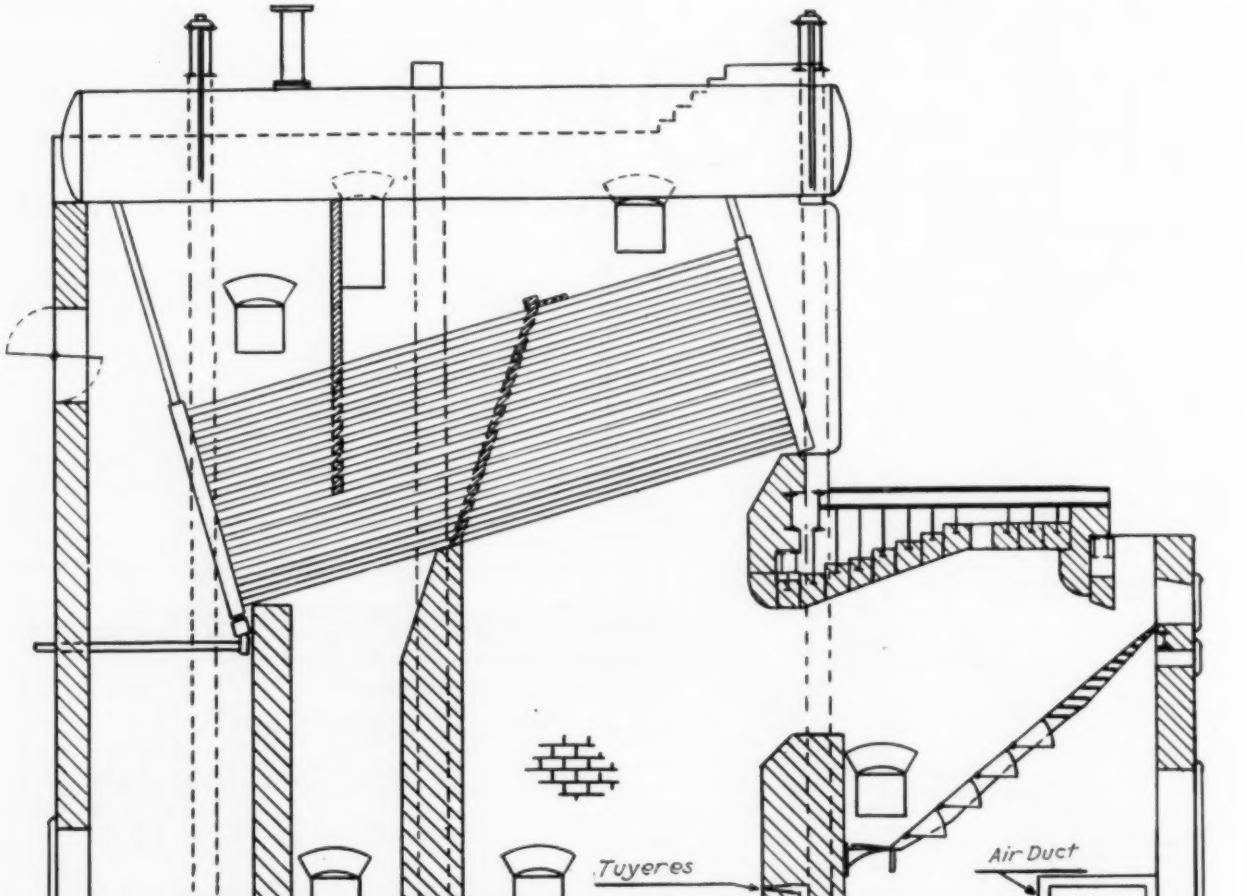


Fig. 4 Inclined-Grate Furnace for Burning Wet Refuse
(600-hp. Babcock & Wilcox boiler, Chicago Mill & Lumber Corporation, Tallulah, La.)

Fig. 3 shows a furnace in conjunction with a Stirling-type boiler designed for burning green wood in which the fuel is admitted through two feed chutes to horizontal grates. In this case the secondary chamber for the combustion of cinders and gases for the wood-burning chamber is located beneath the boiler. This is a typical installation of units of this type which are now being used in the Southern lumber regions. If

space in the boiler room permitted, however, the combustion chamber should be brought forward so that the entire space underneath the boiler could be used as a secondary combustion chamber.

The furnace shown in Fig. 4 for the burning of green or wet wood refuse is arranged with sloping grates so that as the wood is admitted through the feed chute at the top and front of the furnace, the upper surface of the pile of refuse in the furnace is approximately the same distance from the grate line from top to bottom.

The upper portion of the sloping grates which run longitudinally is fitted with horizontal steps. The lower portion is also of longitudinal grates alternating with sector fuel pushers between the grates which control the movement of the fuel toward the dumping grate at the rear. In this furnace the fuel burns progressively from the top to the lower end of the grate, and the slope of the grates is such that the fuel slides readily without avalanching. The thickness of the fuel bed is controlled by the width of the throat at the upper end of the grate. Doors in the front are also provided for hand manipulation as may be necessary.

The suspended arch over the combustion chamber is arranged to reflect the radiant heat back to the surface of the fuel bed and promote the drying and volatilization processes. The back end of the arch is dropped to further reflect the radiation to the pile of fuel and to mix the gases. The thinner and hotter gases at the lower end of the grates being forced to mix with the gasified volatile matter from the upper portion of the drying fuel, thus assist in promoting complete combustion. At the rear of the bridgewall there is also a line of tuyères to supply additional air for burning the gases in the rear combustion chamber beneath the boiler.

In Fig. 5 is shown the installation of a large wet-wood-burning furnace for a 1100-hp. boiler. This installation was made in California for burning water-soaked redwood refuse, which is about the poorest fuel imaginable, the moisture content being very high. In this furnace it will be noted that there are six chutes for supplying wood refuse to the furnace, which automatically form six conical piles of fuel on the horizontal grates. Air is admitted to the combustion chamber through the uncovered grates beneath the fuel and through a series of $4^1/_2$-in. by $4^1/_2$-in. openings on 9-in. centers in the side walls, center wall, and front wall of the furnace. The air ports are all arranged horizontally to admit air near the base of the cones of fuel. The air enters through an air intake in the back of the bridgewall, and through inlets at the top of the furnace in the front wall and in the forward portion of the side walls. The furnace is laid up with hollow-wall-tile construction providing free passage of air through ducts in all portions of the side walls, and thus the air admitted from the outside is first preheated by the air-cooled side-wall refractories before being admitted into the combustion chamber through the series of horizontal openings. Air is also admitted through ports in the bridgewall to the large combustion chamber beneath the boiler. The suspended arch is arranged to reflect the radiant heat back to the fuel cones and assist in the drying and volatilization processes.

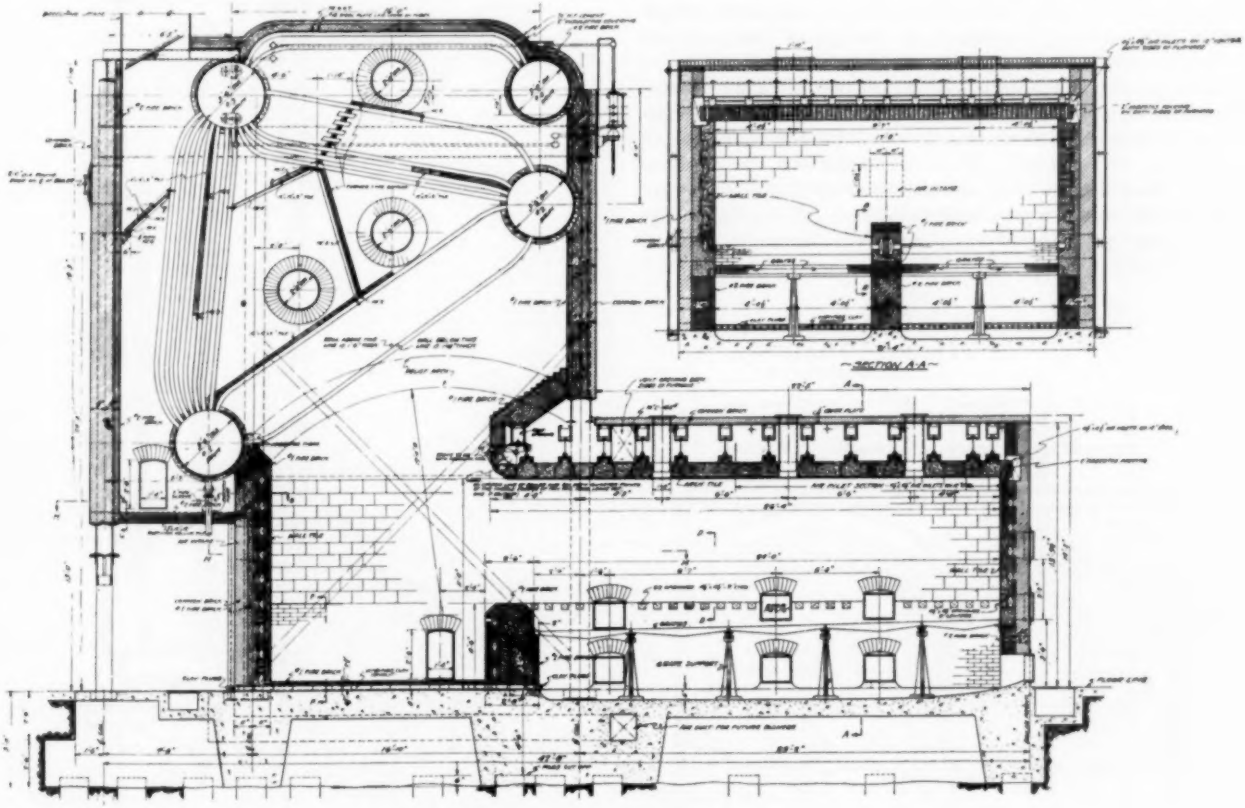


FIG. 5 FURNACE FOR WET REDWOOD REFUSE
(1100-hp. Badenhausen boiler, Hammond Lumber Co., Samoa, Calif.)

It is stated that with this design of furnace it has been possible to obtain outputs of 200 per cent of boiler rating, which is undoubtedly accounted for by the liberal provision in the combustion chamber for the utilization of the radiant heat in drying

the wood, promoting the volatilization of the combustible constituents of the fuel, and by the access of preheated air to all sides of the fuel cones.

In the lumber operations of Oregon and Washington there are some very interesting and well-designed boiler furnaces for burning green or wet wood refuse. These installations include boilers of over 2500 rated hp. in capacity, and the details of the furnace construction are of decided engineering interest.

Designs for Air-Dried and Kiln-Dried Wood

In industries where the maximum use of refuse wood for fuel is required and is commercially economical, the boiler settings should receive very careful engineering consideration. Many of the earlier types of boilers in such plants—and these are still used in most of the present-day furniture operations—employ vertical water-tube boilers with Dutch ovens, receiving the refuse directly from cyclone collectors. An example of this is shown in Fig. 6.

This type of installation is still in common use among furniture manufacturers and is representative of the earlier design for the burning of refuse wood at automobile-body plants dating back about 10 years ago. In such installations the sawdust, shavings, and hogged wood are delivered through the furnace front through chutes from a collector above. The collector also delivers large quantities of excess air to the furnace, and therefore the efficiency of such an installation is considerably reduced.

Certain plants have been arranged to burn dry hogged wood, sawdust, and shavings in furnaces which are also burning coal on underfeed stokers. The wood enters the furnace through chutes and is deposited on top of the coal fire and burns in an atmosphere of carbon dioxide and carbon monoxide without sufficient air. The volatile combustible matter is driven off, and these gases may or may not burn when they mix with the excess air admitted through the wood chutes above the fuel bed. This method of operation, however, usually proves to be a smoke nuisance. A certain amount of coal firing by the stokers is required to protect the stokers from the heat of the burning wood.

The bent-tube Stirling type of boiler with extended front walls is now extensively used in modern industrial plants, and

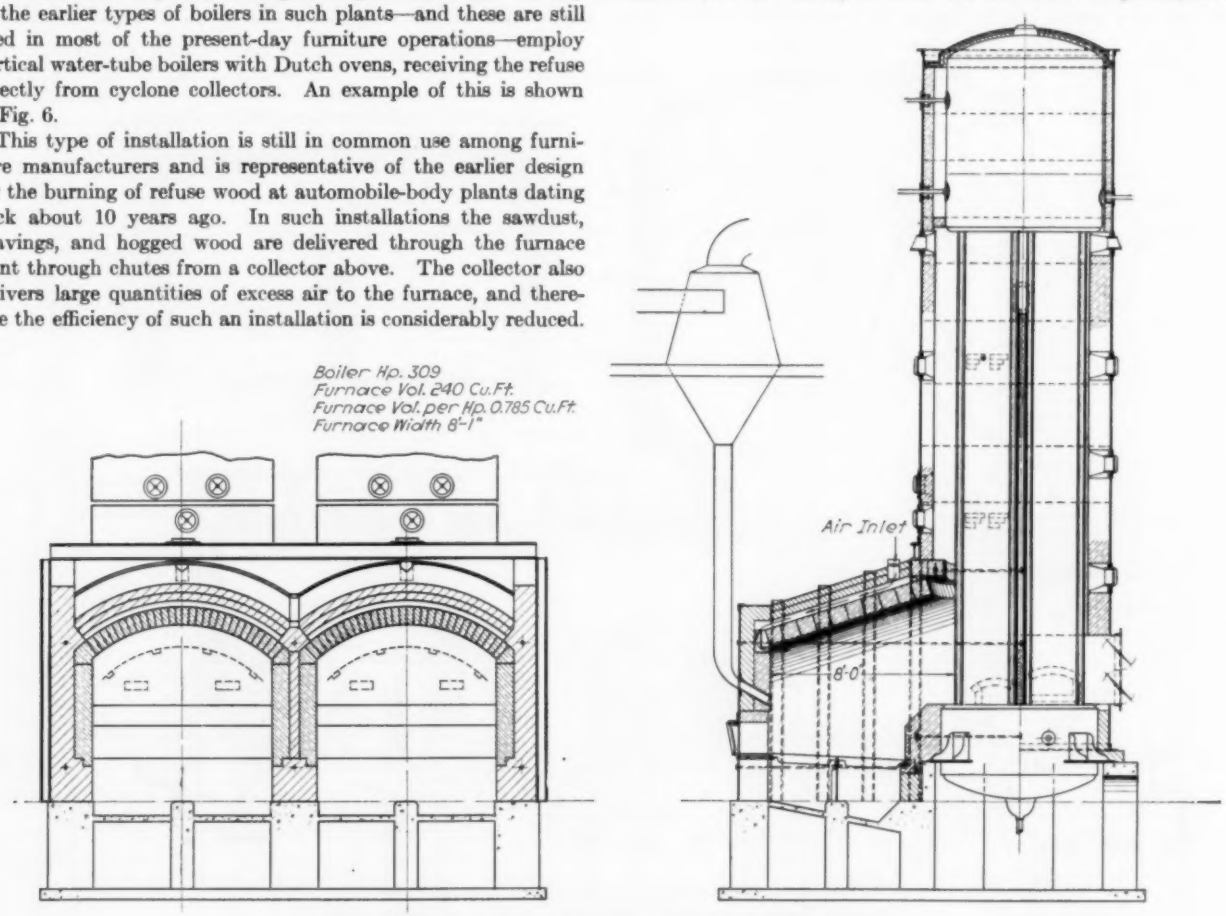


Fig. 6 Dry-Refuse Dutch Oven and 303-Hp. Wickes Boiler
(Original installation of Fisher Body Corporation, Detroit, Mich.)

The combustion space of this furnace is about one-tenth of the combustion space now considered most efficient for the burning of air-dried and kiln-dried refuse wood in suspension. At practically all rates of combustion this boiler and furnace setting would produce dense, black smoke, due primarily to the incomplete combustion of the high-volatile content of the wood and to the flame impingement on the boiler tubes, which produces smoke and soot. Also, certain kinds of refuse wood such as the pines have a large percentage of resinous volatile content which, without complete combustion, is sure to produce dense smoke. An additional air inlet to the furnace is also provided through the air inlet at the upper end of the arch, which serves also to cool the inner sprung arch and at the same time mix with the products of combustion in the upper portions of the furnace.

this type of setting lends itself to the auxiliary firing of the boiler with pulverized fuel as it has been found that the combustion space for burning either wood in suspension or pulverized coal is approximately the same.

Another type of boiler setting which is also largely used is the horizontal straight-tube type of boiler set over the top of the furnace. This arrangement has given very satisfactory operation. However, the Stirling type and the horizontal straight-tube boiler have their respective advantages, depending on the arrangement in the boiler plant and the floor-space and headroom requirements.

St. Louis Wood-Burning Boilers

Immediately after the World War, when one of the body-

manufacturing plants of the General Motors Corporation was being designed for St. Louis, a study of existing plants was made and the boiler setting shown in Fig. 7 was finally decided upon. In this setting it will be noted that the wood is fed through the arches of a double Dutch oven located in the basement below the main firing aisle of the boilers. This plant contained three stoker-fired coal-burning boilers set at the same elevation as shown in Fig. 7, with underfeed stokers operated from the main floor, and four wood-burning boilers set in batteries of two each with the Dutch-oven wood-burning furnaces in the basement in the relative position occupied by the ash bunker

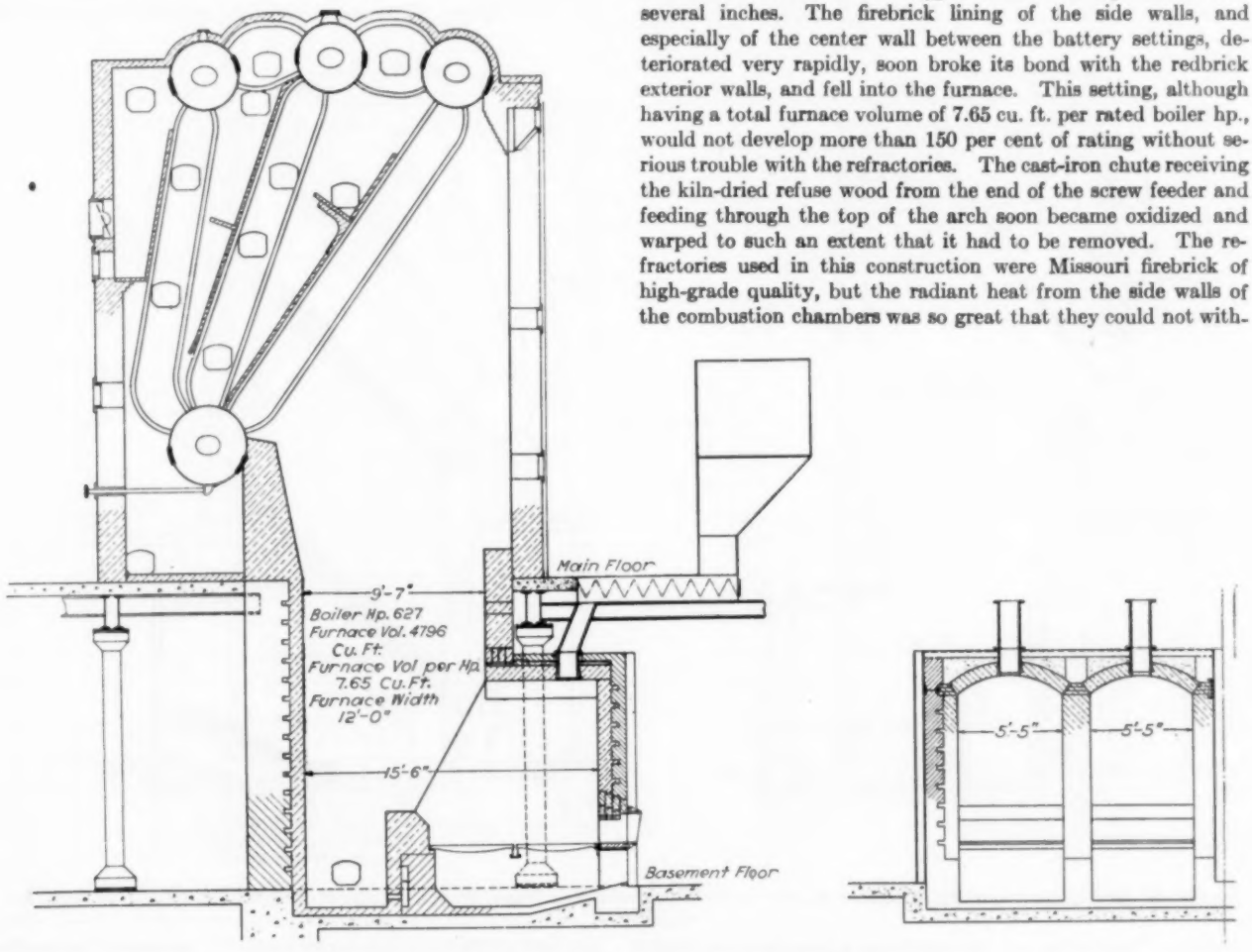


FIG. 7 ORIGINAL SETTING FOR BURNING DRY WOOD REFUSE
(627-hp. Stirling boiler, Chevrolet Motor Co., St. Louis, Mo.)

and the forced-draft duct of the stoker-fired boilers. The Dutch-oven combustion chamber consisted of two compartments separated by a center wall. Above the sprung arches of the two small combustion chambers, a sprung arch supported the curtain wall, sealing the boiler setting inside of the main front wall above the main floor line. This wood-burning setting was considered to be the most up-to-date of its kind at the time. However, at the time this installation was made the effect of reflected radiant heat on refractories and the most advantageous exposure of boiler tubes to the source of heat in the furnace were not fully understood. In this setting, it will be noted that provision is made for removing ashes from beneath the grates and that a cinder chamber at the rear of the bridgewalls is provided to which air is admitted through the bridgewalls. It is of interest to note that no ashes were ever accumulated in the ash-pit, as the draft through the grates would not permit the light ash to fall through. Also very little cinder collected in the back of the boiler setting. Ash removal in wood-burning settings is not a problem, for there is practically no ash.

After these furnaces had been in operation for a time the brickwork in the combustion chambers was found to suffer from overheating, and the sprung arches over the top as well as the curtain wall above the arches began to sag. The center wall between the combustion chambers melted to a considerable degree, and the entire center wall sagged so that the top was lowered several inches. The firebrick lining of the side walls, and especially of the center wall between the battery settings, deteriorated very rapidly, soon broke its bond with the redbrick exterior walls, and fell into the furnace. This setting, although having a total furnace volume of 7.65 cu. ft. per rated boiler hp., would not develop more than 150 per cent of rating without serious trouble with the refractories. The cast-iron chute receiving the kiln-dried refuse wood from the end of the screw feeder and feeding through the top of the arch soon became oxidized and warped to such an extent that it had to be removed. The refractories used in this construction were Missouri firebrick of high-grade quality, but the radiant heat from the side walls of the combustion chambers was so great that they could not withstand the temperature. Another point of interest in the design of this setting is that the sprung arches are covered with concrete slabs for insulation, thus retarding the radiation of heat and causing the arches to fail all the more rapidly.

The correction of these difficulties was made as shown at the left in Fig. 8. It will be noted in this sketch that the center wall of the Dutch oven was removed and the two sprung arches of the combustion chamber were replaced with one suspended arch. No attempt was made to admit air through the bridgewall. The curtain wall was reconstructed on supporting I-beams and a small, thin secondary curtain wall was constructed inside the other and supported on the nose of the suspended arch. The firebrick lining of the rear wall and side wall was increased in

thickness from 9 in. to 13½ in. This setting is much more successful in operation and in refractory maintenance than the preceding installation. It was still found, however, that 150 per cent of rated capacity was the maximum that could be obtained, even with this setting. The side-wall linings constructed of first-quality Missouri firebrick would erode at the rate of 1 in. to 1¼ in. per month, and would require relining every 90 days.

The next step in the development of this installation occurred when it became desirable to increase the capacity of the station for the generation of steam with coal as a fuel when there was insufficient supply of wood. At this time it was decided to install water-cooling protection for the side walls and the upper portion of the bridgewall, and introduce a powdered-coal burner above the main floor as shown at the right in Fig. 8. This construction was carried out in three of the four wood-burning boilers. The protection of the refractory side walls was taken care of by the installation of inclined fin-tube water-cooled construction with vertical headers beneath the mud drum and above the main floor as indicated. There are four 4-in. water-supply connections from the mud drum to each lower vertical header and four 4-in. connections from the upper horizontal extension of the upper vertical header into the front steam drum. The upper part of the bridgewall is protected by fin-tube construction from a water header at the rear and extending to the upper steam drum in front of the main bank of boiler tubes. A burner for pulverized coal was placed in the front wall of the setting. The other alteration consisted in removing the flat grates and installing a Drake patented step grate with refractory facings as shown. Air under fan pressure passes through small nozzles in the steps and horizontally through slots under damper control in the risers of each step. Thus the larger chips (hogged wood) which do not burn in suspension fall to the step grate and are burned by the impinging air. The wood-burning alterations were undoubtedly an improvement over the old. It is interesting to note that it has been possible by the addition of the pulverized-coal-burning equipment to operate the boiler at higher continuous ratings than before. In normal operation about 150 per cent of rating is obtained with wood; pulverized-coal fuel is then added at such a rate that the steam output is increased to 250 per cent of rating, and this seems to be about the maximum continuous steam generation that can be reasonably expected of this installation. These units are used for producing about 88 per cent of the total steam generated, the remainder being generated by the stoker-fired boilers.

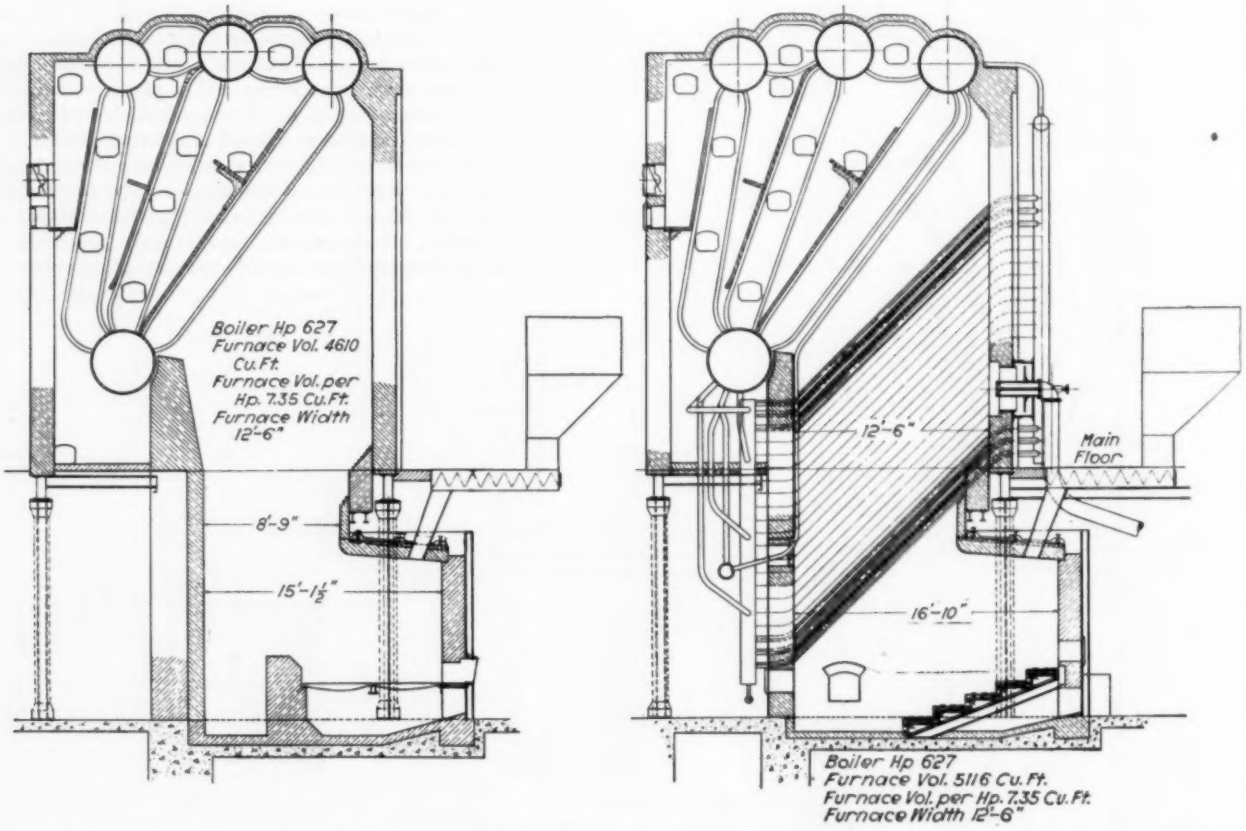


FIG. 8 ALTERATION AND FINAL ARRANGEMENT OF FURNACE AND SIDE-WALL WATER COOLING OF FURNACE SHOWN IN FIG. 7

DETROIT WOOD-BURNING BOILERS

Fig. 9 shows a cross-drum Heine boiler which was the first attempt of one of the large automobile-body manufacturers to employ the principle of air cooling to the side walls. On account of structural conditions in the building, it was impossible to extend the air cooling below the main floor. Consequently the lower half of the combustion chamber is restricted in width and is of solid-refractory construction. This setting is also provided with herringbone grates covering the entire bottom of the furnace. It will be noted in this layout that the air may be supplied either to the space below the grates or above them. This setting is a natural-draft setting in which the air is drawn through the horizontal passages in the side and rear walls, to a collecting duct at the rear wall, passing through ducts into the

air ports on either side of the setting as indicated. The operation of this unit was quite satisfactory, although there was still considerable maintenance required in the lower portion of the combustion chamber where solid walls were still used. It was found that the hogged wood which was not burned in suspension would accumulate on the grates to a depth of three or four feet except at the side-wall ports where the incoming air would cause rapid combustion, and the wood would burn as fast as it fell in these regions. It may also be observed that whenever the front doors of the setting were opened the wood would burn rapidly owing to the admission of air through the doors. In observing the burning of this fuel from air introduced beneath the grates, it was found that the combustion took place at the depth of three or four feet in the center because of insufficient contact with air, until the wood falls into the streams of air through the side ports and is consumed as fast as it falls. This type of setting has proved quite satisfactory in operation. The furnace linings are held to the outside wall by means of refractory blocks and hook irons as shown, with a 9-in. firebrick wall between each row of blocks. Air is admitted at the front of the boiler, passes around the side walls, and is collected at the rear with a fan, which is also provided with a bypass for natural-draft operation when desired, and enters through the side ports near the furnace bottom.

The life of the refractories in the side wall of this setting averages approximately 18 months, during which time the surface

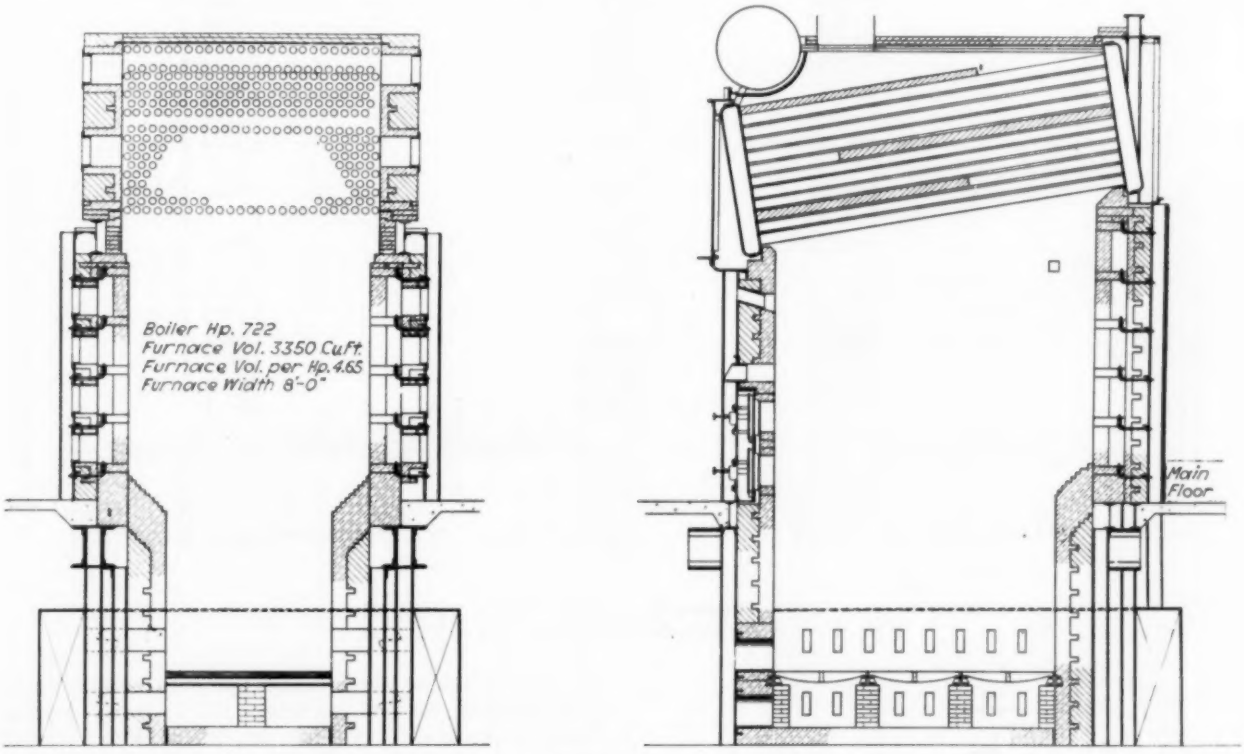


FIG. 9 FIRST DESIGN OF AIR-COOLED FURNACE FOR 722-HP. HEINE BOILER
(Fisher Body Company, Detroit, Mich.)

surface of the pile in small craters of a miniature volcanic nature. From observations taken here it was noted that dry hogged-wood refuse burns from the surface of a pile of fuel and not throughout the entire mass, as would be expected in the case of burning slabs, edgings, or pieces of considerable size.

As the next step in the development of wood-burning-boiler settings for this body manufacturer it was decided to eliminate the grates altogether and burn the wood from the floor of the furnace. The longitudinal cross-section of Fig. 10 shows the method of collecting the air from the horizontal ducts with a fan, which also forces the air into the furnace through the ducts and ports shown in the transverse cross-section. The central duct was provided to supply air to a pulverized-fuel burner for auxiliary firing in case it should be desired to add this feature to the unit at some future time. There is also shown a small port connecting with the outside of the top of the lateral ports which permits the use of steam or air lances for the purpose of stirring up the wood which accumulates in the center of this setting. Here again the hogged wood will pile up to a maximum erodes back from the original surface to the extent of approximately $4^1/_2$ in.

OTHER RECENT DESIGNS

The setting shown in Fig. 11 was a development designed after the one shown in Fig. 10 had been placed in operation. This furnace was designed to burn all the wood as it fell to the bottom of the furnace by introducing the air for combustion through the side walls of the boiler setting through comparatively small ports and at high velocity. The illustration shows the air being taken from the side walls, where it is admitted at both the top and the bottom of vertical ducts in the side walls, to a central collecting duct and after passing through the fan is delivered either to the wood-burning ports or through a riser to the pulverized-coal burner in the front of the boiler setting for auxiliary firing. The wood is admitted through four supply chutes in the upper portion of the front wall about 16 ft. above the furnace floor.

A test was run on this setting to determine the amount of air

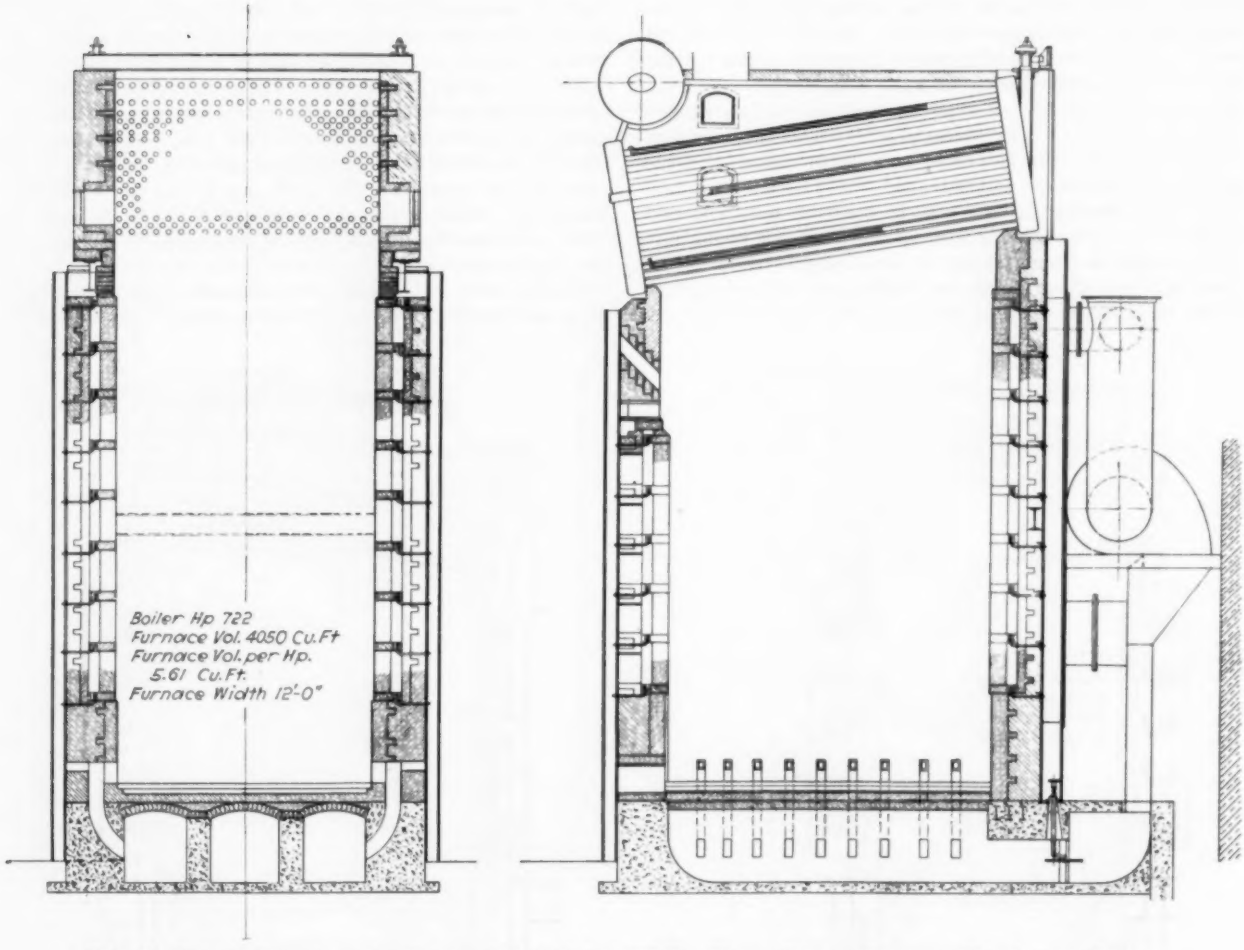


FIG. 10 SECOND DESIGN OF AIR-COOLED FURNACE FOR 722-HP. HEINE BOILER BURNING KILN-DRIED REFUSE
(Fisher Body Company, Detroit, Mich.)

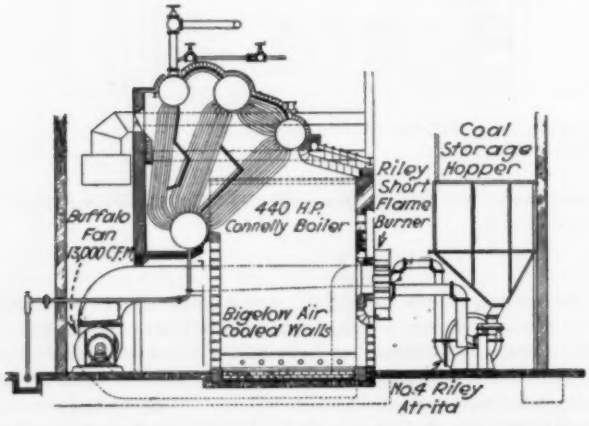


FIG. 11 DRY-REFUSE FURNACE WITH HIGH-VELOCITY AIR PORTS
(440-hp. Connelly boiler, General Motors of Canada, Ltd., Oshawa, Ont.)

required through the lower ports. This test was run at an average load of 836 hp. on the 440-hp. boiler, which is 190 per cent of rating, and it was found that the air required through the lower ports for combustion with CO_2 at 17 per cent, which corresponds with 18 per cent excess air, was 32 per cent of the total air required for combustion. The remainder of the oxygen required for combustion was contained inherently in the wood itself, and was also admitted through the wood chutes from the stoker supplying wood to the furnace and by leakage through the side walls and arch. The ports are designed with a tapering nozzle to provide a velocity of air under this condition of operation of about 4500 ft. per min. The shavings and sawdust burn in suspension before falling to the floor and before coming in contact with the front bank of boiler tubes. In this setting the hogged wood which dropped to the bottom was consumed as rapidly as it fell, and there was no accumulation whatever of wood or cinders on the furnace floor.

The design for the furnace shown in Fig. 12 provides for the collection of the air at the top of the vertical air-ducts in the walls of a boiler setting; this is the natural course for heated air to take, due to the chimney effect or draft produced by the heated air. The air is collected from the top of each of the vertical ducts by a horizontal collecting duct connecting to the suction of a fan. The delivery of the air from the fan is through the steel ducts at the side of the boiler setting at the bottom, and through small tapered ports as shown for delivering the air impinging on the floor of the furnace at high velocity. The operation of this boiler has been entirely satisfactory from the start. No wood accumulates on the bottom of the setting. The furnace is also arranged for auxiliary firing with pulverized coal, the burner being placed in the front wall of the boiler setting. This furnace, however, is rather narrow, and the reflected heat from one side wall to the other may result in overheating of the re-

fractories, and flames from the coal burner may impinge on the side walls to an extent which would not be the case if the furnace were of wider design.

Fig. 13 illustrates an air-cooled wood-burning furnace in which the wood is introduced through four ports in the front wall of the furnace setting and air is admitted through tapered side ports inclined toward the bottom after it has passed through cooling ducts beneath the furnace floor and through the vertical ducts of the side-wall cooling system. This unit has recently been reconstructed as shown with a furnace width of 14 ft. to replace two furnaces each 7 ft. 2 in. in width with an air-cooled center wall between them. On account of the narrow width of the old furnaces and the absence of cooling in the side walls, the refractory maintenance was excessive and it was necessary to of a fan, the fan delivering it to the two outside ducts under the bottom of the furnace, which in turn supply the air for combustion through the side ports at an average velocity of 1700 ft. per min. when operating at about 160 per cent of rating. In this installation the wood accumulates to a depth of about three feet at the bottom of the furnace as in cases previously mentioned (Figs. 9 and 10), but since no ash is deposited by the burning of gas in the auxiliary burners, this piling up of fuel is not objectionable. The operation of this setting has been most satisfactory. The control of the boiler is easily handled, and short-run tests made on the boiler indicate that it is a decidedly efficient installation.

The Stratton furnace which is designed for burning finely crushed coal in suspension has been tried with kiln-dried wood refuse as a fuel. This furnace has a hopper bottom with steep

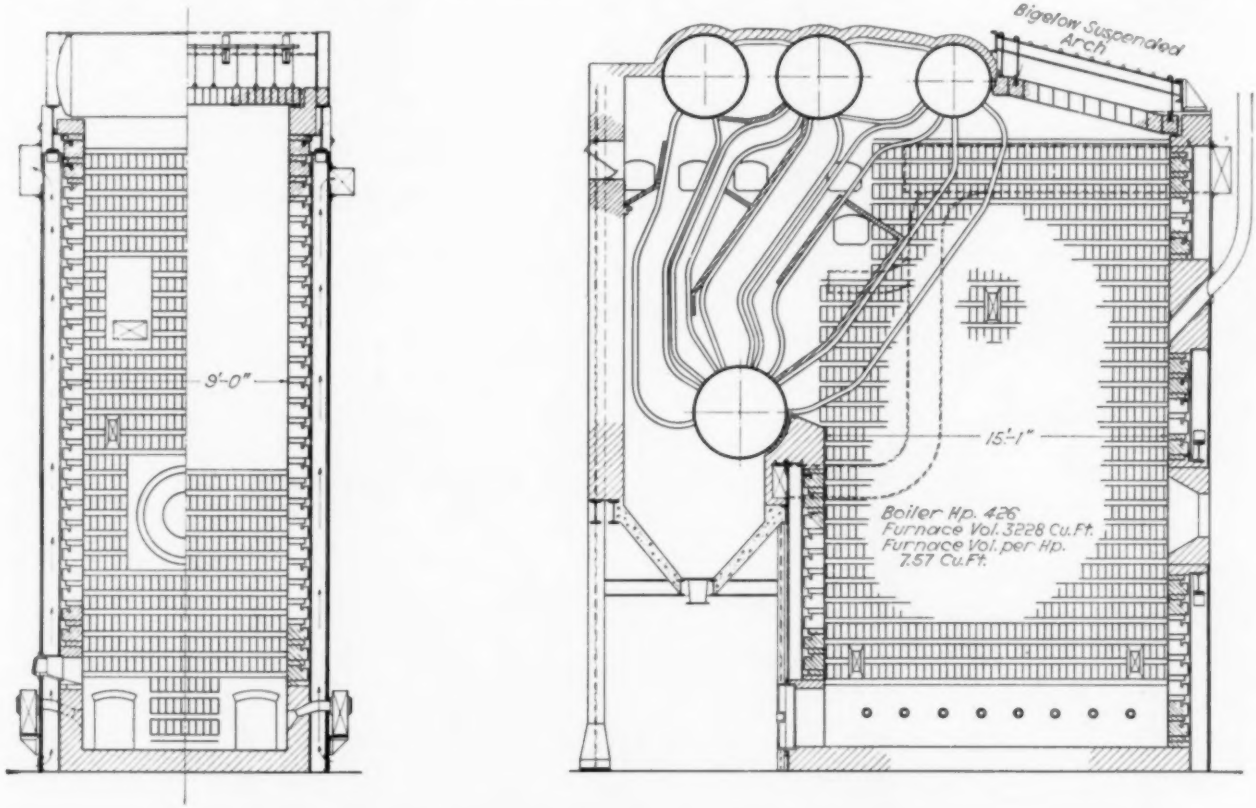


FIG. 12 DRY-REFUSE FURNACE AND 426-HP. STIRLING BOILER
(Rudolph Wurlitzer Company, North Tonawanda, N. Y.)

entirely rebuild the setting. Accordingly, the two 250-hp. cross-drum B. & W. boilers, formerly in battery setting, were combined into one by eliminating the center wall, moving the two banks of tubes together and installing a new cross-drum.

The installation shown in Fig. 14 is one of the most recent installations of an automobile-body plant. This boiler is fired with refuse wood from kiln-dried lumber used in the construction of automobile bodies, and natural gas is used for auxiliary firing, when wood is not available at night or during week ends. The boiler has 8870 sq. ft. of heating surface, and the water-cooling surface of the side walls is 540 sq. ft. Allowing 4 sq. ft. of surface per horsepower in the side-wall tubes, the combined rated capacity of boilers and walls is 1022 hp. In this setting the two side walls are water cooled with fin-tube construction; the front portion of the side walls and the rear walls are cooled with vertical air ducts through which air is drawn to the suction sides and a comparatively small grate so designed that the required amount of air will have sufficient velocity to keep the burning fuel in suspension until it is completely consumed. The trial, however, was not of sufficient duration, and the rates of steam generation were too low to prove much concerning the adaptability of the furnace for burning wood in continuous operation. However, the idea appears to be a good one, and if an installation were made with proper methods of controlling the wood entering the furnace it is quite probable that a satisfactory dry-wood-burning furnace would result.

An interesting design of dry-wood-refuse furnace was installed at Pekin, Arkansas, last year. Two 506-hp. boilers were set with the Drake patented wood-burning system consisting of air-cooled walls lined with interlocking refractory blocks 4 in. in thickness and refractory-faced step grates. The air was supplied by a fan, passing first through the side walls where it was pre-

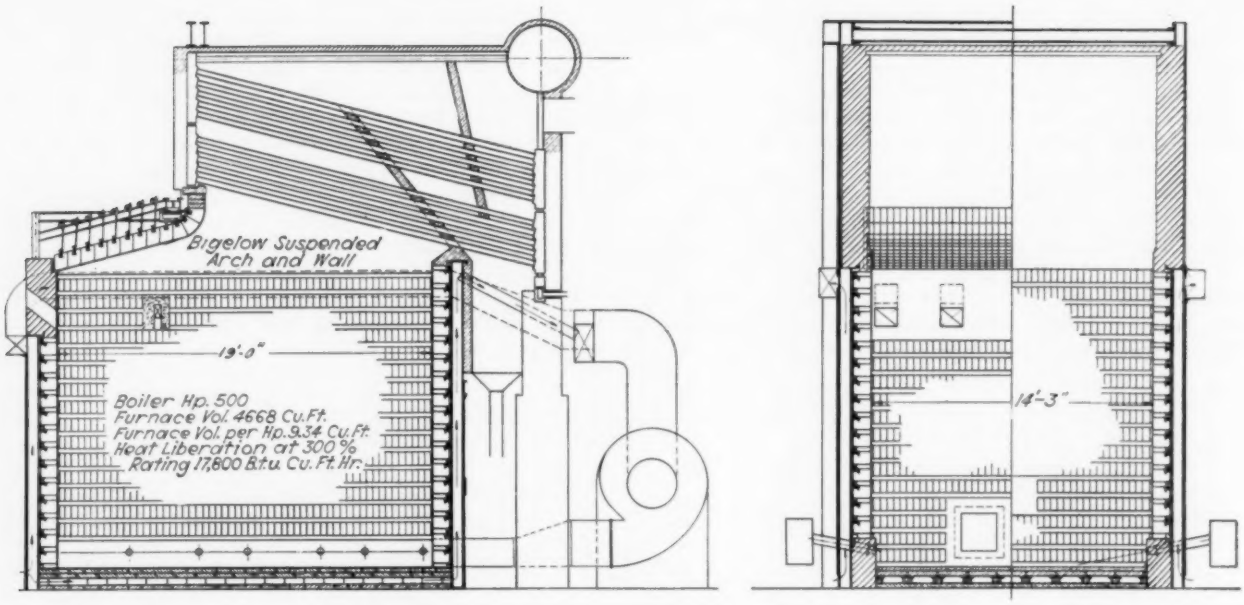


FIG. 13 DRY-REFUSE FURNACE FOR 500-HP. BABCOCK & WILCOX BOILER
(Frigidaire Corporation, Dayton, Ohio.)

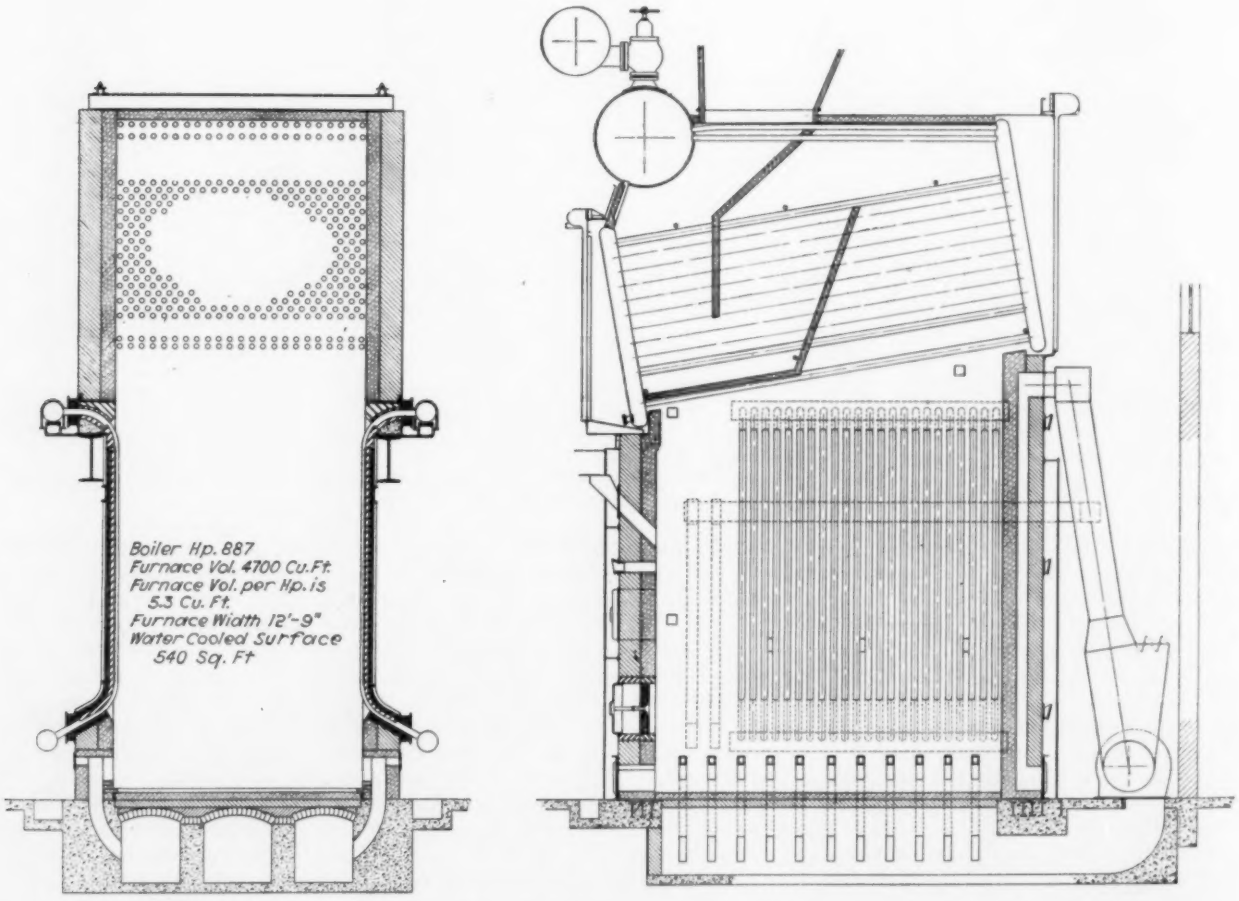


FIG. 14 RECENT DESIGN OF COMBINATION AIR-COOLED AND WATER-COOLED FURNACE FOR 887-HP. HEINE BOILER
(Fisher Body Corporation, Memphis, Tenn.)

heated, then entering the furnace through jets located in the front wall below the wood chutes and through the vertical and horizontal ports in the step grates. This setting has not been subjected to severe service, but the operating results thus far have been entirely satisfactory, and it presents features of unusual interest.

Conclusions

For burning green or wet wood refuse, considerations in the design of the furnace include the average moisture content of the fuel and the relative proportions of sawdust, shavings, hogged wood, and larger pieces.

The ignition arches should be designed to promote the maximum evaporation of moisture and volatilization of the wood. If possible, preheated air should be provided to further promote combustion in the furnace. Because of the low combustion temperatures of wet or green wood refuse, standard brands of firebrick may be used. The furnace volume should be liberally designed and be arranged with a primary combustion chamber for drying, volatilization, and ignition, and a secondary combustion chamber for further and complete burning of the volatile gases. In the case of the setting shown in Fig. 5, which is typical of recent practice, the primary ignition chamber has a volume of 3.3 cu. ft. per rated boiler hp. and the secondary combustion chamber 3.2 cu. ft. per hp., or a total combustion space of 6.5 cu. ft. per hp.

For burning dry wood refuse, the furnace should be arranged to satisfactorily meet the following conditions:

1 General arrangement of the setting as a whole with respect to the boiler to be used;
2 If supplementary firing with other fuel than wood is required, the furnace must also be designed to suit the auxiliary firing equipment decided upon;
3 The maximum rate of steam generation desired; and
4 The walls should be designed to insure minimum refractory maintenance for the rate of heat liberation per cubic foot of furnace volume decided upon.

The first two of these conditions are entirely dependent upon the requirements of any particular case, and upon decisions which do not affect the volumetric proportions of the furnace.

The maximum rate of steam generation, however, requires for any boiler a quite definite furnace volume for satisfactory smokeless combustion. To produce rated capacity of a boiler (i.e., 34.5 lb. evaporation per hour from and at 212 deg. fahr. per 10 sq. ft. of heating surface) would require a heat liberation in the furnace of 47,890 B.t.u. per hour per boiler hp. if we assume an efficiency of the boiler and furnace of 70 per cent. If it is desired to operate the boiler at higher rates of evaporation per 10 sq. ft. of heating surface, the heat liberation in the furnace will increase accordingly.

Table 2 is offered as representing good practice for rates of heat liberation for different wall constructions and the corresponding furnace volumes of varying rates of steam generation, assuming 70 per cent efficiency at all ratings.

The trend of increasing furnace volume per rated boiler horsepower for burning dry wood is indicated by some of the recent designs described in this discussion which average 7.4 cu. ft. of furnace volume per rated boiler hp. as compared with 0.8 cu. ft. per hp. in the Dutch-oven construction of the vertical water-tube boilers which were formerly so extensively used.

TABLE 2 CUBIC FEET OF FURNACE VOLUME SUGGESTED PER 10 SQ. FT. OF BOILER HEATING SURFACE

Type of furnace wall	Maximum rate of heat liberation, B.t.u. per cu. ft. per hr.	Maximum desired rate of steam generation: Normal rating, cu. ft.	200 per cent of rating, cu. ft.	300 per cent of rating, cu. ft.
Solid standard firebrick walls	11,000	4.35	8.70	.....
Special 60 per cent alumina firebrick walls	14,000	3.42	6.84	10.26
Air-cooled wall construction	17,000	2.82	5.64	8.46
Air-cooled walls with water-cooling protection	20,000	....	4.79	7.15

Although refractories are procurable containing up to 80 per cent of alumina in the form of mullite which will withstand extremely high temperatures and no doubt give good service in a furnace burning kiln-dried wood refuse, the cost of furnaces constructed with these brick would be greater than for air-cooled wall construction with standard 40 per cent alumina content refractories, and the additional cost is not economically justifiable.

The furnace construction which will require the least expenditure in maintenance is without doubt the air-cooled wall laid up with standard first-quality firebrick with a certain amount of water-cooling protection. The extent and distribution of the water tubes along the side walls must be carefully determined and not overdone to the extent of lowering the temperature in the combustion space so that smokeless combustion of the wood refuse becomes impossible.

Discussion

R. L. Beers.[2] For burning wet wood refuse, there are many plants in operation that use mechanical stokers. The underfeed stoker has been very successful for this application. It is customary to set the stoker in an extension furnace, coal is fed into the furnace with the underfeed stoker, and the wood is spouted into the furnace through an opening in the roof. With this arrangement, wood of high moisture content can be burned at high rates of combustion, due to the hot coal fire underneath and the reflected heat from the brickwork above.

Due to the rapidity at which wood is burned, if the supply is stopped, the wood in the furnace is quickly exhausted, and then it is important to have an auxiliary means for carrying the load. When this occurs, the load can be quickly picked up and carried entirely with a straight coal fire. In some plants, with wood under 50 per cent moisture content and ample supply, the load is carried on straight wood, and the stoker is held in reserve for emergency use or for periods when there is insufficient supply to carry the load.

The underfeed stoker, with most Eastern coals, does not depend upon reflected heat for ignition, and a coal fire can be carried with a deep pile of wood over it. With an overfeed type of stoker, wet wood cannot be so successfully handled, as the fire seems to be smothered.

There are many small plants that use wood for generating part of the steam. In most of these plants, kiln-dried wood is used. With this fuel, ignition arches are not necessary. In these small plants, reserve boiler capacity is not always available, and it is quite essential that auxiliary fuel-burning equipment be used. Stokers have been widely applied for burning coal in conjunction with sawdust, shavings, and kiln-dried wood.

The furnace setting with a stoker is practically the same as for burning straight coal. An extension furnace is usually not necessary. The wood is spouted into the furnace through the front wall, and most of the wood is burned in suspension. Care should be exercised, however, to get ample furnace volume, depending upon the load and the percentage of wood being burned, and provision should be made to admit a large volume of air over the coal fire. It is also important that the wood supply be controlled with the air supply and vary with the load. With dry wood, used in conjunction with underfeed stokers, straight wood can be burned over long periods, but it is advisable to

[2] Vice-President in Charge of Engineering, Detroit Stoker Company, Detroit, Mich. Assoc-Mem. A.S.M.E.

fill these retorts with ashes and to admit some air through the tuyères to protect the ironwork.

With dry wood, high furnace temperatures, with resultant short life of brickwork, are frequently experienced, particularly with high boiler settings, but this is caused to a great extent by the large amount of brickwork in proportion to the heating surface of the boiler exposed to the radiant heat of the furnace. With wood free from foreign material, and providing furnace water-tube area equal or greater in proportion to what is commonly used with straight coal-burning furnaces, the brickwork maintenance should not be excessive.

L. S. Cockburn.[3] For the past ten years, the writer has been directly connected with the problem of burning kiln-dried wood waste.

In the early days it was looked upon as merely a method of disposal of this waste product with the generation of such steam as could be had from it. In most cases there was more than enough fuel available to produce all the steam required, and efficient operation meant nothing. The question of smoke emission had not arisen, and furnace maintenance was more or less taken for granted.

New processes came into use requiring steam, and more efficient manufacturing methods reduced proportionately the quantity of wood waste available as fuel. It became necessary to purchase fuel for steam generation.

Then, the efficiency of operation meant something, and steps were taken to find out just what was going on, and development took place leading up to what we now have, going through the stages as outlined by Mr. Gladden.

As Mr. Gladden stated, furnace volumes ten years ago were small. The 400-hp. boilers then in use had only 288 cu. ft. of furnace volume. Our latest installation of about 1000 hp. has 4700 cu. ft. of furnace volume. In other words, the rated horsepower of the unit has only increased two and one-half times, while the furnace volume has increased sixteen times.

The writer wishes to point out that the proper furnace design is not the end of the problem. The method of feeding the fuel is important. The feed must be uniform, without belching, and it must be directed to feed into the furnace at the proper points and at the proper angle.

There is one installation we made several years ago that has proved very satisfactory. In this installation, we are using a chain-grate stoker designed for wood-burning. When this installation was first made, considerable trouble was encountered. We attempted to carry the fuel so as to burn out at the end of the grates. This resulted in the grates becoming red hot, warping, and breaking. After some time, it was found that the fire should be burned out a little over half the length of the grates and the air allowed to pass through the grates at the rear end, cooling the grates.

The maintenance on both furnace and stoker has been very reasonable, but the main objection has been that, when wood fuel is not available, the boiler is down, as coal cannot be burned on that type of chain grate satisfactorily.

Mr. Gladden mentioned our latest installation at Memphis, Tenn., and shows a section of this boiler in Fig. 14. We have some figures on the operation of this boiler that might be of interest.

In the first place, this boiler is equipped with a Pratt & Daniels Thermix stack, the fan being driven by a steam turbine, as is the forced-draft fan, and is provided with automatic control. This combination has given us very flexible operation and when set for any desired operating condition, this condition is maintained within a very close range.

When the unit was first placed in operation last November, tests were made to determine the most efficient operating conditions for different ratings. While we were unable to measure the quantity of fuel burned, we were, however, able to make some comparisons as to quantity from the speed of the screws on the fuel feeders.

[3] Assistant Works Engineer, Fisher Body Corporation, Division of General Motors Corporation, Detroit, Mich. Mem. A.S.M.E.

TABLE 3 NATURAL GAS; EVAPORATION TEST

Boiler No. 1, Heine H.-W. T. Located at Memphis, Tenn. Date of test, Nov. 20, 1929. Duration, 3 hr. Conducted by L. S. Cockburn and L. F. Simmons.
Heating surface, sq. ft.: Boiler, 8870; water walls, 540.
Rated horsepower: Boiler proper, 887; water walls, 140; total, 1027.

(1)	Steam pressure, lb. per sq. in	160
(2)	Percentage of moisture in steam	Dry—sat.
(3)	Factor of correction for quality of steam	0
(4)	Feedwater temperature, deg. fahr	200
(5)	Factor of evaporation	1.059
(6)	Equivalent evaporation per hour, from and at 212°F.	69,257.22
(7)	Equivalent evaporation per hour, from and at 212°F. per sq. ft. total heating surface	7.36
(8)	Percentage of rated capacity developed	195.4
(9)	Cu. ft. gas per hr. (standard cu. ft. at 60 deg. and 4 oz. absolute pressure)	84,667.7
(10)	Equivalent evaporation, from and at 212°F., per cu. ft. gas	0.82
(11)	Heating value of gas per cu. ft	960
(12)	Efficiency, per cent	84.3[1]
(13)	Barometer, inches of mercury, sea level	30.45
(14)	Relative humidity of air for combustion	74
(15)	Temperature of air for combustion, deg. fahr	79
(16)	Furnace temperature, deg. fahr	2000
(17)	Temperature of gases leaving boiler, deg. fahr	470
(18)	Furnace draft	0.22
(19)	First pass	0.01
(20)	Second pass	0.10
(21)	Third pass	0.62
(22)	Uptake draft	0.6
(23)	Analysis of natural gas, per cent by volume: Methane, 95.0; ethane, 1.0; inert, 4.0	
(24)	Analysis of flue gas, per cent by volume: CO_2, 10.0; O_2, 3.53; CO, 0.1.	
(25)	B.t.u. release per cu. ft. furnace volume per hour	17,300

[1] In connection with the tests on natural gas, it is pointed out that the efficiency as shown does not check with that found in calculating a heat balance using the fuel analysis given. The heat balance shows an efficiency of 79 per cent. The discrepancy probably is due to the gas having a higher heat content than the figure used, which would reduce the test results, and also to the gas having a lower methane content and a higher ethane content, which would reduce the hydrogen content, and therefore the losses due to combustion of hydrogen, which would increase the efficiency as found from the heat balance, bringing both results in closer agreement. The heat value and analysis were taken from data furnished by the utility furnishing the gas.

TABLE 4 WOOD BURNING; EVAPORATION TEST

Boiler No. 1, Heine cross-drum, inclined baffles. Located at Memphis, Tenn. Date of test, Nov. 21, 1929. Duration, 3 hr. Conducted by L. S. Cockburn and L. F. Simmons.
Heating surface, sq. ft.: Boiler, 8870; water walls, 540; total, 9410.
Rated horsepower (4 sq. ft. water wall per hp.), 1027.
Furnace volume, 4700 cu. ft.
Fuel, wood waste

(1)	Steam pressure, lb. per sq. in	154.6
(2)	Moisture in steam, per cent	Dry—sat.
(3)	Factor of correction for quality of steam	0
(4)	Feedwater temperature, deg. fahr	200
(5)	Factor of evaporation: $H - Q/970.4 = 1195.34 - 168/970.4$	1.057
(6)	Equivalent evaporation per hr., from and at 212°F.	56,021
(7)	Equivalent evaporation per hr., from and at 212°F., per sq. ft. total heating surface	5.95
(8)	Percentage of rated capacity developed	157
(9)	Percentage of moisture in fuel	6
(10)	Dry fuel, per hr	8720
(11)	Equivalent evaporation from and at 212°F., per lb. dry fuel	6.42
(12)	Heating value per lb. dry fuel	8000
(13)	Efficiency, per cent	76.4
(14)	Barometer, inches of mercury, sea level	30.53
(15)	Relative humidity of air for combustion	67
(16)	Furnace temperature, °F	2200
(17)	Temperature of gases leaving boiler, °F	516
(18)	Temperature of air for combustion, forced air through walls, °F	122.6
(18a)	Temperature of air for combustion, suction from boiler room, °F	80.4
(19)	Furnace draft	0.18
(20)	First pass	0.30
(21)	Second pass	1.00
(22)	Third pass	1.54
(23)	Uptake draft	1.60
(24)	Refuse, per cent of dry fuel	0
(25)	Analysis of fuel, per cent by volume: Carbon, 49.2; hydrogen, 5.6; oxygen, 38.2; ash, 1; moisture, 6.	
(26)	Analysis of flue gases, per cent by volume: CO_2, 13.75; O_2, 5.7; CO, 0.4.	

Runs were also made carrying high percentages of CO_2 in the flue gas to determine the effect on the refractories. We found that up to 16 per cent CO_2 and over 80,000 lb. steam per hour generated for a period of several hours, there was no effect on the brick. In our air-cooled jobs, with CO_2 as high as this and at comparable ratings, the refractories would show signs of distress in less than two hours.

The test results given in Tables 3 and 4 were obtained during the trial period in November.

As there was no method of weighing the wood fuel, it was necessary to compute the heat balance, assuming the radiation and unaccounted-for loss, but it is believed that the figures are reasonably correct.

The gas consumption for the natural-gas run was taken from the meter furnished by the gas company from which they bill for gas consumed, so that it can safely be assumed that the quantity shown is not low.

The steam flow was taken from a Bailey meter, which was put in service the day before and also was re-checked afterward by the service man.

One further item that may be of interest is a comparison of the costs of an air-cooled installation as against the combination of water walls and air-cooling. For this purpose, there is compared a unit rated at 1000 hp. with air-cooled walls against the Memphis unit. The costs omit all items that are common to both units and consider only the boiler, water walls, erection, and furnace construction.

COMPARISON OF BOILER AND FURNACE COSTS

	1000-hp. air-cooled	887-hp., 540-sq. ft. water wall
Boiler only	$12,724.00	.....
Boiler and water walls	.....	$24,616.00
Erection and brickwork	17,792.00	14,833.00
Totals	$30,516.00	$39,449.00
Maximum steaming rate, lb. per hr., without excessive maintenance	62,000	90,000
Cost per 1000 lb. of steam per hr	$ 49.76	$ 43.84

Up to May 1, 3912 hours have elapsed since the boiler was first put on the line. Of that item, it has been in service 3181 hours, and of that period it used wood waste as fuel for 2797 hours and natural gas for 384 hours.

Steam generated from wood waste was 120,000,000 and from natural gas 16,063,860. Gas consumed was 20,334,000 cu. ft., having a heating value of 960 B.t.u. per ft., or an actual evaporation of 790 lb. steam per 1000 cu. ft. of gas.

The first shutdown was in March, for about one week, to replace the fan wheel in the induced draft fan, and again in April, to make some changes in the fuel feeders.

An inspection of the furnace showed no signs of any deterioration of the refractories and it was apparently in as good condition as when first started up.

One of our plants is having trouble, due to an unlooked-for condition. The wood waste at this plant contains a considerable amount of glue, which is made from soyo beans, mixed with sodium silicate and caustic soda. This mixture, together with the potash from the wood ash, has caused a rapid breaking down of the refractories, as well as a bad slag condition on the grates. The use of water-cooled grate bars has aided considerably, but we have not yet decided what we will do with the walls. This is a comparatively small installation, without air-cooled walls.

Incidentally, as an experiment, we are trying out the use of monolithic walls in several furnaces, and while this material has been in use too short a time to make any statements as to its life compared to brick, the present indications are very favorable.

H. S. Colby.[4] Mr. Gladden is to be congratulated on the completeness of his presentation of this very interesting subject. The burning of fuels of high moisture content presents problems in the design and construction of furnaces quite different from other solid fuels.

It has been the writer's good fortune to have been closely associated with the burning of solid fuels for some years and to have had an opportunity to study the development of furnace designs for the burning of a wide variety of fuels, including some of the high-moisture-content fuels, such as hog wood refuse, bagasse, and lignites of the Northwest.

In presenting his paper the author touched on the fact that he advocated the use of preheated air when burning high-moisture fuels, but expressed regret that he did not have data available on the effect of preheated air on this practice. The writer has been closely associated with the application of preheated air on several high-moisture fuels and will accordingly confine his discussion to the advantages that apparently have been derived from the use of preheated air in this field.

It is obvious that the use of preheated air for the combustion of wet fuel results in an increase in boiler efficiency as a result of reducing the final gas temperature. However, the effect of reducing the fuel consumption is not always of extreme importance in the burning of waste fuel unless it should so happen that the steam demand is in excess of that which can be generated with the waste fuel available. The savings that can be accomplished in the burning of high-moisture fuels with preheated air are considerable, but are in fact secondary to the very pronounced advantage of greatly increasing the steam-generated capacity of the boiler unit with the same furnace volume or within the limitations imposed by the same overall setting dimensions.

The boiler furnace for the burning of waste wood or other high-moisture fuels which most effectively utilizes the preheated air is that furnace design which provides for the distillation of the gases in one section of the combustion chamber and arranges for the subsequent burning of these gases in a secondary combustion chamber of liberal proportions and exposed to the radiant-heat absorption of the boiler tubes. Provision should also be made for the admission of part of the combustion air over the fuel bed and preferably at such a point that it is brought into intimate mixture with the stream of gases liberated from the fuel bed deficient in oxygen. The intimate mixing of this air and contact with the gases, preferably at a point shielded from exposure to the boiler tubes to insure a high temperature at the point of mixing, results in very prompt and complete combustion and in some cases with considerably less excess air than is customarily possible when supplying cold air to the furnaces.

A most satisfactory and representative design for the burning of high-moisture-content wood refuse is the installation at the plant of the Union Bag and Paper Power Company at Tacoma, Wash., from designs of the H. S. Ferguson Company, consulting engineers, shown in Fig. 15. These boilers are operated up to a maximum rating of 250 per cent, at which rating the temperature of the air leaving the preheater is in the vicinity of 400 deg. fahr. We are advised by the management of the plant that with cold air supplied to the furnaces it is possible to obtain only about three-quarters of the steaming capacity that can be obtained when using preheated air.

Another striking example of the advantages of preheated air in the burning of a high-moisture fuel is the installation of the United Sugar Companies at Los Mochis, Mexico. In this plant there are four 500-hp. and four 700-hp. boilers fitted with bagasse-burning furnaces of the type shown in Fig. 16, from a design prepared by the Honolulu Iron Works Company, New York City.

[4] President, Air Preheater Corporation, Wellsville, N. Y. Mem. A.S.M.E.

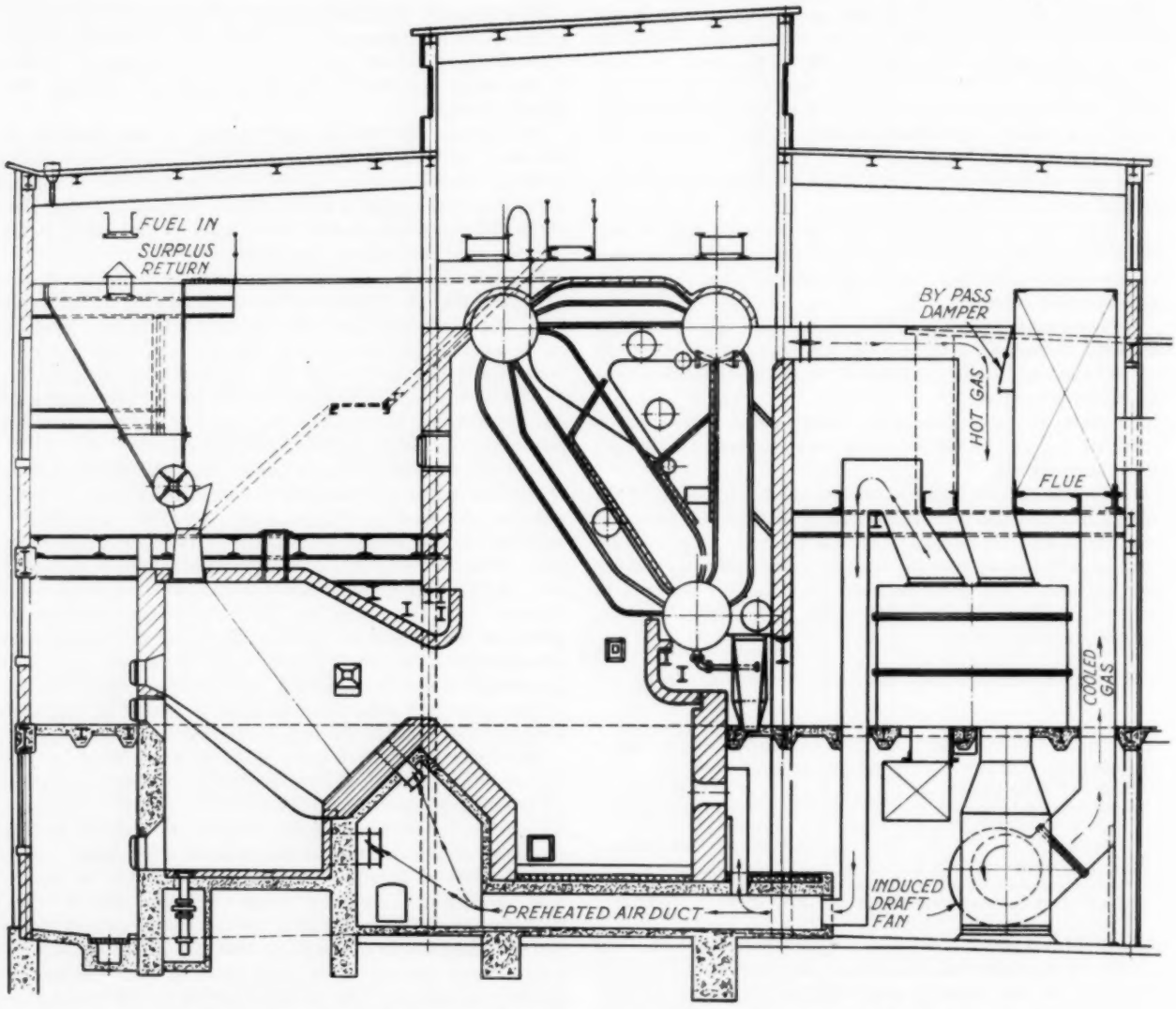


FIG. 15 INSTALLATION AT PLANT OF UNION BAG AND PAPER POWER COMPANY, TACOMA, WASH.

The fuel is bagasse having a moisture content of 40 to 50 per cent. In this plant steam is generated for the driving of the mill crushing the cane and the production of refined sugar. Before using preheated air it was necessary to supplement the bagasse available with fuel oil burned in the secondary combustion chamber of the furnace and in an amount averaging about 7 gal. per ton of cane crushed. The maximum rating that could be obtained from the boilers was in the vicinity of 85 per cent of boiler rating.

TABLE 5 COMPARISON OF PERFORMANCE WITH AND WITHOUT PREHEATED AIR, LUGNVIK SAW MILLS, LUGNVIK, SWEDEN

	With preheated air	Without preheated air
Moisture content of fuel, per cent	49.1	45.8
Fuel consumption per hour, lb	5511	5390
Fuel consumption per hour and sq. ft. grate surface, lb	93.5	91.5
Water evaporated per hour, lb	12,720	11,023
Water evaporated per hour and sq. ft. heating surface, lb	6.53	5.66
Average steam pressure, lb. per sq. in	263	263
Average temperature of steam, deg. fahr	648	637
Temperature of feedwater, deg. fahr	198	196
Temperature of flue gases entering air preheater, deg. fahr	552	.....
Exit temperature of flue gases, deg. fahr	303	.....
Temperature of air entering air preheater, deg. fahr	82	48
Temperature of air leaving air preheater, deg. fahr	456	.....
Temperature of flue gases leaving boiler, deg. fahr	568	539
CO_2 contents at boiler damper, per cent	14.75	14.62
Efficiency of boiler, per cent	73.0	60.4

Some two years ago one preheater was installed furnishing preheated air to four of these boiler units. It was found that the boilers operating on preheated air could be increased in steaming capacity from 85 to 135 per cent of rating. The preheating of the combustion air to a temperature of approximately 400 deg. fahr. resulted in a fuel saving which reduced the auxiliary fuel oil from 7 gal. to about $2^1/_2$ gal. per ton of cane crushed. The second preheater installed and placed in operation during the last grinding season, which now supplies the remaining boilers with preheated air, has resulted in a further economy, so that it is now possible to operate the entire boiler house at a rating of 135 per cent and without auxiliary oil fuel.

It is unfortunate that operating tests are not available on either of these installations so that a direct comparison of the performance with and without preheated air would be available.

However, Table 5 gives comparative performance data ob-

tained on an installation at the Lugnvik Saw Mills, at Lugnvik, Sweden. The furnace design (Fig. 17) parallels very closely the design used in this country, providing means for delivering air under pressure through the bridge wall and over the arch for the admission of secondary air into the gas stream passing up from the combustion chamber, which is located below the boiler room operating floor. The boilers are of the Babcock & Wilcox type, having a heating surface of 1949 sq. ft. and provided with a grate surface of 58.9 sq. ft. The fuel used is hog waste fuel having a moisture content as indicated in the table. It will be observed in this instance that the evaporation when using preheated air has been increased approximately 15 per cent, although the writer is not in a position to state whether this represents the maximum capacity that could be obtained.

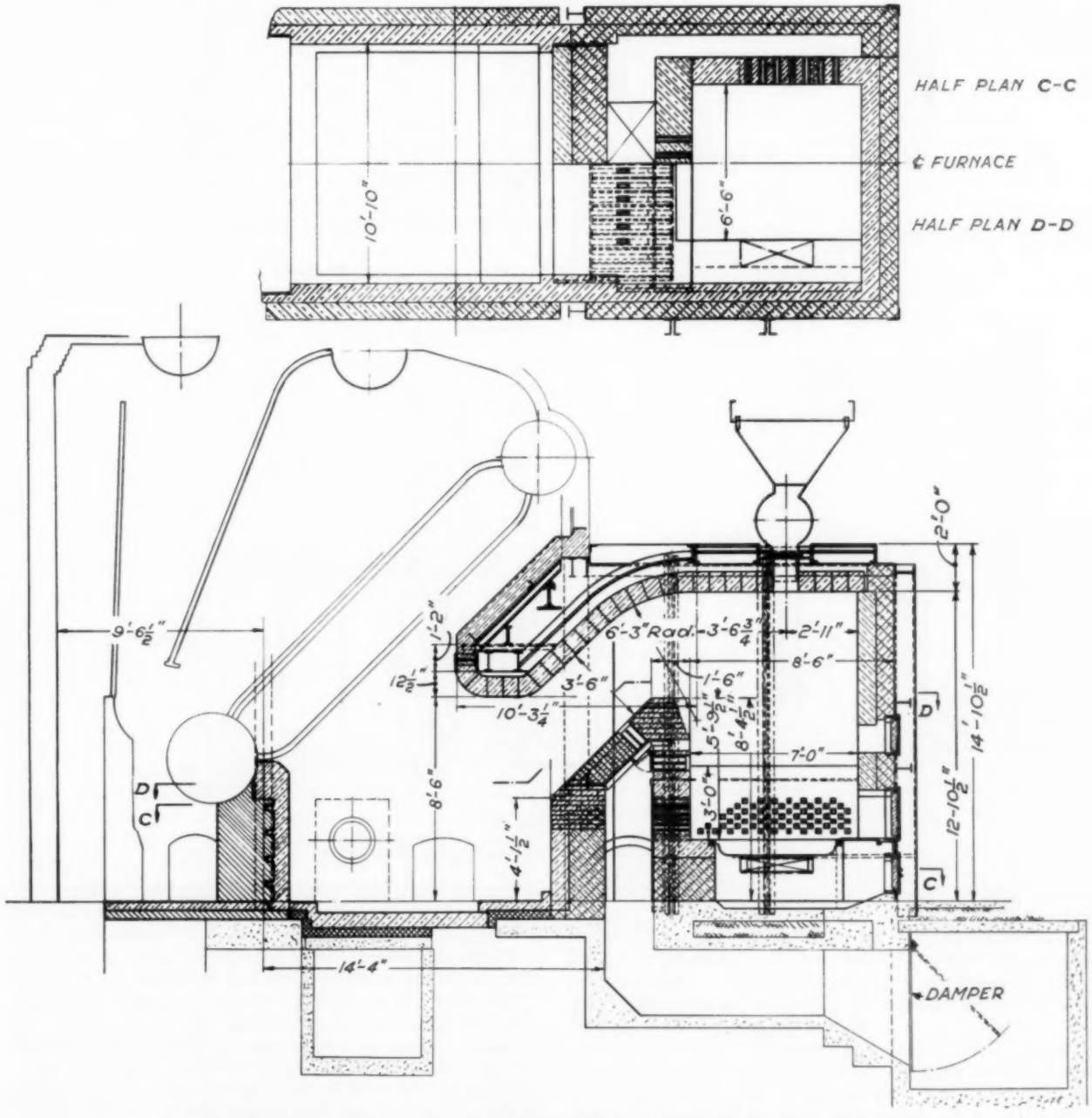


FIG. 16 BAGASSE-BURNING FURNACE TYPE, UNITED SUGAR COMPANIES, LOS MOCHIS, MEXICO

S. N. HEYMAN.[5] There are a few operating characteristics in the burning of dry wood refuse, to which the writer would like to call your attention.

In burning dry wood refuse in a furnace with air-cooled refractory walls or water-cooled walls, it is advisable to install a standby unit to burn oil, pulverized fuel, or natural gas. We find a unit pulverizer installation with each boiler a practical

[5] Fisher Body Corporation, Detroit, Mich.

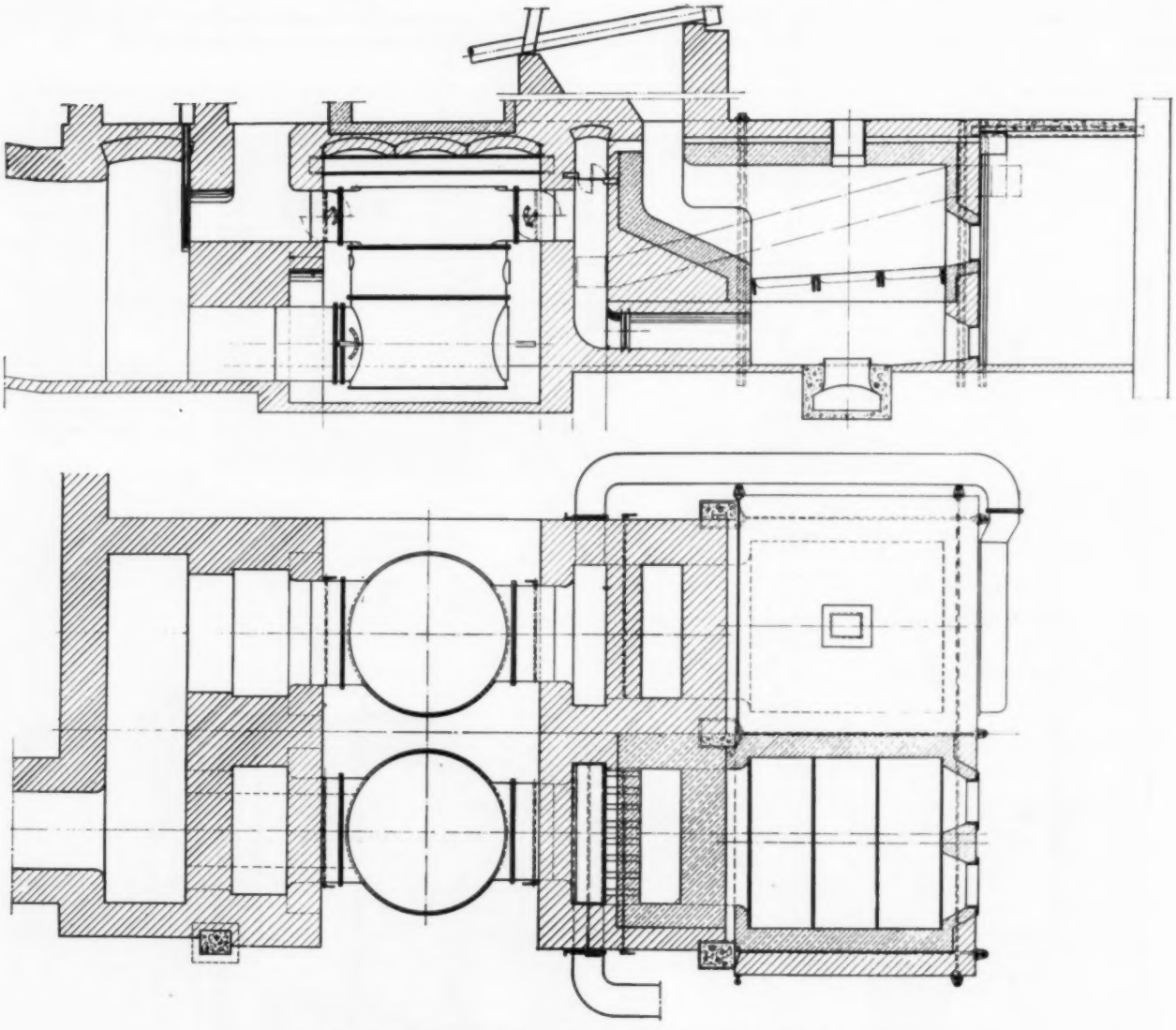


FIG. 17 FURNACE DESIGN, LUGNVIK, SWEDEN SAW MILLS

method, when dry wood refuse is not available on week-ends or shutdown periods.

In changing over from operation with shavings to pulverized coal, it is very essential that the wood refuse which has not burned in suspension and has collected on the bottom of the furnace be removed or burned completely before the pulverizer is started. The pulverized coal sometimes produces a hard crust of slag over the unburned shavings on the furnace floor. This pockets the gases created by the wood refuse, and when the boiler is down and the engineer decides to clean out this furnace, this crust is broken, releasing the gases, which may cause a dangerous explosion.

In the burning of wet hog fuel in Dutch ovens, from the description given by Mr. Gladden, it appears that nobody has conceived the idea of trying to dry the wet refuse with the stack gases. Since the temperature of the stack gases is around 600 deg. fahr. and the wet refuse is conveyed to the spouts on the Dutch oven, it seems to the writer that a very simple means of housing in the conveyor may be used, with a fan discharging the hot gases through the oven, thereby drying out as much of the moisture from the wet refuse as possible. However, there is one difficulty that arises. Although the average moisture content of the wood may be 30 to 50 per cent, there are "whiskers" and threads attached to the wood that are comparatively dry, and in trying to dry the wood with gases around 600 deg. fahr., these may burn and cause a serious fire. If the gases contain a high CO_2 content, the chance of a fire may be remote.

The writer understands that such an installation, with rotary driers, has been installed by the Ford Motor Company in Iron Mountain. Although they have had a few fires, there has been nothing serious. For the average plant, it is felt that the installation of rotary driers (although the best) would cost too much and that it would be necessary only to install a housing around the conveyor leading to the power house.

G. G. HOLLINS.[6] It has been mentioned in the discussion that it was not only a question of furnace volume, but the way the volume was arranged. The writer would assume that this refers to the relative length, breadth, and height of the furnace, and he wishes to emphasize the importance of the furnace width. Boilers are now available in so many types that a given amount of

[6] Chief Mechanical Engineer, J. G. White Engineering Corporation, New York, N. Y. Mem. A.S.M.E.

heating surface may be obtained with a furnace width most suitable for the combustion conditions. For instance, if a bent-tube boiler has approximately 14,000 sq. ft. of heating surface, practically the same amount of heating surface and vertical distance between the mud drum and the steam drum may be obtained with a furnace width of 20 ft. and with a furnace width of 26 ft. The figures, which are taken from actual design of furnace, show that for the wider boiler the furnace volume is increased 30 per cent while the area of the refractory furnace surfaces is increased only 18 per cent. The projected area of the tube bank, which is the heating surface directly exposed to the fire, is for the same drum spacing directly proportional to the width of the boiler and is 30 per cent greater for the wider boiler.

The manufacturer's data on the two types of boilers show nearly the same flue-gas temperatures for the same output and furnace conditions, so that the efficiencies are considered the same.

With the boilers operating at the same load, the boiler with the larger furnace volume and greater area of heat absorbing surface exposed to the fire might be expected so show better maintenance with the air-cooled walls, or permit higher ratings with the same maintenance.

ROBERT H. KUSS.[7] The writer lays no claims to recent contact with the problems of burning refuse wood, but feels that his experience with the art some years ago warrants these remarks.

Comparing the earlier forms of wood-refuse-burning equipments with those submitted by Mr. Gladden, it must be said that, with the exception of the more elastic furnace-roof constructions and the use of heat-absorbing walls, nothing new appears. It is true there is a greater liberality in the cubical contents of furnace or combustion chambers which may well be assumed to be a step in advance, but without taking full advantage of this place of possible improvement, the same problems persist as in the writer's experience more than a quarter of a century ago.

Specifically, there is one particular in which the paper is peculiarly silent, though being of outstanding importance. Reference is here made to the draft facilities or lack of such. Figs. 2, 6, 7, 8, 9, 10, 12, and 13 come within this category to a marked extent, and on this core alone can well account for many of the refractory failures which appear to be treated by the author as fundamental, whereas they are merely symptoms of fundamental faults elsewhere. In the majority of instances cited no precautions of oversupplying chimney capacities or induced-draft fans can overcome the draft restrictions offered by the baffling or the damper size, shape, or placement, or a combination of both faults. In some instances, notably Figs. 9, 10, and 14, the difficulties are accentuated by combustion deposits accumulating to still further restrict gas passages.

The use of excess air to afford oxygen to support delayed combustion by the admission through the rear of bridge walls or equivalent places, while perhaps better than none at all, is not as correct an attack as a distribution as early as possible into the gas-distillation zone.

A critical analysis of the settings offered by Mr. Gladden, viewed from the standpoint of the foregoing remarks, would probably direct attention away from his purpose and besides would become too voluminous. However, it may not be out of place to remark that in the related fields of the combustion of special fuels such as oil and pulverized coal the importance of the same points has been established after almost identical experiences with deficient draft facilities and misplaced auxiliary-air admission.

A. R. WAUGAMAN.[8] The author is to be congratulated on his excellent paper, which shows very clearly the development and trend of furnace design from the inefficient furnace using 0.8 cu. ft. per hp., which developed low rating and created a smoke nuisance, to the present-day furnace having 7.4 cu. ft. per hp. and developing continuous ratings as high as 300 per cent, with complete combustion and efficiency equal to or better than many present-day coal-fired installations.

The trend of development has been parallel to the improvement in design of coal-burning equipment and involves similar combustion principles. It is necessary to obtain intimate contact between fuel and air and maintain this condition until combustion is complete. The constituents of dry wood fuel shown by the average analysis of Table 1 indicate the necessity of relatively large furnace volumes to permit time for complete burning of the large percentage of volatiles in fuel before contact with the relatively cold boiler. To maintain these volatiles in a turbulent mixture presents difficulties which in the case of dry refuse furnaces are largely overcome by the method of feeding the fuel to the furnace. A large part of the fuel is burned in suspension as it falls through burning gases, and additional inherent oxygen is liberated by heat distillation at the proper time to assist the combustion process.

The method and distribution of the remainder of the air for complete combustion is very important, and a further study of the design of the bottom of the furnace where most of this air is admitted will probably further improve this type of furnace by changing the location and size of air openings to get the greatest sustained and distributed turbulence.

The proper application of water cooling or shading simplifies this problem by allowing better air control with a tighter furnace and less infiltration. It permits better turbulence and higher liberation rates without the excessive maintenance met in the earlier furnaces.

HERBERT S. WHITON.[9] The furnaces that the writer will describe are in the power station of the Mountain States Power Company at North Bend, Ore. The fuel is sawmill refuse, known as "hog fuel," and consisting largely of fir and hemlock which contain as high as 50 per cent moisture. Fig. 18 will serve to show you that the furnace consists of a Dutch oven equipped with an inclined step grate. Various details of this furnace have been altered since the first installation.' Fig. 18 gives the latest design in use at the North Bend Station.

The fuel is introduced into the furnace by gravity through diverging chutes across the full width of the front end of the furnace. It avalanches down the inclined grate due to the natural action of gravity. The fuel bed thickness is controlled by an adjustable refractory faced gate. Ashes and clinker are removed from the shelf at the lower end of the inclined grate by raking through openings immediately below the lower end of the inclined grate. The shelf is dimensioned to prevent an automatic flow of refuse into the ashpit so that the depth of the fuel bed at the back end of the furnace may be controlled.

Air is supplied by a forced draft fan, and the gases are removed by natural stack draft. Sufficient air is admitted under the inclined grate to distill the volatiles and drive off the moisture from the fuel, and in addition to burn the resulting charcoal. The air necessary for burning the volatiles is introduced through the bridge-wall tuyères, which face the inclined grate and are proportioned to give the necessary velocity to produce the desired turbulence of air and gases to hasten combustion. Some small particles of fuel carry over the bridge wall and are burned in the combustion chamber directly under the boiler heating

[7] Mechanical Engineer, Chicago Heights, Ill. Mem. A.S.M.E.

[8] Sales Engineer, Combustion Engineering Corporation, Detroit, Mich.

[9] Acting Mechanical Engineer, Byllesby Engineering and Management Corporation, Chicago, Ill. Mem. A.S.M.E.

surface. The small amount of air required in the combustion chamber is admitted by natural draft through dampered openings in the bottom of the side wall.

The boilers have 9834 sq. ft. of heating surface. The feedwater temperature is 275 deg. fahr. The steam pressure is 245 lb. gage, and the steam temperature is 540 deg. fahr. at the superheater outlet. Steaming rates of 80,000 lb. per hour are easily maintained continuously with from 15 to 16 per cent CO_2 and no appreciable CO.

No tests have been run to date. Therefore, efficiency figures

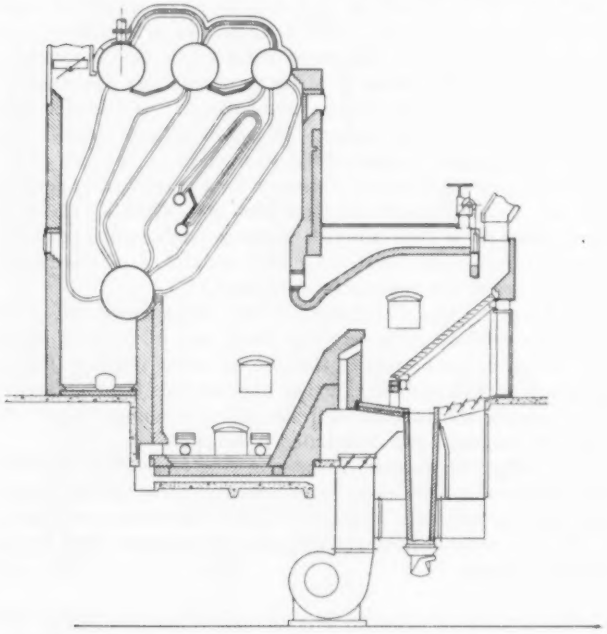


FIG. 18 DESIGN IN USE AT THE NORTH BEND STATION

are not available. However, since the installation of the two furnaces designed and the adoption of the draft adjustment, the station has generated as high as 950 kw. per unit of fuel, whereas previously the rate was about 750 kw. per unit.

AUTHOR'S CLOSURE

It is gratifying to note the interest taken in boiler settings for burning refuse wood, both by the considerable amount of discussion and by the variety of ideas brought forth.

Referring to the discussion by Mr. Beers, the author agrees with his suggestion that for certain plants, where both coal and wood must be used as fuel and especially where the wood may be relatively of minor importance and is subject to interruptions in supply, an underfeed stoker installation in a wood-burning furnace is undoubtedly a satisfactory arrangement. Care should be taken, however, to admit air above the coal for the combustion of the wood, and this should be under the control of the operator, in accordance with the rate of wood combustion.

In Mr. Cockburn's discussion he laid emphasis on the importance of uniform feed, especially when burning kiln-dried wood refuse. This is necessary, not only in the matter of obtaining high efficiency, but also in the prevention of smoke. With a system of feeding wood in which the wood is alternately admitted to the furnace in large quantities and then for a period of time in small quantities, there is the smoky condition of burning with insufficient air in the former case, and a condition of excessive heat losses when burning with excess air in the latter case. Careful attention to the uniform feeding of dry refuse wood, especially where burned in suspension, is desirable.

The evaporative tests given for the Memphis boiler, when burning wood, show excellent operation. The comparative costs of air-cooled and water-cooled boiler settings as given by Mr. Cockburn are of economic interest. The flexibility of control of the water-cooled furnace of the Memphis boiler is an excellent feature.

An application of preheated air to boiler furnaces designed for burning wet refuse wood is clearly illustrated and described by Mr. Colby. The advantage of being able to obtain one-third greater capacity, not to mention the increased efficiency by the application of an air preheater to a boiler furnace burning wet refuse wood, is decidedly worth investigation.

In the burning of bagasse, Mr. Colby has described an installation which, before preheaters were installed, would produce a maximum of 85 per cent of the rated capacity of the boiler by burning oil at the same time for supplementary firing. It is interesting to note that after the installation of the preheater it was possible to cease using oil for auxiliary firing, and the capacity of the unit was increased to 135 per cent of rating when burning bagasse alone. It is also interesting to note that the tests of the wet refuse wood-burning boilers in Sweden indicate that the application of an air preheater increased the efficiency 21 per cent, even though the wood which was burned without using the air preheater contained 7 per cent additional moisture.

Mr. Heyman has suggested a unique idea of utilizing waste furnace gases to dry the wet wood refuse before burning. In this process, care would have to be exercised not to drive off the volatile constituents other than moisture from the wood; that is, a balance between evaporation and volatilization would have to be carefully watched. It would be an interesting engineering study to compare the relative costs of equipment and operating expenses for power for such an installation, as compared with similar data for some of the designs illustrated in the preceding discussions of burning the wet refuse directly on grates in reverberatory furnaces and allowing the steam of the evaporated moisture to pass through the boiler setting carrying with it the heat losses which are inherent.

Mr. Hollins has pointed out a feature of boiler design for use in connection with the burning of refuse wood which is very important; that is, the relation of the heat absorbing area to the furnace chamber. The larger the area exposed to the combustion chamber, the greater the heat absorbing effect, which should result in lower furnace temperature and consequently less maintenance of refractories. The factor of having the side walls of the furnace as far apart as possible, consistent with good design, results in less overheating of the side walls by reflected radiant heat.

In the discussion by Mr. Kuss, the necessity for proper baffling and damper arrangements is mentioned. It is assumed that the baffling in all cases is of such a design that there is no unnecessary restriction of the flow of gases and that the gases are forced into as intimate contact as possible with the boiler tubes and also that there is sufficient draft to remove the products of combustion through the various boiler passages and damper connections. Numerous cases of faults of this kind are found, but if there is sufficient draft and the gases are properly taken away and refractory troubles exist, then it is the furnace design and not the boiler baffling and dampers that is at fault. The admission of air at the most suitable location in the furnace to support combustion of the volatile gases, especially in the cases of wet refuse furnaces, is one that depends entirely on the design of the furnace. It is the author's belief that a number of designs shown in the paper and in the discussion are worthy of careful investigation.

Mr. Waugaman has justly suggested the importance of a careful study of the distribution of air admitted at the furnace bottom for burning dry wood refuse which passes unburned through the

heated furnace. A thorough turbulence of air at the location where the pieces of unburned wood accumulate on the furnace bottom is most essential.

The author believes that the design submitted in discussion by Mr. Whiton for burning hogged fir and hemlock, containing as much as 50 per cent moisture, is of decided interest to engineers concerned with such problems. If an air preheater were also installed in connection with such an installation, it would appear that excellent arrangements for burning wet refuse wood had been provided.

FSP-53-4

Comparative Performance of a Pulverized-Coal-Fired Boiler Using Bin System and Unit System of Firing

By A. E. GRUNERT,[1] CHICAGO, ILL.

An experimental program had for its object the comparative performance of a modern boiler unit, served, first, with the bin system of pulverized-coal firing and, later, with the unit system of firing, thus making possible comparisons on a common basis. The paper deals with specific types of equipment, certain types of coal, and operating conditions, influenced somewhat by local conditions. The operation of these two systems is compared in three ways—first, on an evaporative-test basis; second, on a basis of actual overall performance; and, third, on an operating-cost basis. A record of the pertinent experiences with both systems is given. Owing to the specific nature of the program, no general conclusions are given, and there is no intention to contribute to the controversial phase of the matter.



THE BURNING of coal in pulverized form in large steam-generating units needs little if any argumentative support in view of the present state of development of the art. The controversy between advocates of pulverized-coal firing and of mechanical stokers, so much in evidence during the past few years, seems to have a tendency to settle itself. Each has its particular advantages in the general scheme of things which may make it preferential over the other according to the conditions to be met and the kind of results desired. However, engineering development in general, and more particularly that which concerns combustion processes, is not likely to be without its questions of a controversial nature sponsored by men keenly conscientious and anxious for the further development of the art.

Such a situation has and is further likely to manifest itself in connection with different systems of burning pulverized coal. Both the bin and unit systems are now used more or less extensively, the former probably with a greater span of development behind it, but the latter active with youthful vigor. Each system has appealing characteristics as exemplified by the choice between the two in recent installations. The bin system would naturally attract attention due to its inherent ease and flexibility of control over the combustion process. The unit system also appeals because of its simplicity and its adaptability to existing installations. Although the latter was a more recent arrival in central-station practice, both systems are now undeniably in successful large-scale operation.

There are, however, few definite data as yet available which would serve as a basis for comparing the economic performance of both systems. Owing to the rapid development of equipment and the complexity of so many conditions and the economics involved, this situation is not surprising at this time. Furthermore, it is not likely that any data could ever be presented which would conclusively prove the superiority of one method over the other in toto.

That each will eventually find its own field and enjoy profitable application seems a permissible prophecy in view of similar situations in the past. These data, therefore, are not presented with such a comprehensive scope in mind, but rather from the narrower viewpoint of certain specific conditions which are largely local in character.

No consideration is given in this discussion to the investment angle of the matter. While this is an important factor and perhaps the controlling one in many cases, it is broad enough to be dealt with as a separate study by those more familiar with it. Investment is primarily the concern of the designer and upon completion of a project becomes largely of retrospective interest, while the concern of the operator in subsequent performance of the equipment is continuous, perhaps for years.

Situated as we are with respect to coal supply, the economic necessity of using coal from the Central Illinois fields, as the major portion, is at once apparent. Although these coals have the generally well-known undesirable characteristics of low fusing ash, high sulphur content, and low heat values, they have, however, been successfully utilized by the chain-grate stoker when judged by the standards of a few years ago. That the development in the pulverized-coal process of combustion has forced a revision of former conceptions of good practice must be readily admitted, since there are now many outstanding examples of high capacity-efficiency performance which far surpass any stoker operation. While it is physically possible to build exceedingly large stokers, the growing tendency toward super-generating units served by a few steam-generating units places the stoker in an unfavorable position both in investment and operating considerations.

Such a situation confronted us in 1926. Chain-grate stokers had been developed to a size beyond which it would have been impractical to go. Turbo-generating units of 100,000 kw. were in contemplation, which meant, under the normal fuel and base-load conditions, a minimum of six steam-generating units. The prospect of still larger generating units required that serious consideration of the possibilities of burning Central Illinois coal in pulverized form was necessary.

The data available at that time contained little if any information on the pulverized-coal process of combustion with the kind of coal which we would have to use in future extensions; in fact, there seemed to be a preponderant usage of the better grades of coal with pulverized-coal-fired boilers. While this picture is

---

[1] Efficiency Engineer, Generating Station, Commonwealth Edison Co. Mr. Grunert was graduated from the University of Wisconsin in 1907 with the degree of B.S. He served as an apprentice engineer with Westinghouse, Church, Kerr Co., and then entered the employ of his present concern where he has held the positions of test engineer and boiler-room engineer. During the War he served from 1917 to 1919 with the E.O.R.C.

Contributed by the Power Division and presented at the Semi-Annual Meeting, Detroit, Mich., June 9 to 12, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

somewhat different under present conditions, there still appears to be no striking enthusiasm on the part of operators to use low-grade coal if any more or less reasonable pretext can be found to avoid it.

The fouling characteristic of coals with low fusing ash makes boiler cleaning a major problem in operation, especially when high capacities are maintained. This condition is probably the most important factor in differentiating our problem from that of others. All the improved appliances and methods for coping with this situation seem to fail for one reason or another.

In view of this lack of definite information, an experimental program was obviously necessary, as the commitment to large-scale operation in a new field without previous experience and trial in a smaller way might have led to grievous mistakes. This plan, conservative from the standpoint of general policy, but ambitious from an experimental standpoint, led to the installation of the well-known steam-generating unit No. 22 at Calumet Station, the general plan of which is shown in Fig. 1. Here there are represented most of the innovations in recent development of steam-generating equipment and a type which has become distinctive in character. Having in mind its particular application to pulverized-coal firing with low-grade coals, it has the usual large combustion chamber which is almost completely water-screened. It utilizes the idea of a small boiler proper, to more fully obtain the advantages of increased heat absorption by direct radiation, open tube spacing to facilitate cleaning, combined with a direct-connected steaming economizer and a relatively large air preheater. That all these innovations, and others of minor importance, have their particular distinctive advantages needs no further mention. It is the forerunner of its type; the more recent installations differing from it only in size and more improved methods of handling slag ejection from the furnace. A more detailed description of this unit is given in Table 1, both as it was originally installed and as it is now.

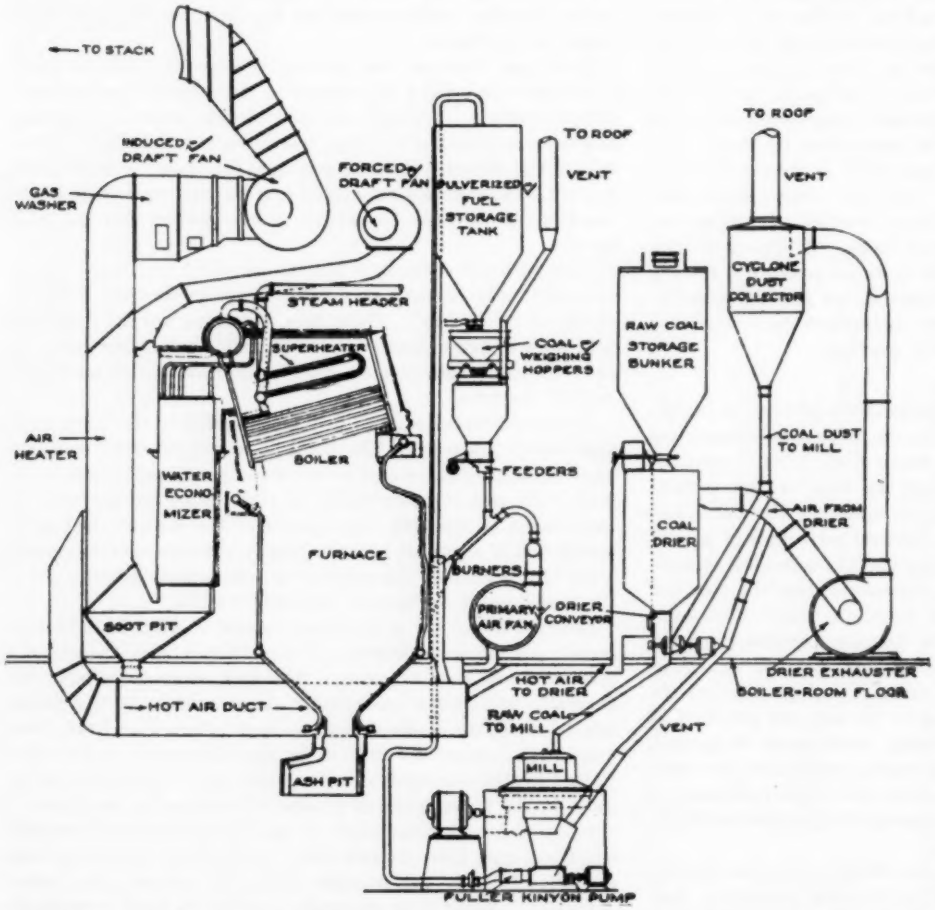


FIG. 1 BOILER UNIT NO. 22, CALUMET STATION, COMMONWEALTH EDISON COMPANY, WITH A BIN SYSTEM OF FIRING

This unit, as shown by Fig. 1, was first put in service November 8, 1926. Owing to certain plant conditions not related to this unit, it was necessary to adopt a 16-hour-per-day operating schedule, with Sunday and night outage, and, except at certain periods for experimental reasons, this has been substantially its normal operation ever since. A steady rating of about 140,000 lb. of steam per hour has been usually maintained during the day, with some increase at the evening peak and a noon-hour drop. During the night all auxiliaries are out of service, the dampers are closed, and the steam pressure is allowed to drop. From such a warm condition the unit could be brought up to normal steaming capacity in about 15 minutes. When the boiler has cooled off over a Sunday shutdown, bringing up requires 50 to 60 minutes. The flue blowing elements are operated on a routine schedule depending on coal conditions. The boiler unit is normally on automatic control, with manual load regulation except when starting up, during flue blowing operations, and when dumping the ashpit. These were the general operating conditions upon which the comparative operating data are based. The frequent shutdowns occasioned by such a schedule affected the operating efficiencies somewhat adversely, as would be expected.

Since the experimental plan was intended to be as thorough and comprehensive as practicable with the specific conditions involved, the operation of this unit has been in effect a continuous operating test. All the metering devices, particularly the steam-flow meters, were installed with adequate means for calibration at frequent intervals, so that the overall results would be within the degree of accuracy ordinarily accepted as good practice. While the burning of Central Illinois coal in pulverized form was the principal consideration, other coals were burned over sufficiently long periods to ascertain the effect on operating conditions, if any, caused by the coal quality. Table 2 summarizes the 24 months' operation of this unit when equipped with a bin and feeder system. From this record is drawn a summary of seven months of operation with Central Illinois coal, two months with Youghiogheny, Pa., coal, and three months with Western Kentucky coal. During the other twelve months not shown separately, various Illinois and Kentucky coals were burned in varying proportions. These months were left out of compara-

TABLE 1 DESCRIPTION OF EQUIPMENT, BOILER NO. 22, CALUMET STATION, COMMONWEALTH EDISON COMPANY

As a Bin System	*As a Unit System*
Boiler:	
Type: B. & W. cross drum	Same
Surface: 5,940 sq. ft.	
Furnace:	
Dimensions: Plan, 19 ft. 9 in. by 23 ft. 8 in.	Same
Volume: 12,397 cu. ft.	
Wall construction: Bailey water-cooled furnace walls and floor. Refractory-faced blocks on furnace walls and bare cast-iron and steel blocks on furnace floor. Blocks bolted to 3½-in. tubes spaced on 6-in. centers. Heating surface of walls; 2,468 sq. ft.	
Superheater:	
Convection, three-pass	Same
Surface: 3,000 sq. ft.	
Economizer:	
Type: B. & W. loop, counterflow	Same
Surface: 8,850 sq. ft.	
Air preheater:	
Type: Tubular, single-pass, counterflow	Same
Surface: 41,700 sq. ft.	
Induced-draft fan: Buffalo Forge Company	Same
Type: Duplex conoidal, size 13	
Capacity: 144,000 cu. ft. per min. at 880 r.p.m., 300 deg. fahr., 12-in. static pressure	
Drive: Two-speed motor	
Forced-draft fans: (2) Buffalo Forge Company	Same
Type: Duplex conoidal, size 6½	
Capacity: 58,000 cu. ft. per min. at 70 deg. fahr., 4-in. static pressure at 1,102 r.p.m.	
Drive: Two-speed motor	
Gas washer equipment:	
Washer: Carrier Air Conditioning Company. Spray capacity, 500 g.p.m. Pump capacity, 900 g.p.m. at 135-ft. head	Same
Drier: Fuller Randolph	Removed
Type: Vertical, tempered preheated air	
Capacity: 30,000 lb. per hour	
Raw-coal scales: None	Richardson Automatic
	Capacity: 400 lb. per dump
Drier exhauster: Buffalo Forge Company	Removed
Type: R, size 130	
Capacity: 57,500 cu. ft. per min. at 140 deg. fahr., 7-in. static pressure, 490 r.p.m.	
Drive: Variable-speed motor	
Cyclones: (2) Fuller Lehigh Company	Removed
Size: 144 in.	
Drag feeder: Fuller Lehigh Company	Fuller Lehigh Company
Drive: Variable-speed d.-c. motor	Variable-speed d.-c. motor
Pulverizer: Fuller Lehigh Company	Fuller Lehigh Company
Type: B, 70-in., screen separation, four-ball, pusher	70-in., air separation, five-ball, pusher type
Capacity: 30,000 lb. per hour	30,000 lb. per hour
Drive: Variable-speed, 375 r.p.m. maximum	Variable-speed, 510 maximum r.p.m.
Mill exhauster fan: None	Overhung paddlewheel type
	Drive: 1150-r.p.m. motor
Pulverized coal-transport system:	Removed
Fuller Lehigh Company	
Coal pump: Fuller Kinyon	
Capacity: At least 60,000 lb. per hour	
Drive: Constant-speed motor	
Pulverized-coal bunker:	Not used
Capacity: 200 tons	
Pulverized-coal feeders:	Removed
Four Bailey feeders	
Capacity: 10,000 lb. per hour each	
Drive: Variable-speed motor	
Pulverized-coal weigh bins and scales:	Removed
For test	
Primary-air fan: Buffalo Forge Company	Removed
Type: R, size 115	
Capacity: 15,000 c.f.m. at 200 deg. fahr., 10-in. static pressure and 570 r.p.m.	
Drive: Constant-speed motor	
Burners: (8) Bailey "Calumet" type	(6) Same type
Size of primary opening: 62 in. by ½ in.	62 in. by 1.12 in.
Control: Bailey Meter Company	Same
Meter: Steam-flow–air-flow type D 25	
Multi-pointer gage: Type P 12 F.	
Temperature recorders: Gas in and out of economizer, and water in and out of economizer; steam temperature, flue-gas temperature at preheater out and air temperature at preheater out	
Steam-flow–air-flow and furnace-draft contactors: Type RS 36	
Master drum-switch: Type R 65	
Control drive: Type RC 76 D.S.T. (draft control)	
Control drive: Type RA 76 ST (fuel control)	
Bank of rheostats: Type R 414	

tive summaries so as to eliminate any question of unfair comparison.

An accurate record of all pertinent operating costs was kept, and is summarized as shown by Table 3. In this cost-accounting schedule only those operating expenses are considered which are affected, directly or indirectly, by the method of burning coal. In general, these accounts include the operating expenses entailed in the plant system from the point at which coal is received into the car dumper to the point at which steam is delivered to the header system. This seemed necessary due to the specific nature of the problem. Some of the items might have a somewhat remote connection, but by simple arithmetical alterations the basis of comparison could be changed to suit the individual ideas of any one interested. Due to changing of the coal supply previously mentioned, it was not practical to attempt to separate the cost for each period, but rather to group them for the entire period of bin and feeder operation.

TABLE 2 SUMMARY OF ACTUAL OPERATION, BOILER NO. 22, CALUMET STATION, AS A BIN SYSTEM

	Ky. and Va.	Cent. Ill.	Yough-iogheny	Ill., Ky., Pa. and Va.
Coal source				
Period, months	3	7	2	24
Steam delivered, M lb.	128,082	308,726	114,759	938,299
Coal fired, tons	7,588	20,650	6,554	60,986
Coal analysis, as fired:				
Moisture, per cent	2.03	7.14	2.31	.....
Ash, per cent	13.31	12.94	12.29	.....
B.t.u.	12,483	11,338	12,613	.....
Coal analysis, as weighed:				
Moisture, per cent	5.77	12.90	6.60	10.7
Ash, per cent	12.80	12.13	11.75	12.0
B.t.u.	12,006	10,626	12,058	11,037
Evaporation per lb. coal, lb.	8.439	7.475	8.755	7.693
Steam temperature, deg. fahr.	687	664	683	676
Steam pressure, lb. gage	310	311	315	310
Steam superheat, deg. fahr.	262	239	257	251
Feedwater temperature, deg. fahr.	171	189	215	182
Added heat per lb. steam, B.t.u.	1220	1191	1175	1203
Evaporative efficiency, per cent	85.75	83.74	85.28	83.85
Auxiliary power, kw-hr. per ton:				
Induced-draft fan	26.44	19.98	22.64	25.00
Forced-draft fans	8.21	7.79	7.07	8.03
Pulverizer	13.98	14.05	12.23	13.65
Drier exhauster	5.38	5.46	4.57	5.11
Primary-air fan	2.36	2.24	2.62	2.32
D.C. feeders and automatic control	0.32	0.35	0.39	0.35
Fuller Kinyon coal pump	1.07	0.85	0.70	0.87
Total, kw-hr. per ton	57.76	50.72	50.22	55.33
Total auxiliary power, M kw-hr.	438	1047	329	3373
Steaming hours	902	2318	792	7124
Average evaporation, M lb. per hr.	142	133	145	132
Mill hours	749	1969	669	5858
Tons coal per mill hour	10.1	10.5	9.8	10
Coal fineness, Tyler standard screens:				
Through 48-mesh	98	99	99	98
Through 100-mesh	93	87	87	86
Through 150-mesh	75	80	80	78
Through 200-mesh	65	68	68	67

TABLE 3 SUMMARY OF ACTUAL OPERATING COSTS, BOILER NO. 22, CALUMET STATION, AS A BIN SYSTEM (SEE TABLE 2)

	Actual cost in dollars	Dollars per ton coal fired	Dollars per M lb. steam delivered
703-A coal- and ash-handling labor	1,066	0.01748	0.00114
703-B boiler-room labor	10,387	0.17032	0.01107
Total operating labor	11,453	0.18780	0.01221
717-A maintenance, furnace	2,569	0.04202	0.00274
717-C maintenance, boiler and superheater	1,157	0.01897	0.00123
717-D maintenance, feeders and burners	532	0.00873	0.00056
718-A maintenance, economizer	1,430	0.02345	0.00152
718-D maintenance, soot blowers	3,060	0.05032	0.00327
718-E maintenance, air preheater	1	0.00001	.....
718-F maintenance, all fans	2,410	0.03952	0.00257
Total maintenance except coal and ash equipment	11,168	0.18312	0.01190
718-B Pro-rate share of coal conveying	2,341	0.03904	0.00254
718-B Maintenance, coal conveying, No. 22	2,260	0.03640	0.00237
718-C Pro-rate share of crusher and breaker	294	0.00479	0.00031
718-C Mill maintenance	3,774	0.06192	0.00402
Total maintenance coal and ash equipment	8,669	0.14215	0.00924
Auxiliary power, actual	12,862	0.21090	0.01371
Fuel cost	240,392	3.94175	0.25618
Total cost	284,544	4.66570	0.30324

In October, 1928, the equipment was converted to the unit system of firing, the general scheme being that shown in Fig. 2. Although this change was not contemplated in the original plan, the many and varied experiences of two years of operation with a bin and feeder system were such that no particular pangs of regret were evidenced by those concerned when this change took place. It is not inferred that this conversion eliminated all objectionable conditions, because the unit system in its later opera-

tion also developed a few inherent mulish traits. Table 4 summarizes eleven months of operation of this unit with a unit system of firing, and Table 5 is the operating-cost record. The operation

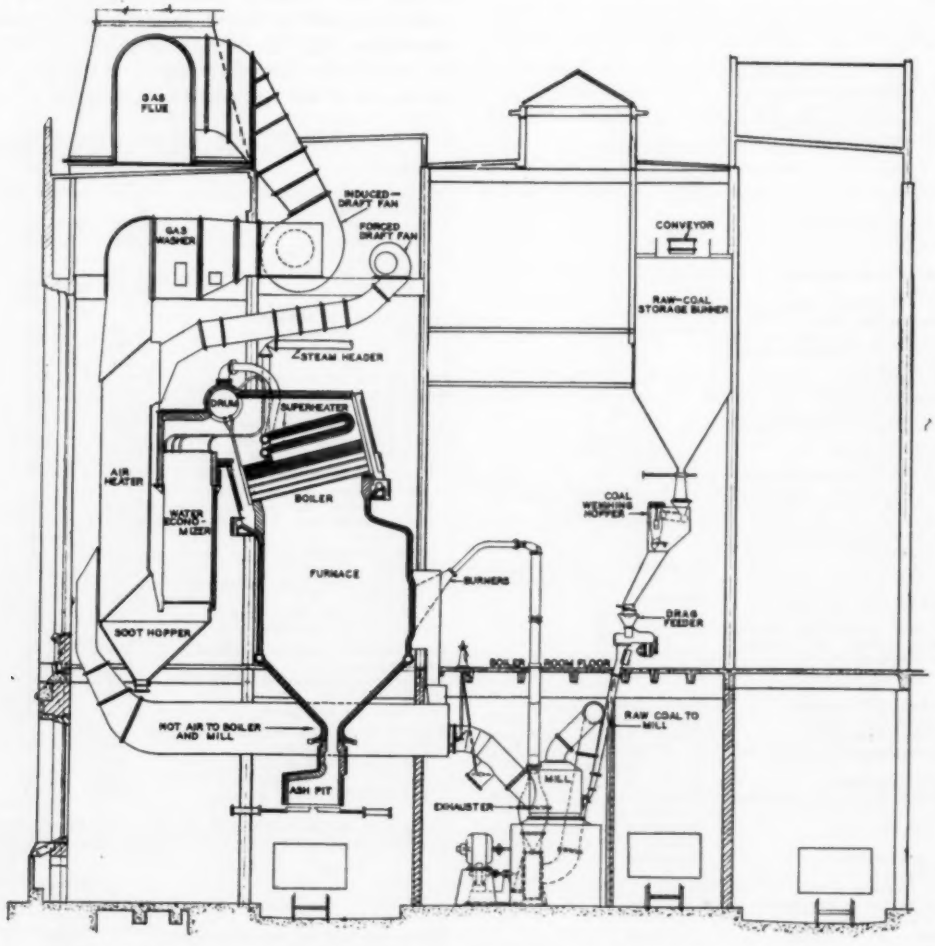


FIG. 2 BOILER UNIT NO. 22, CALUMET STATION, WITH A UNIT SYSTEM OF FIRING

and the experimental work thus far have made it possible to compare results in general, as follows:

Bin System	*Unit System*
Operating experiences and comment	Operating experiences and comment
24 months' overall operating performance, all coals. Table 2	
7 months' operating performance, Central Illinois coal. Table 2	11 months' operating performance, Central Illinois coal. Table 4
3 months' operating performance, Western Kentucky coal. Table 2	
2 months' operating performance, Youghiogheny coal. Table 2	
Test results, Central Illinois coal. Table 6	
Test results, Western Kentucky coal. Table 7	Test results, Central Illinois coal. Table 9
Test results, Youghiogheny coal. Table 8	
Operating costs, 24 months' basis, all coals. Table 3	Operating costs, 11 months, with Central Illinois coal. Table 5

BIN SYSTEM OPERATION

Furnace and Ash Disposal. The water-cooled furnace has probably been one of the most important factors contributing to the successful application of pulverized-coal firing to large-scale and high-rating operation. The exacting furnace conditions to be met with in many cases would make the refractory-lined furnace a sorry spectacle in comparison. This furnace design as then intended, and as verified later, so reduces the factor of destructive slag action that the operating problem in the furnace has become of insignificant importance. So far as the screen walls are concerned, only 30 blocks have been replaced in over three years, and this failure was largely due to the proximity and impingement of the end burners on the side walls. A more practical spacing of the burners was the logical solution. Other mistakes in the design of this furnace are exemplified by the refractory-lined spaces below the inlet header of the side screens and that above the outlet header of the rear screens, the former failing due to slag erosion and the latter largely due to inadequate anchorage. The furnace-maintenance cost shown is almost entirely confined to these areas. Fig. 3 shows an end wall as installed, and Fig. 4 shows the condition of the same wall after 38 months' operation. Any failure of these screen walls is likely to be a matter of the far-distant future.

During the early operation of this furnace a study was made of the heat-transfer condition with these screens. The typical conditions are shown by Fig. 5. Higher heat transfers were obtained over certain areas, such as the rear walls and part of the side walls, than the mean values shown, due to burner and impingement conditions. A maximum heat liberation of 34,000 B.t.u. per cubic foot per hour, for short periods, has been obtained in this furnace.

The method of slag and ash removal from the furnace developed no particular difficulties when firing coals having an ash with a low fusion point such as that from Central Illinois. The molten ash deposited on the walls runs down freely to the ashpit throat

TABLE 4 SUMMARY OF ACTUAL OPERATION OF BOILER NO. 22, CALUMET STATION, AS A UNIT SYSTEM

Steam delivered, M lb.	509,820
Coal fired, as weighed, tons	34,771
Coal analysis, as weighed:	
Moisture, per cent	14.06
Ash, per cent	12.68
B.t.u.	10,314
Evaporation, lb. steam per lb. coal, lb.	7.332
Steam temperature, deg. fahr.	678
Steam pressure, lb. gage	314
Steam superheat, deg. fahr.	253
Feedwater temperature, deg. fahr.	209
Added heat per lb. steam, B.t.u.	1178
Evaporative efficiency, per cent	83.7
Auxiliary power, kw-hr. per ton:	
Induced-draft fan	21.10
Forced-draft fans	6.22
Pulverizer	17.63
Mill exhauster	6.11
Mill motor-cooling fan	0.13
D.C. for feeders and automatic control	0.12
Total	51.31
Total auxiliary power, M kw-hr.	1784
Steaming hours	3479
Average evaporation, M lb. per hour	147
Average tons coal fired per hour	9.9
Coal fineness, Tyler standard screens:	
Per cent through 48-mesh	98
Per cent through 100-mesh	84
Per cent through 150-mesh	74
Per cent through 200-mesh	65

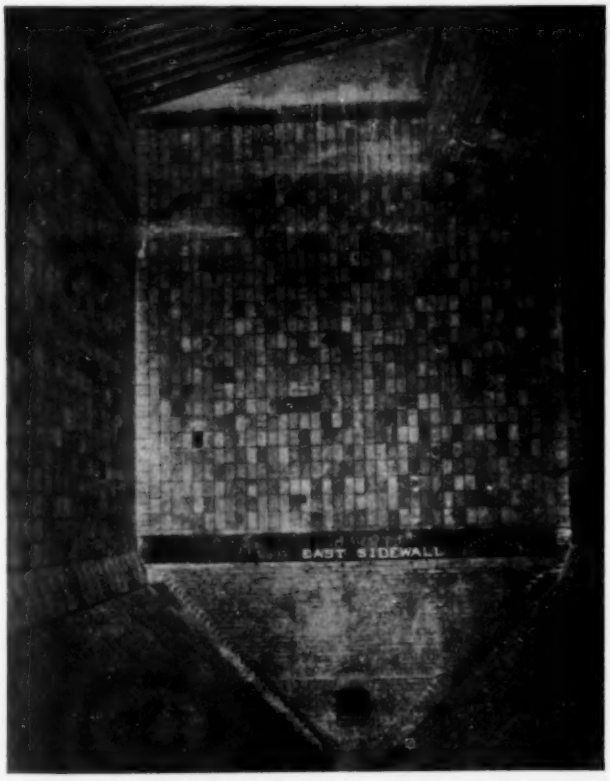


FIG. 3 PHOTOGRAPH OF FURNACE WALLS, BOILER NO. 22, CALUMET STATION, AS INSTALLED

and then drops into the ashpit. A condition which in so many other respects is objectionable is in this respect a distinct advantage. The slag formation is greatest on the rear wall due to burner position, as might be expected. The deposits in the boiler-tube bank that are dislodged in cleaning fall and roll into the pit quite freely. The slag and other deposits are quenched in the pit to facilitate handling. Occasionally slag formations will concentrate on the refractory area at the ends of the hopper throat and require rodding through the special doors provided.

When firing a coal having an ash of higher fusion temperature such as Youghiogheny, the fly ash deposits on the walls do not readily flow. This is apt to cause large accumulations which, on breaking loose during operation, may clog the bottom hopper throat. While some measure of control over this is possible with burner operation, this condition did cause a shutdown and several cases of emergency rodding operations. The throat width of 18 in. could probably have been made larger to advantage.

In firing the normal coal, no particular difficulties are encountered so far as ash removal from the furnace is concerned. However, the operation of dumping the refuse from the pit is a dirty, gassy job and not without conceivable hazards. Later develop-

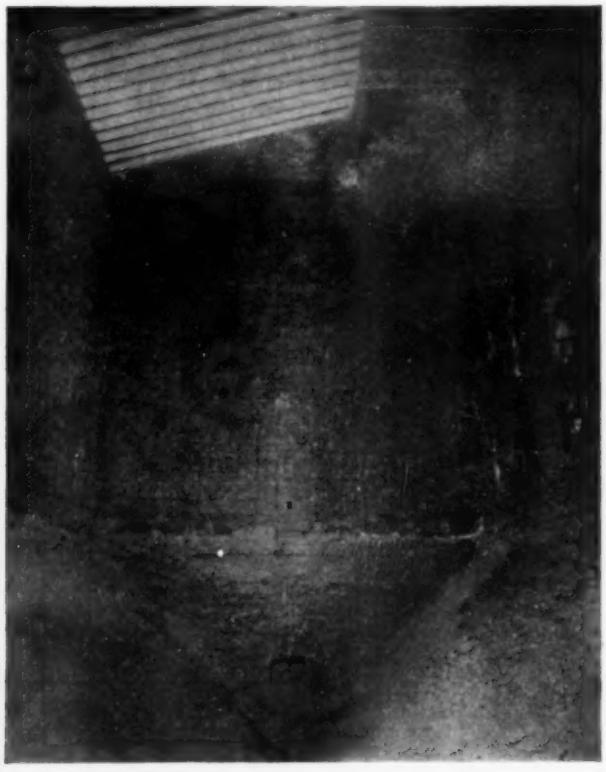


FIG. 4 PHOTOGRAPH OF FURNACE WALLS, BOILER NO. 22, CALUMET STATION, AFTER 38 MONTHS' OPERATION

ments in the manner of slag removal seem to offer improvement over this method.

TABLE 5 SUMMARY OF ACTUAL OPERATING COSTS, BOILER NO. 22, CALUMET STATION, AS A UNIT SYSTEM (SEE TABLE 4)

	Actual cost in dollars	Dollars per ton coal fired	Dollars per M lb. steam delivered
703-A coal- and ash-handling labor	712	0.02407	0.00141
703-B boiler-room labor	4,900	0.14093	0.00960
Total operating labor	5,612	0.16140	0.01101
717-A maintenance, furnace	1,009	0.02898	0.00198
717-C maintenance, boiler and superheater	530	0.01524	0.00104
717-D maintenance, feeders and burners	2,403	0.06916	0.00472
718-A maintenance, economizer	354	0.01017	0.00069
718-D maintenance, soot blowers	251	0.00722	0.00049
718-E maintenance, air preheater	....	....	....
718-F maintenance, all fans	817	0.02349	0.00160
Total maintenance except coal and ash equipment	5,364	0.15426	0.01052
718-B Pro-rate of maintenance, station coal and ash equipment	1,396	0.04015	0.00274
718-B Maintenance, No. 22 coal and ash equipment	478	0.01375	0.00094
718-C Pro-rate of maintenance, crusher and breaker	1,916	0.05510	0.00375
718-C Maintenance, mill	6,209	0.17857	0.01218
Total maintenance, coal and ash equipment	9,999	0.28757	0.01961
Auxiliary power, actual	7,684	0.22099	0.01507
Fuel cost	125,592	3.61196	0.24633
Total cost	154,251	4.43618	0.30254

The economizer and preheater hopper deposits are piped and flushed to the ash car. Clogging of these drains is an occasional annoyance. Normally, about 40 per cent or less of the original ash in the coal is accounted for by way of the ashpit; the remainder is carried out of the furnace as fly ash in the flue gas.

Boiler, Superheater, Economizer, and Preheater. Although burning Central Illinois coal in pulverized form has greatly reduced the operating difficulties due to fouling of the boiler heating surface at ordinary ratings, the situation at relatively high sustained ratings still leaves much to be desired. The rate of fouling in the lower deck increases faster than the increase in rating until finally a rating is reached where the usual methods of cleaning during operation fail miserably. Probably no operating problem has

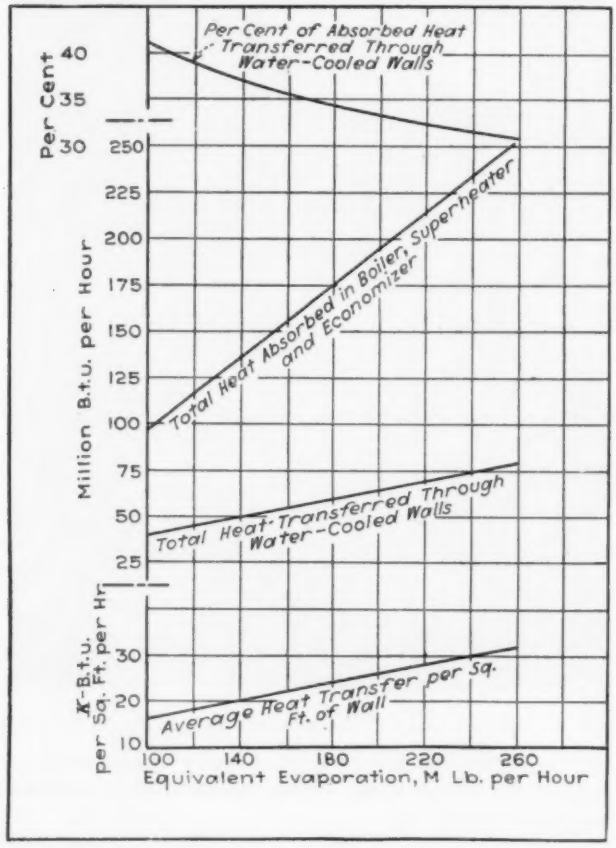


FIG. 5 TYPICAL HEAT TRANSFER THROUGH FURNACE WALLS, BOILER NO. 22, CALUMET STATION

been given more thought and attention than this, and invariably all the pet schemes contrived fall short of obtaining the desired results due to one cause or another. The flue blowing equipment in the fouling zone is subjected to temperatures beyond the limitation of the metal involved. This matter was an important factor in limiting the capacity of every-day operation to that shown, although short-time operation at much higher ratings was possible. Sustained ratings up to about 200,000 lb. per hour could be obtained with reasonable efficiency by frequent blowing and rodding. Beyond this capacity fouling difficulties increase to such an extent that continued operation was increasingly difficult if not impossible. After operating at a rating of about 275,000 lb. for a few hours it was usually necessary to shut down for cleaning.

The situation with the better grades of coal is greatly improved due to lesser ash quantities involved and more favorable fusion temperatures. In fact, a sustained rating of 250,000 lb. was not only possible, but could be maintained with reasonable efficiency. The idea that pulverized-coal firing is better suited to the poorer grades of coal seems to have suffered a relapse in this case.

The cleaning of the economizer and preheater surfaces has not been a serious problem. The nature of the deposit is such that it can be removed by ordinary means. It does not adhere like that from stoker firing, due to its different character. The corrosive action of these deposits is far less; in fact, after three years' operation, none is in evidence. During the early operation of the preheater the inlet ends of the tubes would plug shut. A special movable blowing element which, in moving across the preheater inlet-tube sheet, allowed each tube to receive the full force of a steam jet, gave quite satisfactory results. Some manual work is required occasionally on the outlet ends and upper tube sheet during an outage period.

Gas Washer. Public opinion forces attention to the consideration of fly ash and sulphur fumes ejected with the flue gases. This is true of any method of burning coal, but more particularly of pulverized coal. The experimental program included investigations as to the efficacy of the washer principally in eliminating these nuisances. The washer-type equipment installed between the preheater and the induced draft fan is practically the same as ordinarily used for cleaning generator cooling air. A separate 900-g.p.m. pump supplies water to the spray nozzles. Much has and is yet to be learned about equipment for this purpose.

In its early operation, serious corrosion due to the sulphur acids manifested itself. As a means of protection, all duct work between sprays and fan suction were covered with "gunite." Rubber-lining the drain piping was necessary due to the acid conditions. Subsequent operation for about three months so corroded the spray elements and support work that further operation was useless. The eliminator plates at the bottom were badly eaten away and somewhat embrittled. It follows then that use of this type of eliminator with high-sulphur coals requires materials highly resistive to corrosion. Experiments with special alloy metals are now under way. In a large installation satisfactory disposal of the drain water would present certain problems.

The effect of the flue gases on the induced-draft-fan runner is important. The eroding effect of fly ash over the periods of no washing was severe, resulting in a fan wreck after 20 months' operation. On the other hand, although the period of washer operation was short, some corrosion was evident, which may indicate a choice of two evils.

It was expected that the washer would not only eliminate dust, but would also considerably reduce the amount of sulphur in the gases. Upward of 90 per cent of the solid matter in the gas can be eliminated by this method, which compares quite favorably with other methods in use, but no appreciable reduction of the SO_2 and SO_3 could be found. As these gases existed in the flue gases to the extent of about 0.3 to 0.4 per cent by volume, it follows that a high degree of spray atomization would be necessary for complete effectiveness. Since the solubilities of these gases decrease with their partial pressures, it would follow that the degree of absorption would become increasingly difficult, involving large quantities of water and possibly multiple-stage spray banks for high humidity. That the refinements claimed to be necessary to the process were only crudely met in this case is readily admitted. It seems that a treatment which has the inherent possibilities of removing both dust and noxious gases is one worthy of serious consideration and effort to overcome the difficulties involved.

Pulverizer. Pulverizing this high-ash coal by either method of firing still leaves much to be desired, and the developments taking place in this case show it to be no exception. The operator

visualizes the situation as one that costs a lot of money for maintenance of such equipment and something which falls short of ordinary ideas of dependable service. Bin-system operation makes mill emergency outage of less importance than unit-system operation because of its principal premise. In this case ordinary repair and emergency outage affected the boiler operation but little, which is a point of importance in comparing the two systems.

The bin-system pulverizer was a pusher, ball, and screen type. Although it could have been operated at varying speed, it was usually operated at a fixed speed favorable to drying and sizing considerations. The maintenance of this mill was mostly confined to the renewal of balls, broken pushers, ball race, and screens. Chilled-iron balls gave more service than forged-steel balls. The balls became egg-shaped after a time, with the usual consequences of sliding and causing the breakage of other parts. The wearing of the working parts was also dependent upon the kind of coal being milled. The conditions with Central Illinois coals were much more severe than with the others, due to the abrasive action of ash and impurities. The maintenance costs over this period were influenced favorably due to milling more friable coals over part of the time.

Drier. This type of separate drier gave only the usual commonplace difficulties, but ones that could be agreeably dispensed with. Firing of the coal due to hanging up was frequent. This was particularly the case when coal was allowed to lie in the drier for a considerable length of time. The milling conditions in this case were tied up to a large extent with this drier. When the mill capacity was increased too much, there was insufficient drying, which resulted usually in plugged screens. At low capacities the coal was apt to be overdried, causing smoldering. Careful control over the temperature conditions was necessary at all times. As this drier was sectionalized, some approximate control over these conditions was possible. The best milling conditions were obtained with Central Illinois coal when the original moisture of 14.5 per cent was reduced to about 9 per cent and with Kentucky coal when the moisture was decreased from 8.5 per cent down to about 3 per cent.

Vent Cyclones. Two cyclone separators were installed to take care of the mill vent and drier dust combined. Their early operation was in parallel and later in series; and still later a supplementary washing apparatus was installed in the cyclone discharge. Of course, the usual amount of separation took place, the recovery being returned to the mill, but the amount of dust getting through in spite of all the opportunities given it to separate was sufficient to be a nuisance, not only to the plant itself, but to the neighborhood as well. The coal loss was insignificant, but a small amount in this form becomes an intolerable nuisance. Some consideration was given to installing special equipment for separating this residue but the fact that such equipment was necessary is an objection in itself. Perhaps no single item influenced those concerned in deciding to change to a unit system more than this particular situation. It is doubtful even if any great measure of satisfaction would have been obtained with such a system had the elimination been perfect.

Transport and Bunkers. The method of coal transport offered no particular difficulties. Those experienced at the bunker were confined to irregular flow and bridging of the coal at the outlets. One bunker fire was experienced and successfully smothered with carbon dioxide. Hanging up and packing of coal on the sides of the bunker due to reabsorption of moisture necessitated infrequent cleaning. In general, bunker conditions were probably no different from those of many others.

Feeders, Weigh Scales, and Burners. Excessive wear on the feeder mechanism, due to the abrasive action of the coal, developed early in the operation. "Stelliteing" of the wearing parts improved this situation, but probably not permanently. The volumetric metering feature of this feeder was not satisfactory for accurate measurement. The weighing of coal in pulverized form developed certain difficulties. This scheme, used only for evaporative test purposes, was similar to that used for water. It was found that the weigh tanks must be positively vented above and below to prevent diaphragm action. This system of weighing coal has many discouraging aspects.

The burners as first installed gave an unequal coal distribution through the primary openings, which resulted in an excessive amount of coal running out at the bottom. This was a burner condition and was remedied by placing deflector plates so that no reduction in velocity could occur in the burner. The combustion conditions obtained with this type of burner as changed leave little to be besired.

Control System. The automatic combustion-control equipment used on this boiler serves the purpose of regulating the air supply to the furnace and the furnace draft. The control of boiler output is entirely manual. Corrective adjustments to the air supply originate in the air-flow-steam-flow meter, which also records the result. The control is arranged to supply 25 per cent excess air at normal output and 30 per cent at low rating to avoid smoke.

In order to avoid any hazardous situation that might arise due to the failure of any electrically driven boiler auxiliary, an interlock system is provided. The principle of this system is shown in the following example; if the induced- or forced-draft fan should trip out, the delivery of coal to the furnace would cease, due to the automatic stopping of the necessary equipment. This system forces a definite sequence in the starting up and shutting down.

Unit-System Operation

The operation of the boiler unit proper was influenced very little by the conversion to the unit system. The conditions described as to the furnace, ash disposal, boiler, economizer, and preheater are much the same except as explained in the following.

The unit system reduced the maximum capacity obtainable in this case because of the limitation of mill capacity. This is obvious with this system. Ordinarily more than one mill would be provided for such a steam-generating unit, but inasmuch as we were concerned primarily with the experimental angle of the problem, one mill was installed—that one being the former mill rebuilt.

Drying of the coal for suitable pulverization was accomplished in the mill, the air being taken from the main duct at about 350 deg. fahr. and the coal-air mixture discharged to delivery ducts at about 140 deg. fahr. About the same amount of drying was obtained by this means as in the bin system, which is quite satisfactory with the kind of coal handled. The coal-air mixture is split two ways, each stream serving three burners. The coal sizing is controlled somewhat by cone classification and primary air volume, these methods having certain limitations, as might be expected.

Probably the reliability and satisfactory operation of a unit system of firing will in general be dependent upon the milling problem and related factors. As continuity of the combustion process is directly dependent upon these, the better position of the bin system in these respects must be admitted whether that advantage be considered small or large. We must accept the premise that an emergency outage of a unit will immediately affect steam output; and hence duplication, and perhaps multiplicity of mill units, is required to maintain continuity. Also there are the requirements of the proper air volume for milling, the required air for the best burner conditions, and the matter of distribution to various burners. That the coordination of these three more or less conflicting requirements is difficult developed

TABLE 6 EVAPORATIVE TESTS ON BOILER NO. 22, CALUMET STATION, WITH A BIN SYSTEM OF FIRING, USING KENTUCKY COAL

Test no.	18	12	16	9	15	10	14	11	13	17	20	21	19
Date of test	Nov. 3	Oct. 12	Oct. 27	Oct. 4	Oct. 26	Oct. 6	Oct. 20	Oct. 11	Oct. 19	Oct. 28	Nov. 10	Nov. 11	Nov.4
Duration, hours	7	7	7	7	7	7	7	7	7	7	7	7	4
Coal as fired:													
Moisture, per cent	2.40	2.06	2.78	2.11	2.57	2.23	2.52	2.19	2.42	2.52	2.48	2.60	2.47
Ash, per cent	14.37	14.27	13.43	15.21	13.67	14.38	13.69	14.42	14.16	13.80	14.38	14.03	14.07
B.t.u. per lb.	12089	12202	12160	12010	12152	12152	12169	12113	12139	12165	12081	12098	12117
Ultimate fuel analysis, per cent:													
Carbon	83.17	83.69	83.22	83.69	83.25	83.69	82.96	83.69	83.03	83.16	83.03	82.99	83.03
Hydrogen	5.47	5.51	5.47	5.51	5.47	5.51	5.45	5.51	5.46	5.47	5.46	5.46	5.46
Sulphur	1.47	1.63	1.42	2.20	1.37	2.30	1.72	2.29	1.64	1.48	1.64	1.68	1.63
Nitrogen	1.78	1.79	1.78	1.79	1.79	1.79	1.78	1.79	1.78	1.78	1.78	1.78	1.78
Oxygen	8.11	7.38	8.11	6.81	8.12	6.71	8.09	6.72	8.09	8.11	8.09	8.09	8.10
Size of coal (Tyler screens), per cent:													
Through 28-mesh	100	100	100	100	100	100	100	100	100	100	100	100	100
Through 48-mesh	96.6	97.3	97.4	97.7	97.2	97.7	97.1	97.9	95.4	97.1	96.8	97.5	96.6
Through 100-mesh	80.0	78.5	80.1	81.0	79.5	84.2	80.0	83.7	81.1	80.2	80.0	82.7	79.2
Through 150-mesh	72.1	70.7	72.1	72.9	71.0	73.2	71.2	76.4	73.1	72.3	72.0	75.2	70.6
Through 200-mesh	62.7	59.7	59.7	61.0	60.4	64.1	61.6	64.4	64.0	58.2	61.5	61.0	60.3
Feeders in service	2 & 3	2 & 3	2 & 3	2 & 3	2 & 3	2, 3 & 4	2, 3 & 4	2, 3 & 4	All	All	All	All	All
Firing control	Man.	Auto.	Auto.	Auto.	Auto.	Auto.	Auto.	Auto.	Man.	M. & A.	Man.	Man.	Man.
Coal burned per hr., lb.	12947	13515	13940	16456	16651	19155	19113	21874	24085	24889	26869	28879	34434
B.t.u. liberated per cu. ft. furnace, vol. per hr.	11730	13300	13680	15920	16320	18800	18760	21400	23600	24400	26230	28170	33700
Water evaporated per hr., lb.	108292	121419	124534	146285	147639	167777	170195	187474	213690	214311	228749	250783	290179
Actual evap. per lb. coal per hr., lb.	8.99	8.98	8.93	8.90	8.87	8.77	8.90	8.57	8.87	8.62	8.52	8.69	8.43
Total equiv. evap. per lb. coal per hr., lb.	10.97	11.11	11.10	11.06	11.02	11.01	11.16	10.84	11.21	10.90	10.88	10.89	10.70
Temperatures, deg. fahr.:													
Air to preheater	92	95	110	100	108	109	111	108	109	102	107	110	99
Air leaving preheater	281	293	304	320	301	357	355	375	386	387	384	397	400
Primary air	112	111	119	112	122	117	122	116	123	118	132	134	124
Secondary air	258	275	285	304	302	330	328	348	361	364	368	379	377
Air to drier	229	253	245	253	254	249	250	256	251	248	250	259	256
Air leaving drier	118	135	136	130	140	132	120	136	130	137	126	131	123
Gases leaving boiler	910	977	972	1064	1081	1154	1138	1213	1241	1269	1276	1413	1361
Gases leaving economizer	357	369	385	428	414	463	453	494	522	530	521	545	568
Gases leaving preheater	187	203	218	226	221	257	257	274	290	290	280	290	293
Feedwater to economizer	186	182	177	186	180	184	184	183	185	188	180	194	177
Superheated steam	652	672	676	693	680	713	707	728	727	735	744	727	723
Pressure, superheated steam, lb. per sq. in. gage	328	326	324	320	325	324	325	324	326	326	326	328	326
Pressures, air, in. of water:													
Primary-fan discharge	11.95	11.99	12.03	12.04	11.98	11.99	11.89	11.76	11.53	11.73	11.48	11.48	11.50
Burners in service	4.05	4.42	4.67	5.93	5.95	5.55	5.00	6.74	5.00	4.92	5.69	5.78	5.68
Secondary forced-draft-fan discharge	2.18	1.83	2.17	2.92	3.16	2.70	2.91	3.34	3.96	3.80	4.85	+5.0	+5.0
Main duct	1.34	0.87	0.99	1.54	1.74	1.07	1.13	1.64	1.42	1.30	1.92	2.41	2.73
Burners in service	0.24	0.32	0.80	1.35	1.36	0.81	1.17	1.34	1.40	1.41	2.09	2.49	2.77
Drafts, in. of water:													
Furnace	0.11	0.10	0.10	0.11	0.11	0.10	0.10	0.11	0.10	0.09	0.09	0.08	0.15
Uptake	0.40	0.41	0.46	0.63	0.64	0.76	0.85	1.00	1.38	1.68	1.68	1.89	2.33
Economizer out	1.65	1.72	2.03	2.66	2.88	3.64	4.00	4.77	6.45	6.55	7.71	8.64	10.30
Preheater out	1.76	1.82	2.24	3.15	3.22	4.03	4.53	5.48	7.66	7.77	9.30	10.49	12.34
Induced-draft fan inlet (after washer)	1.99	2.06	2.48	3.29	3.49	4.30	4.88	5.89	8.08	8.31	10.87	12.23	13.44
Analysis of gases:													
CO_2 boiler exit	14.22	15.97	14.59	15.16	15.10	15.72	14.68	16.32	14.75	15.41	15.11	15.87	16.77
CO_2 economizer exit	13.80	15.12	14.23	14.45	14.24	14.52	14.04	14.57	14.46	14.76	14.30	14.74	15.93
CO_2 preheater exit	12.89	14.00	13.35	13.46	13.33	13.99	13.24	13.90	12.34	13.56	13.68	14.06	14.64
O_2 preheater exit	6.49	5.01	5.95	5.65	5.97	5.20	6.18	5.05	7.24	5.54	5.51	5.20	4.30
CO preheater exit	0	0	0	0	0	0	0	0	0	0	0	0	0.09
Ashpit analysis:													
Ash, per cent	98.25	98.73	99.21	99.30	98.99	99.01	99.38	99.01	99.18	99.17	98.83	99.30	99.06
B.t.u. per lb.	254	184	115	102	145	144	90	144	119	120	170	102	136
Flue dust analysis:													
Ash, per cent	94.76	94.01	96.24	95.27	95.86	93.03	96.16	95.75	95.07	95.53	94.64	94.50	90.17
B.t.u. per lb.	760	869	545	686	600	1011	558	616	715	648	777	798	1425
Composite (ashpit 40 per cent, flue dust 60%):													
Ash, per cent	96.15	96.90	97.43	97.00	96.45	95.42	97.50	97.06	96.67	96.99	96.32	96.42	93.73
B.t.u. per lb.	548	595	373	435	515	664	362	427	483	437	533	519	909
Heat balance, per cent:													
Absorbed by boiler, economizer and superheater	88.20	88.50	88.80	89.50	88.10	88.00	89.20	87.00	89.70	87.10	87.60	87.50	85.90
Loss in dry flue gases	3.08	3.10	3.17	3.77	3.30	4.12	4.18	4.63	5.24	5.01	4.91	4.56	5.06
Loss due to preheated air for drier	1.70	2.34	1.95	1.94	1.79	1.53	1.47	1.40	1.24	1.14	1.14	0.82	0.96
Loss due to moisture in coal	0.22	0.19	0.25	0.20	0.23	0.21	0.23	0.20	0.23	0.23	0.23	0.24	0.23
Loss due to H_2 in coal	3.74	3.76	3.72	3.80	3.74	3.82	3.75	3.85	3.85	3.85	3.86	3.83	3.92
Loss due to unburned carbon	0.68	0.72	0.42	0.57	0.60	0.82	0.42	0.52	0.58	0.51	0.66	0.63	1.13
Loss due to carbon in CO	....	....	....	....	....	....	....	....	....	....	....	....	0.35
Unaccounted for losses	2.38	1.44	1.69	0.22	2.24	1.50	0.75	2.40	−0.84	2.16	1.60	2.42	2.45
Total	100	100	100	100	100	100	100	100	100	100	100	100	100
Heat transferred by preheater, per cent	4.25	3.78	4.12	4.93	4.88	5.10	4.90	6.02	5.63	5.66	6.02	6.43	6.03
Heat returned to furnace, per cent	1.85	1.45	2.17	2.16	2.28	3.58	3.43	4.62	4.39	4.52	4.90	5.60	4.37
Boiler auxiliary power, kw-hr. per ton of dry coal:													
Feeders	0.39	0.30	0.31	0.28	0.29	0.28	0.28	0.24	0.27	0.26	0.27	0.26	
Primary air	2.90	2.64	2.52	2.30	2.20	2.30	2.06	1.97	1.80	1.72	1.53	1.41	
Induced-draft fan	23.27	20.10	21.45	17.85	17.77	25.20	24.20	23.30	22.84	25.10	26.70	25.50	
Forced-draft fans	9.29	8.00	7.81	6.74	6.64	7.70	5.39	9.47	7.38	7.12	8.41	7.80	
Total	35.85	31.04	32.09	27.17	26.90	35.48	31.93	34.98	32.29	34.20	36.91	34.97	45.20
Coal preparation auxiliary power, kw-hr. per ton of dry coal:[1]													
Pulverizer	17.33	17.33	17.33	17.33	17.33	17.33	17.33	17.33	17.33	17.33	17.33	17.33	17.33
Exhauster	6.57	6.57	6.57	6.57	6.57	6.57	6.57	6.57	6.57	6.57	6.57	6.57	6.57
Fuller-Kinyon pump	1.08	1.08	1.08	1.08	1.08	1.08	1.08	1.08	1.08	1.08	1.08	1.08	1.08
Total	24.98	24.98	24.98	24.98	24.98	24.98	24.98	24.98	24.98	24.98	24.98	24.98	24.98
Total auxiliary power, kw-hr. per ton of dry coal	60.83	56.02	57.07	52.15	51.88	60.46	56.91	59.96	57.27	59.18	61.89	59.95	70.18
Heat equivalent of auxiliary power, B.t.u. per lb. coal as fired	504	467	471	433	430	503	471	499	475	490	513	497	582
Heat equivalent of auxiliary power, per cent	4.17	3.82	3.87	3.61	3.54	4.14	3.87	4.12	3.91	4.02	4.14	4.11	4.80
Overall efficiency, per cent	84.50	85.10	85.30	86.30	85.00	84.35	85.70	83.50	86.20	83.50	84.00	83.90	81.75

[1] Prorated figures based on total figures of eight months' operation.

TABLE 7 EVAPORATIVE TESTS ON BOILER NO. 22, CALUMET STATION, WITH A BIN SYSTEM OF FIRING, USING YOUGHIOGHENY COAL

Test number	1	2	3	4	5	6	7	8	9	10
Date of test	Aug. 2	Aug. 7	Aug. 9	Aug 14	Aug. 16	Aug. 21	Aug. 23	Aug. 30	Sept. 5	Sept. 20
Duration	10	12	12	9	12	12	8	8	8	7
Coal as fired										
Moisture, per cent	2.52	2.63	2.30	2.24	2.21	2.12	2.29	2.22	2.43	2.41
Ash, per cent	12.30	12.84	12.10	12.70	12.70	12.36	12.59	12.53	12.80	11.74
B.t.u. per lb.	12589	12426	12625	12601	12558	12651	12568	12563	12479	12693
Ultimate fuel:										
Carbon	83.25	83.26	83.51	83.27	83.23	83.28	83.31	83.30	83.22	83.37
Hydrogen	5.17	5.17	5.18	5.17	5.17	5.17	5.17	5.17	5.17	5.18
Sulphur	2.48	2.47	2.19	2.46	2.50	2.45	2.41	2.42	2.52	2.34
Nitrogen	1.63	1.63	1.63	1.63	1.63	1.63	1.63	1.63	1.63	1.63
Oxygen	7.47	7.47	7.49	7.47	7.47	7.47	7.48	7.48	7.46	7.48
Size of coal (Tyler screens) per cent:										
Through 28-mesh	99.9	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0
Through 48-mesh	98.3	98.4	98.7	99.0	99.0	99.1	98.5	98.7	99.0	98.5
Through 100-mesh	82.6	93.8	86.2	87.3	87.8	87.6	85.8	85.0	86.5	85.1
Through 150-mesh	75.0	74.8	72.8	79.1	79.6	79.8	77.1	76.3	78.6	76.0
Through 200-mesh	60.0	64.7	68.6	68.5	66.9	64.2	63.8	65.8	69.5	66.5
Feeders in service	1, 2, 3 & 4	1, 2, 3 & 4	1, 2, 3 & 4	2 & 3	1, 2, 3 & 4	2 & 3	1, 2, 3 & 4	1, 2, 3 & 4	1, 2, 3 & 4	1, 2, 3 & 4
Firing control	Auto.	Auto.	Auto.	Auto.	Auto.	Man.	Auto.	Auto.	Auto.	Man.
Coal burned per hour, lb.	16567	19864	16056	11977	16133	9198	16331	16255	23679	
B.t.u. liberated per cu. ft. furnace, vol. per hr.	16800	19900	16330	12170	16350	9380	16540	16470	23850	
Water evaporated per hr., lb.	153,631	180,124	148,061	116,042	149,025	87,666	151,539	150,878	212,107	247,805
Actual evap. per lb. coal per hr., lb.	9.27	9.07	9.23	9.68	9.24	9.53	9.28	9.28	8.96	9.50
Total equiv. evap. per lb. coal per hr., lb.	11.49	11.28	11.37	11.67	11.56	11.38	11.47	11.45	11.30	11.84
Temperatures, deg. fahr.:										
Air to preheater	111	109	108	93	113	86	98	80	90	91
Air leaving preheater	338	342	324	277	326	259	326	313	368	377
Primary air	113	111	111	112	113	103	107	102	112	107
Secondary air	296	321	306	257	304	235	306	394	342	350
Air to dryer	200	202	191	195	193	194	188	201	188	194
Air leaving dryer	126	124	124	125	126	121	124	120	123	126
Gases leaving boiler	1021	1034	984	838	945	753	975	955	1122	1151
Gases leaving economizer	441	452	411	347	405	305	412	401	509	530
Gases leaving preheater	252	260	239	197	239	187	238	221	289	293
Feedwater to economizer	188	189	190	188	190	187	189	189	187	190
Superheated steam	688	699	682	630	676	610	686	682	727	707
Pressure-superheated steam, lb. per sq. in. abs.	341	338	337	341	340	337	339	341	337	339
Pressures, air, in. of water:										
Primary-fan discharge	11.54	11.40	11.50	11.65	11.55	11.76	11.58	11.70	11.54	11.51
Burners in service	4.00	4.90	3.95	5.00	3.90	3.45	4.08	4.01	6.02	6.00
Secondary forced-draft-fan discharge	2.92	3.30	2.40	1.73	2.10	0.60	2.65	2.64	4.06	3.96
Main duct	1.83	1.80	1.40	1.07	1.13	0.14	1.56	1.66	1.95	1.54
Burners in service	0.26	0.59	0.28	0.26	0.12	−0.15	0.23	0.20	1.15	1.53
Drafts:										
Furnace	0.07	0.08	0.08	0.07	0.08	0.09	0.08	0.08	0.08	1.0
Uptake	0.70	1.02	0.70	0.40	0.65	0.32	0.75	0.70	1.54	1.85
Economizer out	3.06	4.40	3.00	1.60	2.76	1.10	3.30	3.08	6.78	7.95
Preheater out	3.94	5.70	3.70	1.90	3.50	1.17	4.15	3.89	8.85	10.12
Induced-draft fan inlet (after washer)	4.33	6.20	4.10	2.05	3.83	1.30	4.56	4.22	9.62	11.00
Analysis of gases, per cent										
CO_2 boiler exit	15.17	14.92	14.68	14.42	14.83	14.17	14.20	14.59	14.93	15.50
CO_2 economizer exit	14.02	13.24	13.66	14.32	14.55	13.95	13.80	13.93	13.90	14.25
CO_2 preheater exit	13.33	12.46	13.01	13.42	13.61	12.76	13.02	12.89	12.90	13.92
O_2 preheater exit	6.16	6.59	6.12	5.85	5.74	6.53	6.28	6.35	6.28	4.85
CO preheater exit										
Ashpit analysis:										
Ash, per cent	98.42	98.75	98.64	97.90	98.03	98.38	99.30	98.62	94.92	98.01
B.t.u. per lb.	229.0	181.0	197.0	305.0	286.0	235.0	102.0	200.0	737.0	289.0
Flue-dust analysis:										
Ash, per cent	97.3	93.65	96.05	94.76	95.12	88.1	94.37	91.08	93.81	93.17
B.t.u. per lb.	384.0	921.0	573.0	760.0	708.0	1717.0	816.0	1293.0	898.0	990.0
Composite (ashpit 40%, flue dust 60%):										
Ash, per cent	97.75	95.69	97.28	96.02	96.28	92.25	96.34	94.10	94.25	95.11
B.t.u. per lb.	322.0	625.0	423.0	578.0	539.0	1124.0	531.0	856.0	834.0	710.0
Heat balance										
Absorbed by boiler, economizer and superheater	88.90	88.20	87.70	90.00	88.00	87.30	88.75	88.60	87.90	
Loss in dry flue gases	3.74	4.35	3.61	2.65	3.39	2.60	3.74	3.52	5.20	4.99
Loss due to preheated air for drier	0.76	0.67	0.74	1.11	0.74	1.21	0.69	0.64	0.55	0.54
Loss due to moisture in coal	0.22	0.23	0.20	0.19	0.19	0.18	0.20	0.20	0.22	0.22
Loss due to H_2 in coal	3.47	3.52	3.49	3.42	3.47	3.42	3.50	3.50	3.60	3.61
Loss due to unburned carbon	0.32	0.67	0.42	0.61	0.57	1.19	0.55	0.90	0.91	0.69
Loss due to carbon in CO										
Unaccounted for losses	2.59	2.36	3.84	2.02	3.64	4.10	2.57	2.64	1.62	
Total	....	....	....	....	....	....	....	....	....	....
Heat transferred by preheater, per cent	4.67	5.02	4.37	3.67	3.90	2.74	4.35	4.39	5.37	
Heat returned to furnace, per cent	3.94	4.35	3.64	2.46	3.16	1.53	3.58	3.75	4.82	
Boiler auxiliary power, kw-hr. per ton dry coal:										
Feeders	0.37	0.32	0.39	0.40	0.38	0.50	0.37	0.37	0.28	0.27
Primary air	2.62	2.29	2.73	3.03	2.68	3.80	2.71	2.72	2.03	1.78
Induced-draft fan	18.70	29.10	19.10	23.60	18.90	27.70	18.76	19.16	28.00	25.00
Forced-draft fan	6.58	8.10	6.75	9.20	6.73	10.93	6.72	6.99	9.32	8.48
Total	28.27	39.81	28.97	36.23	28.69	42.93	28.56	29.24	39.63	35.53
Coal preparation, auxiliary power, kw-hr. per ton dry coal:[1]										
Pulverizer	12.90	12.90	12.90	12.90	12.90	12.90	12.90	12.90	12.90	12.90
Exhauster	4.80	4.80	4.80	4.80	4.80	4.80	4.80	4.80	4.80	4.80
Fuller Kinyon pump	0.74	0.74	0.74	0.74	0.74	0.74	0.74	0.74	0.74	0.74
Total	18.44	18.44	18.44	18.44	18.44	18.44	18.44	18.44	18.44	18.44
Total auxiliary power, kw-hr. per ton dry coal	46.71	58.25	47.41	54.67	47.13	61.37	47.00	47.68	58.07	53.97
Heat equivalent of auxiliary power B.t.u. per lb. coal as fired	387	482	394	453	392	512	391	397	481	448
Heat equivalent of auxiliary power, per cent	3.08	3.88	3.12	3.60	3.12	4.04	3.11	3.16	3.85	3.53
Net efficiency, per cent	86.20	84.70	84.90	86.80	85.30	83.80	86.10	85.80	84.50	

[1] Prorated figures based on total figures of two months' operation.

TABLE 8 EVAPORATIVE TESTS ON BOILER NO. 22, CALUMET STATION, WITH A BIN SYSTEM OF FIRING, USING CENTRAL ILLINOIS COAL

Test number	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
Date of test	Mar. 27	Mar. 28	Mar. 29	Mar. 30	Apr. 4	Apr. 5	Apr. 6	Apr. 11	Apr. 12	Apr. 13	Apr. 16	Apr. 17	May 16	May 17	May 21	May 23	May 25	May 28	June 1	June 29	July 3
Duration, hours	7	7	7	7	7	7	5	7	7	7	7	3½	5	7	12	11	11	12	7	12	7
Coal as fired:																					
Moisture, per cent	8.11	8.49	8.36	8.26	8.75	9.07	8.23	8.08	8.37	8.48	8.01	9.11	8.83	9.33	8.30	8.65	8.12	7.68	8.60	8.85	9.72
Ash, per cent	15.39	14.51	14.61	15.39	14.47	15.00	14.10	13.31	13.64	12.30	13.27	13.28	13.46	13.58	15.06	14.31	14.77	12.83	14.83	14.47	14.98
B.t.u. per lb.	10,761	10,818	10,803	10,744	10,787	10,692	10,926	11,048	10,973	11,160	11,088	10,919	10,933	10,866	10,764	10,853	10,868	11,210	10,795	10,788	10,586
Ultimate fuel analysis, per cent:																					
Carbon	75.0	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Hydrogen	5.5	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Sulphur	5.3	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Nitrogen	1.2	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Oxygen	13.0	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Size of coal (Tyler screens), per cent:																					
Through 28-mesh	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100
Through 48-mesh	98.5	97.5	98.8	99.1	98.8	98.7	98.7	99.5	99.4	99.5	99.2	99.5	98.9	99.2	99.0	99.0	99.0	99.3	98.7	99.0	99.2
Through 100-mesh	88.1	87.5	88.8	90.1	89.3	86.6	88.6	82.2	90.2	85.7	80.0	82.2	87.2	85.8	89.2	88.0	88.1	90.0	87.9	84.3	87.5
Through 150-mesh	81.4	80.4	82.0	83.7	82.5	78.2	81.8	82.0	82.9	74.5	82.6	80.4	79.7	78.0	82.0	80.7	80.8	82.4	80.0	75.2	79.5
Through 200-mesh	72.3	69.8	73.2	75.2	75.4	65.5	68.7	72.2	72.0	60.6	68.0	70.1	67.7	60.8	70.4	70.3	69.4	72.7	69.3	65.5	67.5
Feeders in service	2 & 3	2 & 3	2 & 3	1, 2 & 3	1, 2, 3 & 4	2 & 3	1, 2, 3 & 4	1, 2, 3 & 4	1, 2, 3 & 4	2, 3 & 4	1, 2, 3 & 4	1, 2, 3 & 4	1, 2, 3 & 4	2 & 3	1, 2, 3 & 4	1, 2, 3 & 4	1, 2, 3 & 4	1, 2, 3 & 4	1, 2, 3 & 4	2 & 3	1, 2, 3 & 4
Firing control	A	A	A & M	A & M	M	M	M	A	A	A	M	A	A	M	M	A	A	A	M	M	M
Coal burned per hour, lb.	.....	.....	.....	.....	.....	.....	32,313	.....	.....	.....	28,741	27,675	15,576	13,274	18,191	22,195	25,999	15,762	29,743	10,086	31,764
B.t.u. liberated per cu. ft. furnace, vol. per hr.	.....	.....	.....	.....	.....	.....	28,200	.....	.....	.....	25,500	24,200	13,600	11,600	15,750	19,400	22,700	14,200	25,850	8,700	26,650
Water evaporated per hour, lb.	105,759	122,441	142,286	160,692	181,849	90,341	251,312	198,877	142,470	103,999	231,152	218,310	126,735	105,590	140,400	172,792	199,395	128,030	220,291	80,012	239,074
Actual evaporated per lb. coal per hr., lb.	.....	.....	.....	.....	.....	.....	7.78	.....	.....	.....	8.05	7.89	8.13	7.93	7.72	7.78	7.67	8.12	7.42	7.93	7.53
Total equivalent evaporation per lb. coal per hr., lb.	.....	.....	.....	.....	.....	.....	9.40	.....	.....	.....	9.77	9.37	9.83	9.57	9.56	9.64	9.48	9.90	9.15	9.42	9.40
Temperatures, deg. fahr.:																					
Air to preheater	98	97	97	99	109	101	120	103	98	99	101	94	106	85	92	77	101	92	103	103	103
Air leaving preheater	272	303	315	342	371	282	426	376	324	294	402	373	.....	294	354	366	397	314	400	233	378
Primary air	110	116	120	105	110	105	117	112	95	108	104	105	103	95	99	120	118	98	103	92	103
Secondary air	252	286	304	335	351	256	402	320	305	264	369	341	298	271	324	328	359	293	370	203	357
Air to dryer	233	240	226	233	236	227	251	243	255	227	267	260	217	213	233	243	235	230	225	176	221
Air leaving drier	109	121	128	115	127	118	128	124	120	111	126	120	120	121	.....	121	1224	122	123	136	122
Gases leaving boiler	843	935	1001	1005	1076	762	1256	1134	1007	851	1251	1088	....	....	1067	1152	1201	988	1270	784	1182
Gases leaving economizer	334	373	412	440	465	325	583	499	401	340	547	490	379	340	455	488	544	410	549	298	541
Gases leaving preheater	180	222	232	255	286	190	384	283	243	224	324	286	227	174	254	257	297	230	319	160	297
Feedwater to economizer	183	184	183	182	192	180	194	187	182	181	194	185	195	185	187	192	194	189	189	187	191
Superheated steam	591	617	646	638	655	589	649	670	634	600	642	593	648	627	686	701	695	657	685	599	709
Pressure, superheated steam, lb. per sq. in. gage	327	328	328	328	325	322	330	333	329	326	328	327	327	325	341	330	330	325	327	327	328
Pressures, air, in. of water:																					
Primary-fan discharge	12.26	12.18	12.90	12.00	11.64	12.07	11.44	11.56	12.00	12.14	12.00	12.00	12.00	12.10	12.00	11.72	11.58	12.00	11.59	12.02	11.34
Burners in service	5.74	6.49	7.13	6.11	5.88	5.00	7.31	5.91	4.05	4.00	5.82	5.93	4.00	4.82	3.99	5.44	6.00	3.51	6.50	3.05	7.00
Secondary, forced draft fan discharge	1.75	2.19	3.10	2.94	2.13	0.89	3.98	3.18	2.20	1.39	3.32	2.55	1.43	1.06	2.00	2.81	3.58	1.86	3.27	0.88	3.95
Main duct	0.94	1.28	1.74	1.68	0.76	0.28	1.65	1.50	1.13	0.71	1.24	0.78	0.61	0.36	0.79	1.11	1.60	1.15	1.24	0.38	1.57
Burners in service	0.12	0.49	1.15	0.64	0.40	0.00	1.37	0.63	1.05	—0.12	1.02	0.69	.....	0.00	0.17	0.50	0.92	0.00	1.20	0.18	1.38
Drafts, in. of water:																					
Furnace	0.10	0.10	0.11	0.11	0.11	0.10	0.09	0.12	0.11	0.10	0.09	0.14	0.12	0.10	0.12	0.13	0.12	0.08	0.08	0.09	0.10
Uptake	0.30	0.39	0.52	0.65	0.73	0.30	1.51	1.00	0.50	0.33	1.30	1.89	0.60	0.40	0.79	1.18	1.50	0.50	1.99	0.25	1.78
Economizer out	1.39	1.94	2.52	3.28	3.78	1.19	7.60	5.06	2.64	0.14	6.60	6.94	2.34	1.60	3.24	4.95	6.32	2.50	2.53	0.98	7.30
Preheater out	1.40	1.94	.....	3.46	4.10	.....	9.03	5.66	2.82	0.14	7.74	8.01	2.49	1.63	3.63	6.13	7.87	2.86	9.67	0.99	9.63
Induced draft fan inlet (after washer)	1.68	2.42	3.26	4.20	4.86	1.43	10.31	6.74	3.45	1.77	9.05	9.03	3.00	1.93	4.18	6.56	8.78	3.32	10.75	1.11	10.81
Analysis of gases, per cent:																					
CO_2 boiler exit	14.47	14.69	14.87	15.04	15.71	13.58	16.30	15.68	14.87	14.24	16.55	15.30	14.15	14.84	13.28	13.25	13.73	14.11	15.76	14.48	16.50
CO_2 economizer exit	14.27	14.40	14.24	14.26	15.30	13.68	15.35	14.66	14.29	14.10	14.82	14.61	.....	13.50	13.03	13.12	13.59	13.45	13.61	13.82	15.45
CO_2 preheater exit	13.11	13.08	13.51	13.69	14.14	12.85	14.40	13.92	13.35	12.76	14.37	13.47	12.93	13.09	12.35	12.32	12.71	12.42	13.16	12.16	14.35
O_2 preheater exit	6.00	6.01	5.58	5.35	4.77	6.25	3.90	5.14	5.78	6.33	4.61	5.13	6.23	6.03	6.90	6.82	6.40	6.73	5.91	6.92	4.56
CO preheater exit	0.00	0.00	0.00	0.00	0.00	0.00	0.40	0.00	0.00	0.00	0.00	0.30	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.11
Ash pit analysis:																					
Ash, per cent	99.5	99.2	98.8	98.1	97.5	97.8	97.5	98.4	97.3	98.1	96.3	97.7	98.7	97.1	99.5	99.4	99.2	98.7	99.3	98.2	98.8
B.t.u. per lb.	72.0	122	165	276	368	318	357	220	386	274	538	339	193	421	77	87	115	186	109	262	173
Flue dust analysis:																					
Ash, per cent	95.14	97.41	97.21	98.81	97.65	96.39	95.37	95.82	97.06	97.32	95.62	95.19	92.40	96.48	98.92	99.27	98.14	99.00	97.80	93.48	94.55
B.t.u. per lb.	705	367	405	173	341	623	671	606	426	389	635	697	1102	510	157	103	270	145	319	945	790
Composite (ash pit 40%, flue dust 60%)																					
Ash, per cent	96.88	98.14	97.87	98.55	97.58	96.86	96.24	96.88	97.17	97.64	95.89	96.18	94.91	96.80	99.14	99.33	98.57	98.89	98.38	95.35	96.20
B.t.u. per lb.	452	269	309	214	350	441	555	452	410	343	596	555	738	474	125	96.6	208	161	243	672	543

TABLE 8 EVAPORATIVE TESTS ON BOILER NO. 22, CALUMET STATION, WITH A BIN SYSTEM OF FIRING, USING CENTRAL ILLINOIS COAL (*Continued*)

Heat balance, per cent:																								
Absorbed by boiler, economizer, and superheater	.....	.....	.....	.....	.....	83.80	.....	.....	85.20	83.44	87.40	86.70	86.32	86.60	84.78	85.90	82.50	85.00	83.83					
Loss in dry flue gases	3.15	3.64	3.76	4.41	4.48	6.21	4.72	3.99	5.44	5.57	3.39	2.17	4.42	4.68	6.40	3.88	6.47	2.98	4.49					
Loss due to preheated air for drier	2.05	1.52	1.51	1.29	1.12	1.43	0.77	1.11	1.38	1.34	1.09	1.34	1.60	1.32	1.15	0.91	1.43	0.70	1.47	0.61				
Loss due to moisture in coal	0.84	0.91	0.87	0.90	0.92	0.98	0.89	0.84	0.87	0.85	0.84	0.95	0.89	0.93	0.87	0.90	0.86	0.76	0.91	0.89	0.82			
Loss due to H_2 in coal	3.92	4.06	3.97	4.13	3.95	4.06	4.13	4.02	3.97	3.92	4.08	4.04	3.89	3.81	3.96	3.95	4.02	3.91	4.10	3.82	3.63			
Loss due to unburned carbon	0.67	0.36	0.42	0.33	0.48	0.62	0.74	0.66	0.51	0.39	0.74	0.96	0.61	0.18	0.14	0.29	0.19	0.34	0.95	0.73				
Loss due to carbon in CO	.....	.....	.....	.....	.....	.....	1.45	.....	.....	.....	0.00	1.16	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.42				
Unaccounted for losses	.....	.....	.....	.....	.....	.....	2.01	.....	.....	.....	2.64	3.05	2.13	5.18	2.93	2.56	3.74	3.93	5.98	6.47				
Total	.....	.....	.....	.....	.....	.....	100.00	.....	.....	.....	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00				
Heat transferred by preheater, per cent	2.99	3.16	4.25	4.22	3.80	2.32	4.16	4.94	3.68	2.52	5.15	4.55	.....	5.22	5.88	6.25	4.32	5.93	3.16	6.91				
Heat returned to furnace, per cent	0.95	1.65	2.74	2.94	2.67	0.90	3.40	3.83	2.20	1.18	4.09	3.66	.....	2.21	3.61	4.73	5.33	2.67	6.22	1.67	6.28			
Boiler auxiliary power, kw-hr. per ton of dry coal:																								
Feeders	.....	.....	.....	.....	.....	.....	0.29	.....	.....	0.32	0.32	0.49	0.37	0.36	0.33	0.28	0.40	.....	0.48	0.36				
Pulverizer	.....	.....	.....	.....	.....	.....	1.66	.....	.....	1.71	1.78	3.13	3.25	2.60	2.28	1.91	2.82	1.70	3.78	1.66				
Exhauster	.....	.....	.....	.....	.....	.....	21.22	.....	.....	22.00	24.78	20.40	21.98	17.86	28.60	25.35	20.03	22.70	27.70	22.30				
Fuller Kinyon pump	.....	.....	.....	.....	.....	.....	7.22	.....	.....	6.50	6.61	7.58	6.56	6.54	8.35	7.25	7.52	6.12	11.20	7.29				
Forced draft fans	.....	.....	.....	.....	.....	.....	30.29	.....	.....	30.63	33.49	31.60	34.16	27.36	39.56	34.79	30.77	30.52	43.16	31.62				
Total	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....				
Coal preparation auxiliary power, kw-hr. per ton dry coal[1]																								
Pulverizer	.....	.....	.....	.....	.....	.....	17.67	.....	.....	17.67	17.67	17.67	17.67	17.67	17.67	17.67	17.67	17.67	17.67	17.67				
Exhauster	.....	.....	.....	.....	.....	.....	7.21	.....	.....	7.21	7.21	7.21	7.21	7.21	7.21	7.21	7.21	7.21	7.21	7.21				
Fuller Kinyon pump	.....	.....	.....	.....	.....	.....	0.68	.....	.....	0.68	0.68	0.68	0.68	0.68	0.68	0.68	0.68	0.68	0.68	0.68				
Total	.....	.....	.....	.....	.....	.....	25.56	.....	.....	25.56	25.56	25.56	25.56	25.56	25.56	25.56	25.56	25.56	25.56	25.56				
Total auxiliary power, kw-hr. per ton of dry coal	.....	.....	.....	.....	.....	.....	55.85	.....	.....	56.09	58.73	57.16	59.72	52.92	65.12	60.35	56.33	56.08	68.76	57.26				
Heat equivalent of auxiliary power, B.t.u. per lb. coal as fired	.....	.....	.....	.....	.....	.....	436.0	.....	.....	438.0	455.0	443.0	461.0	412.0	505.0	472.0	442.0	436.0	532.0	449.0				
Heat equivalent of auxiliary power, per cent	.....	.....	.....	.....	.....	.....	3.98	.....	.....	3.96	4.16	4.05	4.25	3.83	4.66	4.33	3.94	4.03	4.93	4.16				
Overall efficiency, per cent	.....	.....	.....	.....	.....	.....	80.25	.....	.....	82.00	80.00	83.80	82.10	82.90	82.50	81.20	82.60	79.20	80.80	80.04				

[1] Prorated figures based on total figures of three months' operation.

in the experience with this system. While this situation was in no sense important enough to condemn, these inherent factors of unit operation must surely be recognized in making further refinements.

Operation was at first attempted with a table mill having a single set of 11 15-in. balls with no pressure springs for holding the balls to a fixed course. Although successful operation of this type of mill has been obtained in the smaller sizes, it failed in this case to give the desired capacity and proper sizing. The mill was reconverted to a five-ball pusher type, using the same motor and reduction gearing. The operating performance over the period shown is based on this mill as thus changed.

The high maintenance costs of this mill were due to the same causes as experienced in the operation of the original screen mill—viz., breakage of pushers, scoopers, ball race, pusher arms, and renewal of balls due to wearing egg-shaped. The wearing of the various mill parts was much more severe, due to the necessity of operating at a variable capacity on the unit system rather than at a fixed capacity as on the bin system. In other words, this particular mill as altered for the unit system was not suitable when running at the speeds at which it was necessary to run to give the desired capacities.

The experience thus far in the mill problem still leaves much to be desired. At the present time further investigations are being carried out with an improved type of table mill and a tube mill, on which data are not yet available.

With this background and the difficulties of coordinating mill speed, primary air volume, and sizing, the idea of a relatively slow constant-speed mill with primary-air volume interlocked with coal feed begins to have appealing aspects, not only from the standpoint of maintenance and regulation, but also from that of automatic control.

Among the indirect effects of unit-system operation may be mentioned those in connection with the superheater. In the starting-up procedure it is difficult to hold down the coal feed to three burners and retain ignition due to requirements at the mill in the way of primary air volume. As practically all of the air at this time is primary air, long flaming results, causing excessive superheater-tube temperatures, with resultant sagging and damage to the supporting structure. This situation was improved by increased venting of the superheater boxes, although the resultant losses were not desirable.

Some improvement in this respect was obtained incidental to changes in the burners. Unit-system operation was started, using the same burners as were used with the former bin system. The greater primary-air volume requirements and the resulting longer flame caused excessive slag deposits on the rear wall and lower boiler tubes. More satisfactory conditions were obtained by increasing the burner throat openings from 62 in. by 0.50 in. to 62 in. by 1.12 in. The primary-air requirements are normally about 33 per cent of the total air used in this system. With bin-system operation the primary-air requirements were about 20 per cent.

Test Performances

Test data are of value in determining the optimum possibilities of equipment and furnish a means of analyzing performance in detail. While many factors may enter into it, the best evaporative efficiency is the item given primary consideration, and for this reason the efficiency characteristics only of both systems using the various coals are graphically compared in Fig. 6. The net efficiencies shown are the actual evaporative efficiencies corrected for the equivalent of the total auxiliary power used. The complete data for each series of tests are given in Tables 6, 7, 8, and 9.

The evaporative efficiencies obtained with Central Illinois coal

TABLE 9 EVAPORATIVE TESTS ON BOILER NO. 22, CALUMET STATION, WITH A UNIT SYSTEM OF FIRING, USING CENTRAL ILLINOIS COAL

Test number	1	2	3	4	5	6	7	8
Date of test	June 27	July 17	Aug. 6	Aug. 7	Aug. 28	Aug. 29	Nov. 5	Nov. 6
Duration, hours	7	7	7	7	7	6	6	6½
Coal as fired:								
Moisture, per cent	15.31	14.04	14.37	14.23	14.12	14.26	13.69	13.32
Ash, per cent	13.27	12.81	12.12	13.51	14.53	11.94	13.77	15.13
B.t.u. per lb.	10383	10338	10390	10190	10063	10438	10173	10050
Ultimate fuel analysis, per cent:								
Carbon	75.0	75.0	75.0	75.0	75.0	75.0	75.0	75.0
Hydrogen	5.5	5.5	5.5	5.5	5.5	5.5	5.5	5.5
Sulphur	5.3	5.3	5.3	5.3	5.3	5.3	5.3	5.3
Nitrogen	1.2	1.2	1.2	1.2	1.2	1.2	1.2	1.2
Oxygen	13.0	13.0	13.0	13.0	13.0	13.0	13.0	13.0
Size of coal (Tyler screens), per cent:								
Through 28-mesh	100	100	100	100	100	100	100	100
Through 48-mesh	99.7	99.7	99.6	99.7	99.7	99.7	99.7	99.3
Through 100-mesh	91.8	93.0	92.2	93.7	91.1	93.0	92.7	87.5
Through 150-mesh	83.9	87.0	84.7	87.8	82.6	86.1	85.7	79.2
Through 200-mesh	74.0	77.8	75.8	78.9	71.8	76.5	76.2	69.2
Feeders in service	1	1	1	1	1	1	1	1
Firing control	Auto.	Hand	Auto.	Auto.	Auto.	Auto.	Hand	Hand
Coal burned per hour, lb.	20577	26816	20678	24188	17143	23895	14735	17146
Coal per hour, dry, tons	8.72	11.50	8.85	10.35	7.37	10.25	6.35	7.42
B.t.u. liberated per cu. ft. furnace, vol. per hr.	17150	22350	17340	19900	13940	20100	12080	13900
Water evaporated per hour, lb.	153,650	195,953	154,412	172,468	124,280	173,078	108,726	122,865
Actual evaporation per lb. coal per hr., lb.	7.45	7.30	7.47	7.14	7.25	7.24	7.37	7.16
Total equivalent evaporation per lb. coal per hr., lb.	9.22	9.11	9.23	8.89	8.88	9.12	8.97	8.80
Temperatures, deg. fahr.:								
Air to preheater	108	120	102	98	91	101	97	102
Air leaving preheater	388	428	391	408	369	426	349	371
Gases leaving boiler	1045	1142	1059	1080	978	1168	919	997
Gases leaving economizer	414	488	423	458	391	470	358	393
Gases leaving preheater	278	326	281	298	261	300	246	263
Feedwater to economizer	189	190	183	185	190	185	186	189
Superheated steam	680	704	670	690	657	714	643	664
Pressure-superheated steam, lb. per sq. in.	321	320	318	318	317	321	319	314
Pressures, air, in. of water:								
Secondary-forced draft fan discharge	1.80	4.00	1.56	2.30	1.06	2.88	0.74	1.42
Main duct								
Burners in service	0.90	2.30	0.82	1.23	0.56	1.60	0.26	0.60
Drafts, in. of water:								
Furnace	0.09	0.10	0.09	0.10	0.09	0.08	0.05	0.08
Uptake	1.00	1.27	0.80	1.14	0.50	1.10	0.40	0.50
Economizer out	3.00	5.54	3.00	3.90	2.18	4.30	1.45	2.10
Preheater out	3.50	6.65	3.50	4.50	2.30	5.00	1.70	2.23
Exhauster fan inlet	9.35	8.25	8.90	8.96	8.42	7.60	8.70	6.60
Analysis of gases, per cent:								
CO_2 boiler exit	15.25	16.10	15.20	15.80	14.50	15.35	14.40	14.40
CO_2 economizer exit	14.80	14.70	14.50	15.10	13.45	13.56	13.90	14.20
CO_2 preheater exit	13.55	12.65	13.60	13.85	12.85	13.46	11.70	11.70
O_2 preheater exit	5.40	6.75	5.65	5.25	6.60	5.86	7.70	7.50
CO preheater exit								
Ashpit analysis, per cent								
Ash	97.98	97.70	98.55	98.14	98.08	98.66	98.27	98.10
B.t.u. per lb.	293	334	210	270	278	194	251	276
Flue dust analysis, per cent:								
Ash	97.63	96.48	97.45	98.73	97.56	98.03	97.67	97.63
B.t.u. per lb.	344	519	370	183	353	286	345	343
Composite (ash pit 40%, flue dust 60%):								
Ash, per cent	97.75	96.85	97.89	98.55	97.77	98.28	97.74	97.82
B.t.u. per lb.	324	446	306	218	323	249	307	316
Heat balance, per cent:								
Absorbed by boiler, economizer, and superheater	86.0	85.40	86.20	84.60	85.50	84.60	85.70	84.81
Loss in dry flue gases	4.27	5.67	4.44	4.81	4.28	5.00	4.42	4.95
Loss due to moisture in coal	1.50	1.55	1.56	1.59	1.58	1.55	1.50	1.50
Loss due to H_2 in coal	4.09	3.98	3.95	4.00	3.93	3.99	3.95	4.00
Loss due to unburned carbon	0.42	0.50	0.36	0.29	0.48	0.29	0.42	0.48
Loss due to carbon in CO								
Unaccounted for losses	3.72	2.90	3.49	4.71	4.23	4.57	4.01	4.26
Total	100	100	100	100	100	100	100	100
Heat transferred by preheater, per cent	2.62	3.19	3.20	3.51	3.13	3.96	2.10	2.34
Boiler auxiliary power, kw-hr. per ton of dry coal:								
Feeders	0.27	0.21	0.26	0.23	0.23	0.23	0.25	0.22
Induced draft fan	30.00	26.70	17.00	27.00	19.40	28.00	20.90	19.00
Forced draft fans	6.15	8.72	5.90	7.42	7.05	7.72	7.92	6.98
Total	36.42	35.63	23.16	34.65	26.68	35.95	29.07	26.20
Coal preparation auxiliary power, kw-hr. per ton dry coal:								
Pulverizer	21.50	20.20	20.40	19.80	21.00	19.30	22.90	21.50
Exhauster	8.70	7.00	8.48	7.57	9.97	7.70	10.95	10.00
Total	30.20	27.20	28.88	27.37	30.97	27.00	33.85	31.50
Total auxiliary power, kw-hr. per ton of dry coal	66.62	62.83	52.04	62.02	57.65	62.95	62.92	57.70
Heat equivalent of auxiliary power, B.t.u. per lb. coal as fired	479	457	378	458	421	459	462	425
Heat equivalent of auxiliary power, per cent	4.61	4.42	3.64	4.50	4.18	4.40	4.54	4.22
Overall efficiency, per cent	82.00	81.60	83.00	80.80	81.90	80.90	81.80	81.20

are about 2½ per cent lower than those obtained with the better coals. This is due somewhat to the greater inherent moisture losses, higher excess air requirements for smokeless combustion, and the usual fouling of surfaces. These three conditions characterize this coal and are in effect a penalty.

It will, however, be noted that there is no appreciable difference between the evaporative efficiencies of the bin system and the unit system when firing Central Illinois coal. As this same conclusion is also verified by actual performance, it seems fair to conclude that in this respect there really is not much difference between these methods of firing. There seems to be a tendency to detract from the unit system for reasons based mostly on theoretical considerations, but in actual performances these differences do not seem to be very noticeable.

The high capacity of 290,000 lb. per hour obtained with Kentucky coal is not necessarily confined to this particular coal. It so happened that during this series of tests the gas washer was in operation. The lowered gas temperatures permitted of much higher induced-draft-fan capacities. All of the other test series were performed with the washer out of service. It is quite probable that with the washer in operation such a high capacity could have been obtained with Central Illinois coal for short periods. The limit of capacity obtained with unit-system operation is that imposed by the mill.

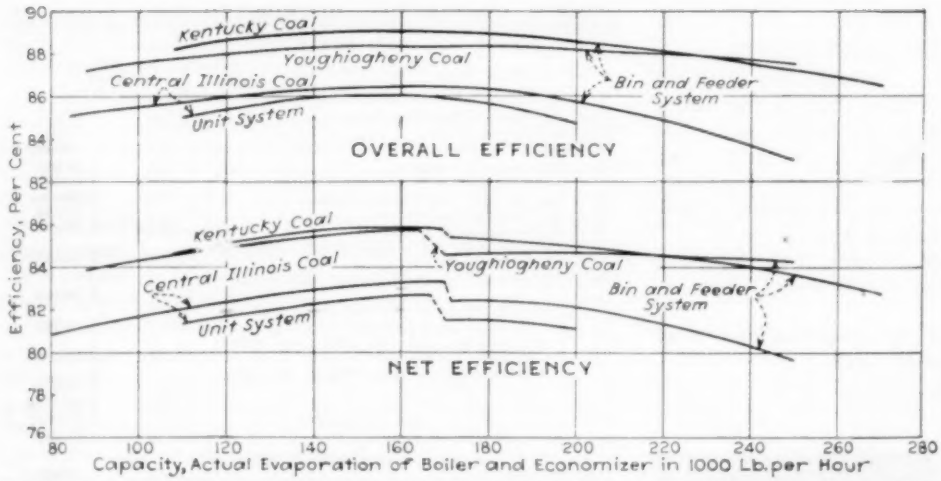


FIG. 6 COMPARATIVE EFFICIENCIES, BOILER NO. 22, CALUMET STATION, WITH DIFFERENT COALS, WITH BOTH BIN AND UNIT SYSTEMS OF FIRING

OPERATING PERFORMANCE

The actual overall result of a system of firing coal is probably the most important consideration in judging the commercial value of it. One cannot operate a system usually without having contingencies of some kind or other—routine outages for cleaning and repairs, occasional errors in operation, fluctuating load conditions, and a host of other things detract from the ideal possibilities. The difference between evaporative efficiency as determined by test and that of actual operation is somewhat a measure of all these as a whole. In this case these differences are about 2½ to 3 points for the bin system of firing with the coals used, which may be regarded as remarkably good practice with the conditions described. The actual performance of the unit system using Central Illinois coal compared to its test evaporative efficiency shows about the same result. We may therefore conclude that there is little if any real grounds to regard the unit system unfavorably on a commercial operating basis under the conditions as outlined.

On the other hand, the matter of reliability did favor the bin system in this case. Practically all emergency outages of the unit system were those caused by the mill. While multiplicity of units may be regarded as an assurance of continuity, it nevertheless follows that greater reliability of mills themselves is desirable.

The auxiliary power consumption of the unit system was lower, although the possibilities are not shown by the actual figures. The high energy consumption of milling was due to poorly adapted motor equipment with high rheostat losses, which could, in a more suitably designed combination, be avoided or at least reduced.

Automatic control is becoming more and more closely associated with the various combustion processes, as it undoubtedly should be. In this case, automatic control, while limited somewhat, was enough to demonstrate the advantages of steadier operation and with far less attention to detailed regulations than would be the case in manual operation. Far better pressure regulation is also possible. While the justification of the control system from an efficiency standpoint is not clearly demonstrated, nevertheless, for the other reasons, automatic control in some limited form with our future installations will be given quite serious consideration. It seems out of place to contemplate large steaming units with no other control than that of a man with divided attention, as is usually the case.

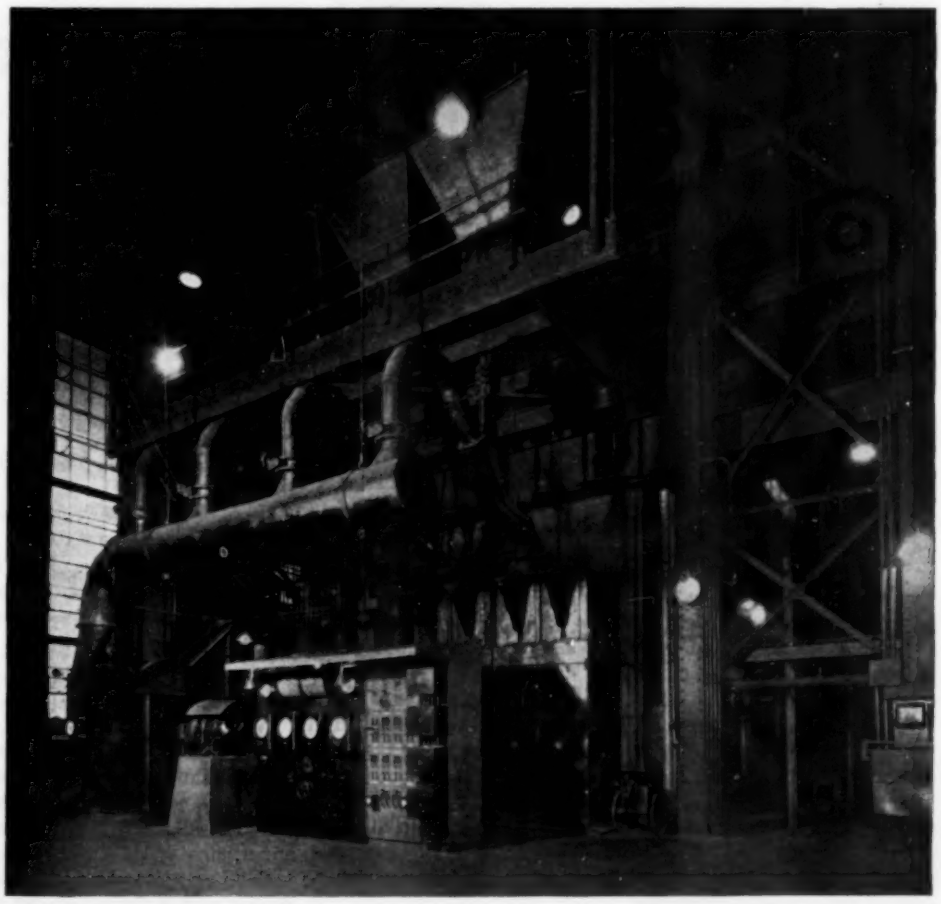


FIG. 7 CONTROL PANEL AND FRONT VIEW OF BOILER NO. 22, CALUMET STATION, WITH A BIN SYSTEM OF FIRING

Operating Costs

The operating costs of Tables 3 and 5 have been arranged on a more comparative basis in Table 10. The final cost unit selected was the cost of 1,000,000 B.t.u. input to the steam-header system, and in order to eliminate the effect of varying coal cost, a standard fuel-heat cost of $18 per 100,000,000 B.t.u. was used. Thus Table 10 shows actual costs with the exception of the fuel item. Inasmuch as these accounts may be somewhat influenced by local conditions, a more detailed explanation of them follows:

Operating labor accounts include the labor of preparing and firing coal, cleaning boiler and furnace, disposing of refuse, and water tending.

Maintenance accounts include the labor and material for repair and maintenance of the equipment listed.

Account 718-B (coal- and ash-conveying equipment maintenance) was divided, because a portion of this cost was prorated from maintenance charges on equipment common to all boiler units in the station.

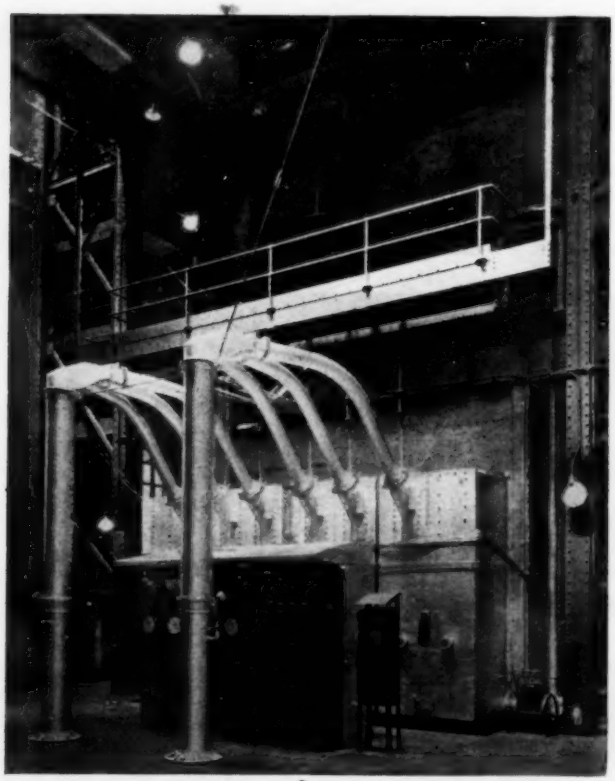


Fig. 8 Control Panel and Front View of Boiler No. 22, Calumet Station, With a Unit System of Firing

Account 718-C (crusher and breaker maintenance) was also prorated for the same reason.

Auxiliary power costs were taken as the station cost per kw-hr. at the bus, after the coal portion of this cost had been adjusted to allow for the advantage in efficiency of boiler No. 22 over that of the boiler room as a whole.

The costs shown in Table 10 are directly comparable between the two systems except as they may have been affected by the conditions explained in the notes. These costs do not include any expense of manufacturers as a result of being required to meet original guaranteed performance.

No attempt is made to draw any general conclusions from either these costs or the operating performance. As they apply to the specific problem of this report, the results should speak for themselves.

TABLE 10 COMPARATIVE DISTRIBUTED UNIT COSTS OF BOILER NO. 22, CALUMET STATION, BIN SYSTEM VERSUS UNIT SYSTEM

	Dollars per million B.t.u. delivered	
	Bin	Unit
703-A coal- and ash-handling labor	0.00095	0.00120
703-B boiler-room labor	0.00920[a]	0.00815
Total operating labor	0.01015	0.00935
717-A maintenance, furnace	0.00228	0.00168
717-C maintenance, boiler and superheater	0.00102	0.00088
717-D maintenance, feeders and burners	0.00047	0.00400[e]
718-A maintenance, economizer	0.00126	0.00059
718-D maintenance, soot blowers	0.00272[b]	0.00042
718-E maintenance, air preheater	.....	.....
718-F maintenance, all fans	0.00214[c]	0.00136[f]
Total maintenance except coal and ash equipment	0.00989	0.00893
718-B Prorate share of station maintenance, coal and ash equipment	0.00211	0.00233
718-B Maintenance, No. 22 coal- and ash-conveying equipment	0.00197[d]	0.00080
718-C Prorate share of maintenance, crusher and breaker	0.00026	0.00318
718-C Maintenance, mill	0.00334	0.01034[g]
Total maintenance coal and ash equipment	0.00768	0.01665
Auxiliary power	0.01140	0.01279
Fuel cost (reduced to a standard basis of $18 per 100,000,000 B.t.u.)	0.21462	0.21496
Total cost (labor, maintenance, auxiliary power, and fuel)	0.25374	0.26258

[a] One fireman and one mill operator for No. 22 bin system alone for nine months.
[b] Additional soot blowers installed (50 per cent of account).
[c] New rotor for induced-draft fan (45 per cent of account).
[d] Coal gates on weigh bins for tests (40 per cent of account).
[e] New burners (80 per cent of account).
[f] Repairs on spare induced-draft fan rotor (35 per cent of account).
[g] Unusually high maintenance costs due to changes in mill and high operating speed.

It would be perfectly natural to compare the principal final results of these two systems to other methods of combustion at this same station using the same unit basis. They are as follows:

	18 stokers	No. 22, bin system	No. 22, unit system	Another type, unit system
Period covered	93 mos.	24 mos.	11 mos.	26 mos.
Total cost in dollars per 1,000,000 B.t.u. delivered, with a coal cost of $18 per 100,000,000 B.t.u.	0.26847[a]	0.25374[b]	0.26258[b]	0.26216
Evaporative efficiency, per cent	77.4	83.85	83.7	81.1[c]
Auxiliary power, kw-hr. per ton	13.56	55.33	51.31	54.41

[a] Unusual expenses incurred by outages, maintenance, and renewal of all economizers included.
[b] Unusual expenses cited in Table 10 included.
[c] Records of the last few months indicate that rehabilitations have improved performance.

In concluding this presentation an attempt has been made to record herein only the facts, as they were and are in this particular case, for what they may be worth to the increasing store of information on these matters. It has been borne in mind that a precise critical analysis of the two methods of firing may as yet be premature in view of the many complications of the situation and developments at present in progress. It is as such that the author prefers to leave it.

Discussion

R. J. Bender.[2] Mr. Grunert's paper has over any previous similar studies the marked superiority of relating actual experience in an existing power plant and covering several years, and the author should be congratulated for the completeness and impartiality of his study.

But one point should be emphasized more than was done in the last part of the paper—not at all to criticize the paper, but to

[2] Whiting Corporation, Harvey, Ill. Mem. A.S.M.E.

warn against any rash conclusion as regards the financial ends of the two types of pulverized-coal firing; this is the fact that, during the period in which observations were made and records kept, considerable work had to be done on the equipment to adapt it to its needs, and the cost of this work has been included, of course, in the cost figures given in Tables 3, 5, and 10. As a fact, this should not be considered at all as inherent to any bin system or direct-fired system in particular; these costs were due to the fact that the equipment was of a comparatively new type and that improvements had to be made during this experimental period.

For instance, a furnace-maintenance cost of 0.042 is given for the bin system against 0.028 for a direct-fired system. Is this not due to the fact that the water-cooled furnace, as mentioned in the first part of the paper, had to be slightly corrected in the first period, while no repairs at all were done during the 11 months of the second period? It would be unfair to draw the conclusion that a difference of about 50 per cent should be expected in the cost of furnace maintenance between the two types of firing.

Further, a mill-maintenance cost of 0.061 is given for the bin system against 0.178 for direct firing. Here again it would be impossible to conclude that a direct-fired system is almost three times as hard on the mills as a bin system. But the paper states that the pulverizer, although modified, was not entirely suitable when running at the speed at which it had to run to follow the desired capacities and that changes had to be made for a better adaptation; then it is clear that the figures are not comparable.

In other words, it is safe to state that if Mr. Grunert would install today two boiler rooms, one with a bin system and one direct-fired, he would pick out his equipment such as to overcome those difficulties which have increased the costs of the plant described in the paper; and the comparative figures might be entirely different from what they are in Tables 3, 5, and 10.

It would therefore be very valuable if, at one of the next meetings, Mr. Grunert or some other engineer could compare the two types of firing not taken as two consecutive phases of the life of a boiler, but taken from two boilers of similar type, burning the same fuel, one direct-fired and the other fired with a bin system, and after a stage of perfection, or, rather, almost perfection, has been reached.

In this comparison it would be extremely interesting to take into account the comparative initial costs and fixed charges, and also to compare the room required for each type and to establish a cost balance of all these conditions—first, for a constant load and, second, for an extremely variable load.

Another point should be mentioned. Mr. Grunert figures the costs with the assumption that the cost of 1 kw-hr. is the same during 24 hours of a day; this is true in this case. But in some plants, and especially those making steam for processing purposes, it is possible to have cheaper power during the night than during the day, in which case a difference would show in favor of the bin system; and it remains to be seen whether this difference is sufficient to compensate the higher initial cost of such an installation.

E. G. Bailey.[3] Mr. Grunert is to be congratulated on his impartial analysis of the advantages and disadvantages of these two systems of firing pulverized coal as they apply to the Calumet installation and upon the completeness of his comparison of operating results. No. 22 boiler at Calumet was a distinct innovation in power-plant equipment at the time it was placed in operation, and the success of this pioneer installation has been an important contribution to the art. This installation has demonstrated to the engineering world that the use of low-grade coal in pulverized form is not only possible but highly practicable for large, high-capacity steam-generating units.

[3] President, Fuller-Lehigh Company, Fullerton, Pa. Mem. A.S.M.E.

At the time the original design was under consideration, a furnace capable of high-capacity operation using coals having a high ash content with a low ash-fusing temperature had not previously been developed. The Calumet furnace was the first completely water-cooled, hopper-bottom type furnace with water-cooled, intertube-type burners to be installed.

Experience with low-grade Midwestern coals on stoker settings had demonstrated that the accepted refractory furnace constructions were subject to high maintenance and periodic outages for furnace repairs because of the damaging scouring action of molten ash. The burning of this coal in pulverized form meant that the destructive action of the coal ash would be even more severe on any refractory-furnace construction.

The operation of this unit over a period of approximately 42 months, with results such as reported in this paper as to combustion efficiency and furnace maintenance, most certainly makes this installation an achievement from the standpoint of overall economy and reliability of this type of furnace construction. The basic design features of this furnace have been applied to many later installations. It is of interest to note that an undertaking in furnace construction of the magnitude of the Calumet furnace has required no major alterations. Improvements have been made on later similar installations as regards extending the water-cooling further down in the sides of the furnace hopper, widening the throat opening in the hopper, and the use of a more satisfactory junction between the top of the water-cooled front and rear walls and the boiler proper.

Although later developments have indicated the desirability of periodically removing the ash from the furnace in a molten condition through a tap hole, where the ash-fusing temperature is less than 2600 deg. fahr., nevertheless this water-cooled hopper bottom furnace has proved entirely satisfactory from an ash-removal standpoint. Ash having a high fusing temperature is removed in a dry condition, while low-fusing-temperature ash runs down the hopper floors and drips into the ashpit, so that the type of coal used has no ill effect on ash removal from this furnace.

The burners, which are of the horizontal intertube type, were the first turbulent burners of this type to be installed. They worked out very satisfactorily in this and many subsequent installations.

Joe A. Bergin.[4] From the valuable data presented, it appears to be a toss-up between stokers and pulverized-fuel firing when selecting power-plant fuel-burning equipment. The greater demand for steam per boiler is probably the deciding factor in favor of pulverized-fuel firing. The 80 lb. of coal per square feet of grate per hour is nearing the maximum fuel-burning rate for a stoker, and 300,000 lb. of steam per hour from a stoker-fired boiler is about the upper limit. Therefore, when the demand for steam exceeds that figure, it becomes necessary to pulverize the fuel for burning.

It is possible that efforts to increase the dollar efficiency of the power plant and the necessity for greater quantities of steam from a given space in the power-plant building will result in pulverized-fuel burning becoming more popular.

O. Von Voigtlander.[5] It would be desirable to know whether charges for capital investment, insurance, etc., have been considered in the author's set-up. It appears to be rather difficult to place a bin system into the same space occupied by a unit-mill system; therefore, considerable difference in plant cost will exist.

[4] Superintendent, Ohio Power Company, Philo, Ohio. Mem. A.S.M.E.

[5] Assistant Mechanical Engineer, Allied Engineers, Inc., Jackson, Mich. Mem. A.S.M.E.

Furthermore, the insurance rate on a bin system is in some known installations three times as great as the unit-mill system. This increased insurance rate may apply to the complete boiler room depending upon the method of construction used. Before a financial set-up for an appropriation can be made, both the operating and construction costs must be known in their entirety.

CONWAY PIERCE.[6] Engineers will appreciate the practical step made by the author in presenting an authoritative report in this form. In addition to the valuable data already presented, many engineers, particularly those from the Middle West, would be glad to learn how the two pulverized-coal systems performed with respect to their ability to handle various grades of coal efficiently and practically as compared with the chain-grate stokers.

Many of us are confronted with the problem of using coals purchased from numerous sources. For this reason, information of this matter is particularly acceptable when available.

AUTHOR'S CLOSURE

In reference to Mr. Bender's discussion, the cost figures in Tables 3, 5, and 10 are actual expenses incurred. They do not include any expense on the part of the manufacturer to meet contract guarantees. The equipment in both systems was supposedly "adapted to its needs" by contract agreement, and wherever it failed in this respect it was regarded as the responsibility of the manufacturer until accepted.

As stated, there was no attempt to make general conclusions on the various item costs. The footnotes to Table 10 were an attempt to point out why direct comparisons on individual items should be made only with reservations. The wide differences between 0.061 for the bin-system mill maintenance and 0.178 for the direct-firing apparently is not clear from footnote *g*, Table 10. When this mill was changed over to a unit mill, one-half the expense of conversion was taken over, as it was a mutual experimental program. Inasmuch as only 11 months' expense was available at the time of preparation of this paper, this one-half reconstruction cost had to be distributed over this relatively short period. If this cost had not been included, the unit-mill maintenance costs would have been nearly the same, and if the direct comparison is to be considered, this would probably be a fair conclusion.

In the furnace maintenance cost of 0.042 as against 0.028, here again the relatively short period of 11 months causes a seeming discrepancy. It so happened that a repair job was done on this furnace shortly following the period taken, which, if included, would have made the unit costs much the same. There is no reason why the furnace costs with both systems on this unit should not be the same in the long run and practically all of that confined to the refractory sections shown. The expense of removal of the 30 blocks in the furnace during the bin-system operation is included, but is relatively of minor consequence in the account.

While the cost figures as given are comparative between the two systems as described, except as to certain irregularities pointed out, they would have to be interpreted in a hypothetical plant. One would certainly attempt to overcome "those difficulties which have increased costs." Some of these reservations might be pointed out. The unit-boiler-room labor item could be reduced, because the same personnel could handle more equipment. The newer improved screen construction should reduce furnace maintenance costs to little or nothing, and it would certainly be expected that developments in mill design should reduce mill-maintenance costs and power consumption. However, these are opinions, while the data given are the actual facts in this particular case. The author makes no attempt to conclusively settle this generally controversial matter.

In regard to Mr. Von Voigtlander's discussion, the investment matter has not been dealt with. This is an important angle to the problem and might even be the controlling factor.

The author is in substantial agreement with the remarks of Mr. E. G. Bailey and Mr. J. A. Bergin.

---

[6] Field Engineer, Combustion Engineering Corporation, New York, N. Y.

FSP-53-5

Recovery of Fly Ash From Pulverized-Fuel Stacks by Use of Stack Sprays

By J. W. MACKENZIE,[1] JACKSON, MICH.

One of the first central stations designed for the unit system of pulverized-fuel operation with preheated air encountered difficulty due to fly ash from the stack. The extent of the trouble was determined by locating boards freshly painted with white paint throughout the district. A water spray installed in the stack stopped all complaints, but necessitated lining the stack with acid-resisting material. Tests show that with the spray in operation about 5 per cent of the coal is discharged as solids from the stack, and that the spray takes out about 35 per cent of the solids coming to the stack. A detailed description is included of the equipment and methods used for collecting representative samples of the solids in the flue gases.

This paper outlines the influences which led the Consumers Power Company to adopt a spray system in the stacks of its pulverized-fuel fire plants as a means of combating the fly-ash problem, and shows the results which operation and tests have revealed as to the effectiveness of the system.



STARTING of the Kalamazoo steam-generating plant of the Consumers Power Company in 1927 marked the advent of this company into the field of pulverized-fuel firing, and was in a sense a pioneering step in unit-system operation. While several industrial plants had been designed and built for this method of firing coal, and a few central stations had substituted unit pulverizers for a portion of their stoker equipment, the Kalamazoo plant was one of the first of the larger central stations to be designed and built entirely for this method of operation.

A general survey of the field before this plant was designed showed that while the use of preheat for both primary and secondary air had been experimented with by several companies, there was very little of accurate data to show what could be expected from the continuous operation of unit mills under these conditions. It is, therefore, not surprising that a few changes to the equipment were found desirable after the plant was started. Indeed, the surprising part was that so few changes were necessary.

The matter of possible complaints due to the settling of fly ash in the vicinity of the plant was carefully considered before the plant was designed, and visits were made to quite a number of existing pulverized-fuel plants to learn what had been the experience of others. From the information thus gained, it was thought that, with proper furnace design and good operation, cinder eliminators would not be required. Space, however, was provided so that cinder traps could be installed later.

Determining Extent of Nuisance

During the early operation of the plant, trouble with coarse pulverization and burner difficulties resulted in more solids being discharged from the stack than was expected, and before these troubles could be overcome the complaints made it necessary to do something at once. To get some record of the actual conditions, the area surrounding the plant for a radius of about 12 city blocks was divided into sections. Each morning a board about 16 in. square was placed near the center of each section, the upper side of the board having been freshly painted with white paint. Any particles of dust which settled on the board were caught in the surface of the paint, and when the paint dried this made a permanent record. By watching the wind direction and comparing the record of each test location obtained when the stack gases were passing over that district with records of the same location when the wind carried the gases in another direction, the relative effect of the stack discharge was determined.

These records showed that objectionable quantities of material were being deposited from the stack, with the worst condition about three or four city blocks from the plant, and disappearing at a radius of about nine blocks. A considerable part of this deposit appeared to be carbon in the form of small particles of coke which was undoubtedly largely the result of poor pulverization. To correct this condition necessitated changes in the mills and equipment requiring time, so in view of the local complaints, it was decided to install a water spray in the concrete stack serving these boilers, in the hope of washing out part of the solid materials.

Stack-Spray Arrangement

This stack is 12 ft. inside diameter, with a double breeching connection about 21 ft. above the floor of the stack. A group of four spray nozzles located about 20 ft. above the top of the breeching connection and arranged to form a combined downward spray to fill the cross-section of the stack was first tried out. The results were rather better than was anticipated, but the pipe connections to the spray nozzles soon corroded, and a single center spray with a capacity of about 30 gal. per min. when supplied with water at 10 lb. pressure was substituted in an arrangement shown as the main spray in Fig. 1. This arrangement has the advantage that it can be inspected and repaired from outside the stack, and operation has shown that when the spray covers the area of the stack, an increase in the quantity of water has but little effect on the results.

The water and entrained ash were drained from the bottom of the stack into the sump of the ash-sluice system installed for handling the ashes from the plant, and was discharged from the plant with the regular ash pumps.

---

[1] Mechanical Engineer, Consumers Power Company. Mr. Mackenzie was graduated in 1916 from the University of Michigan, where he took the mechanical engineering course. Upon graduation he worked for a year and a half for automobile companies, resigning to enter the employ of the Consumers Power Co. in 1917, as efficiency man in their Jackson plant. He has continued in their employ since then, having held the position of mechanical engineer since 1924.

Contributed by the Power Division and presented at the Semi-Annual Meeting of the A.S.M.E. at Detroit, Mich., June 9 to 12, 1930.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

After about a week's operation trouble was experienced due to flakes of accumulated fine ash coming from the stack and settling in the immediate vicinity of the plant. These deposits were confined to a radius of less than 500 ft. An investigation showed that an accumulation of this fine moist ash had collected

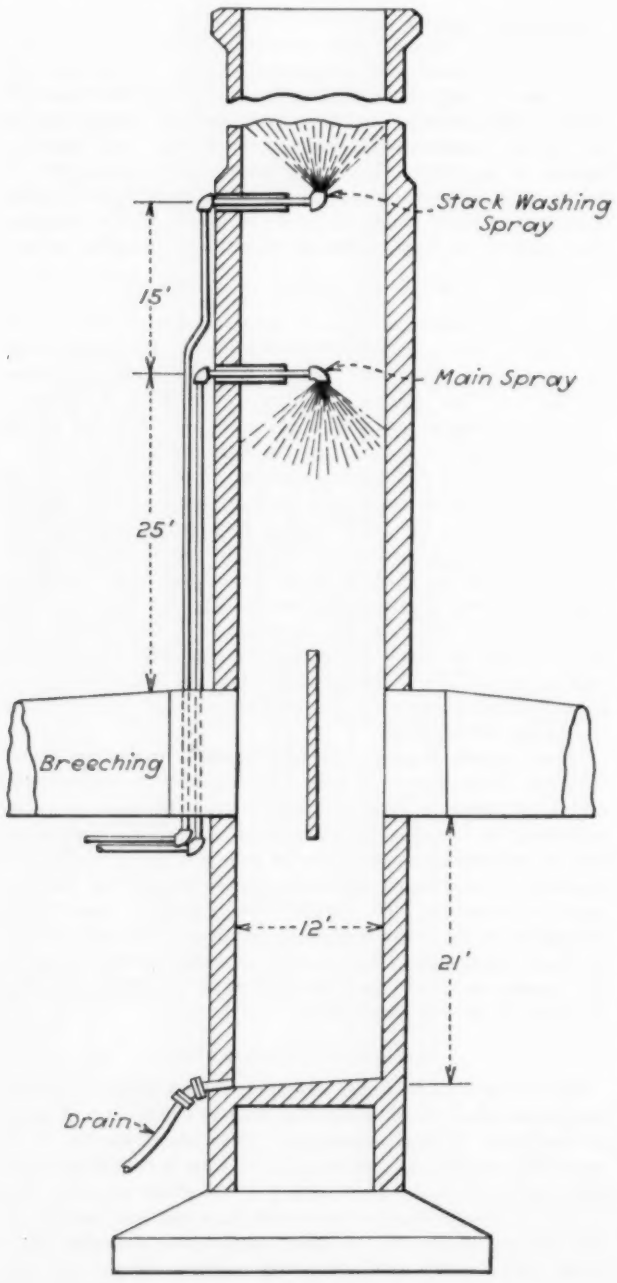


FIG. 1 ARRANGEMENT OF STACK SPRAYS

in a circular band about 3 in. thick and extending from a point about 15 ft. above the level of the spray nozzle upward for about 3 or 4 ft. The moist ash gradually builds up at this point until it begins to flake off and is carried out of the stack. This difficulty was readily overcome by installing a second, or stack washing spray, about 15 ft. above the main spray as shown in Fig. 1. This second spray is a duplicate of the main spray, but is turned upward so that the water from it strikes the wall of the stack at a point above the ash deposit, and this water running down the stack wall washes the accumulation down. The use of this stack-washing spray about 20 min. a day is sufficient to eliminate all trouble from this source.

Some attempts were made to measure the per cent of material removed from the gases by the spray, but the breeching arrangement made it impracticable to obtain samples where the gases were traveling in a vertical direction, and it was found that due to a much higher concentration of solids at the bottom of the horizontal ducts, it was impossible to obtain anything like a representative sample or measurement of the solids under these conditions. The records obtained by means of the painted boards, however, indicated that, with the spray in operation, the amount of dust settling per square foot of area was not perceptibly different when the wind was carrying the gases from the stack over that district from that when the wind was in another direction.

In view of the results obtained at the Kalamazoo plant, it was decided to incorporate the same system in the design of the company's Wealthy Street plant No. 2 located at Grand Rapids, Michigan. This plant is inside the city limits, and no complaints have been received after about one year's operation.

MEASUREMENT OF FLY-ASH RECOVERY

A breeching arrangement at this plant, which includes a fairly long vertical duct from the induced-draft fan of each boiler, offers excellent facilities for obtaining characteristic samples of fly ash in the gas, and a number of tests have been made.

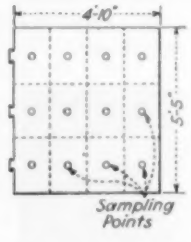


FIG. 2 A CROSS-SECTION OF THE DUCT WAS DIVIDED INTO TWELVE EQUAL PARTS, AND A SAMPLE OF GAS ASPIRATED FROM THE CENTER OF EACH SECTION

The method adopted in making these tests was to divide the cross-section of the vertical duct into 12 equal parts as indicated in Fig. 2, and to aspirate a sample of gas with its entrained solids from the center of each section.

It is very important in obtaining a sample of this kind that the rate of flow into the entrance of the sampling tube shall be the same as the velocity of the gases past the tube. Not only will the total quantity of solids vary with the velocity of the gases entering the tube but the percentage of fines is also affected. This is more readily understood by referring to Fig. 3. If the velocity into the sampling tube exceeds the velocity of the surrounding gases, some of the materials which would otherwise pass the sampling pipe are diverted from their normal path and carried into the sample, and since the finer particles are more easily deflected, the percentage of fines in the sample will show too large. If, on the other hand, the velocity into the sampling tube is lower than that of the surrounding gases, some of the gases are forced outside the area of the sampling-pipe entrance, resulting in less total solids in the sample with too low a percentage of fines. By maintaining the same velocity within as without the sampling pipe, the least possible disturbance in the gas stream is obtained with a quantity and quality of sample which is characteristic of the total material in the gases. To insure this minimum disturbance at the entrance of the sample pipe, it is desirable to use a thin straight-walled tube for this purpose, with the entrance edges sharpened.

The apparatus used for collecting the samples in these tests is indicated in Fig. 4, and consists of a thin-walled sampling pipe (*a*) connected to a vacuum cleaner bag (*b*) through an air-operated ejector (*c*) with a valve (*d*) for controlling the flow in the sampling pipe. The sensitive inclined draft gage (*e*) is

connected to indicate the differential pressure between the pitot tube (*g*) with its open end in the entrance of the sampling pipe and the pitot tube (*h*) with its open end outside the sampling pipe.

It will be noted that the pitot tubes are turned with the flow of the gases, which would theoretically show the static head minus the velocity head. This arrangement was adopted due to the fact that if the pitot tubes are turned against the flow of the gases they become clogged with the fly ash. While this arrangement is not accurate for actually measuring velocities, tests made with a sampling tube equipped with both types of pitot tubes showed that the arrangement adopted was satisfactory for establishing the same velocity into the tube as existed in the surrounding area.

The procedure followed in collecting samples was to set the sample tube to draw from the center of one section of the duct area, and with a vacuum-cleaner bag in place the ejector was started and the flow in the sample tube adjusted by means of the valve (*d*). The vacuum-cleaner bag was then removed, and a duplicate bag which had been dried and accurately weighed

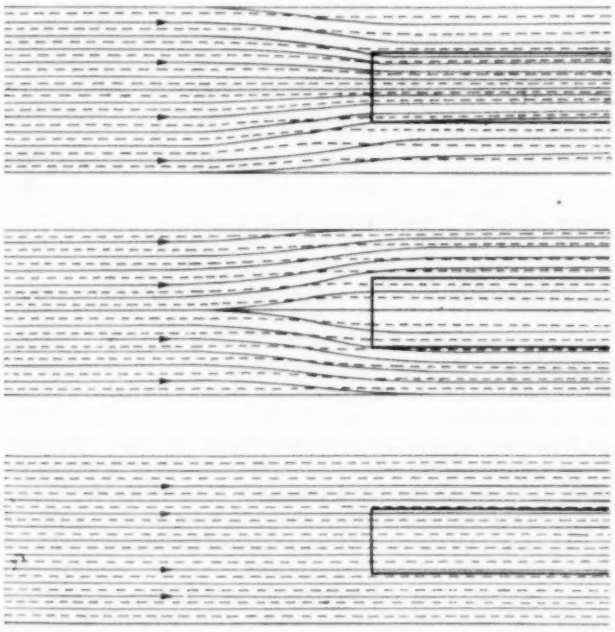


FIG. 3 QUANTITY OF SOLIDS AND PER CENT OF FINES VARY WITH VELOCITY OF GASES

(Above, velocity in sampling pipe greater than surrounding gas; center, velocity in sampling pipe less than surrounding gas; at bottom, velocity in sampling pipe equal to that of surrounding gas.)

was substituted, and a three-minute sample collected, after which the sampling tube was moved to the next section of the duct area and another three-minute sample taken. In this way a three-minute sample was gathered from each section of the duct, careful watch being kept of the gage (*e*), and the flow controlled so as to maintain equal velocities at the sampling end of the tube. The vacuum-cleaner bag containing the sample was then dried and reweighed. The increase in weight times the ratio of the area of one section of the duct to the area of the entrance of the sample tube equal the total solids passing through the duct in a 3-min. period.

The sample was then removed from the bag and tested for fineness, and analyzed for percentage of combustible. Some of the very fine particles became imbedded in the bag so that it was impossible to brush them out; this varied from 15 to 40 per cent of the total weight of the sample. In making up the data showing the fineness of these samples it has been assumed that all material lost would pass through the 200-mesh screen, and has been included in this figure. This undoubtedly is slightly in error, but it is believed that the error is within the limits of the accuracy of the other data.

The percentage of combustible is given as determined from the amount of material actually recovered from the bag, and takes no consideration of the percentage of combustible in the

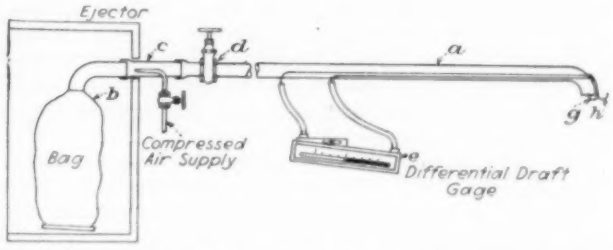


FIG. 4 ARRANGEMENT OF APPARATUS FOR COLLECTING SAMPLES

material lost or left in the bag. Preliminary tests, together with the data given here, show quite consistent results when the same firing conditions were maintained, indicating that the results are fairly accurate. The following procedure was used for determining the quantity and nature of the material removed from the stack gases by the spray.

While the samples of fly ash were being collected as outlined, all of the water and ash discharged from the bottom of the stack was diverted for a short time into a small tank, the actual time being determined by means of a stop watch. This sample was filtered, and the residue dried, weighed, and tested for fineness and combustible. From the data then available, the total weight, as well as the percentage of carbon and ash recovered from the stack per hour, was easily determined.

RESULTS OF TESTS

The results of these tests are shown in the accompanying tables, and represent conditions from about three-quarters of normal

TABLE 1 HOURLY RESULTS OF TESTS

Test number	1	2	3	4
Number of boilers in use	2	2	2	1
Total lb. coal fired	23,183	17,711	16,888	11,644
Fly ash to stack, lb.	1,951	997	1,498	1,299
Fly ash recovered, lb.	756	624	463	437
Ratio, fly ash to coal, %	8.42	5.63	8.87	11.16
Carbon in fly ash, %	.....	49.0	35.9	40.4
Carbon in recovered fly ash, %	.....	46.2	37.4	46.8
Per cent of fly ash recovered	38.8	46.16	30.90	33.60

TABLE 2 FINENESS OF ASH

Test no.		Through 200	Through 100	Through 48	Through 28	Over 28
2	Total fly ash	62.7	81.2	95.6	99.6	0.4
	Recovered fly ash	43.6	62.3	91.2	99.6	0.4
3	Total fly ash	60.4	81.5	96.4	99.8	0.2
	Recovered fly ash	60.3	79.9	96.5	99.9	0.1
4	Total fly ash	47.4	62.5	95.1	99.8	0.2
	Recovered fly ash	53.2	71.1	96.0	99.8	0.2

TABLE 3 PER CENT CARBON AND FINENESS OF RECOVERED ASH

Test no.		Per cent carbon	Through 200	Through 100	Through 48	Through 28	Over 28
5	With spray	34.0	61.1	86.6	98.8	99.9	0.1
	Without spray	56.4	27.3	56.2	93.4	99.6	0.1
6	With spray	38.8	73.5	93.0	99.6	99.9	0.1
	Without spray	48.3	48.6	75.4	97.4	99.8	0.2

boiler capacity to full normal capacity. The ratio of the fly ash to the coal fired is undoubtedly much lower with the reduced velocities at lower ratings, but no satisfactory data have been

obtained at the lower ratings, due to difficulty with the solids settling out in the breechings with the reduced velocities.

It will be noted that, under the conditions covered by these tests, the ratio of the fly ash to the total coal varies from about $5^1/_2$ to 11 per cent of the coal used, and with the spray in operation about 37 per cent of this amount is recovered. The tests show that under normal operation with the stack spray in use there is still about 5 per cent of the total weight of the coal discharged from the stack.

The average velocity of the gases in the stack under normal full load has been determined to be about 1000 ft. per min., with a volume of 127,500 cu. ft. per min., which gives a loading of about 0.156 lb. of solids per 1000 cu. ft. of gas, and of these solids more than 65 per cent will pass through a 100-mesh screen, a loading and fineness which apparently give sufficient dispersion to avoid detection.

Tests were also made to determine the per cent and the relative fineness of the solids which would be deposited in the stack without the use of the spray. These tests show that with the spray in operation from three to four times as much material is recovered as without it. The spray also greatly increases the recovery of the fines, and is much more effective on the ash than on the carbon in the gases.

Effect of Water on the Stack

After about nine months' operation, an inspection of the inside of the Kalamazoo stack showed that the concrete surface was being seriously eroded by the use of the water. The effects were most marked at the bottom of the stack, diminishing in an upward direction.

While the spray had originally been installed only as a temporary measure, it was decided best to keep it in operation after all firing troubles were corrected. Accordingly, an acid-resisting lining was installed extending from the base of the stack to a point about 50 ft. above the spray. This lining is constructed from vitrified tile laid with thin joints with an acid-resisting cement; a similar lining extending the full height of the stack was installed in the stack at the Grand Rapids plant, and to date these linings are showing no signs of erosion.

From the experience thus far obtained it appears that where a spray is to be used in a stack for removing fly ash from flue gases, the spray should be installed at least 20 ft. above the point where the gases enter the stack, and that there should be a distance of at least two stack diameters from the bottom of the breeching connection to the bottom of the stack.

Conclusions

From the data obtained thus far it appears that, providing the gas velocities do not exceed approximately 1200 ft. per min., a spray in the stack of a pulverized-fuel installation can be used as one means of reducing the fly ash discharged, and that a recovery of about 35 per cent of the solids can be expected. The use of larger quantities of water offers no advantage providing the spray covers the cross-sectional area of the stack. Means must be provided for periodically washing down accumulations of ash which will collect on the stack wall above the spray. An acid-resisting lining is necessary to protect the stack interior.

It appears that reducing the dust loading of the gases increases their carrying power, thus increasing the dispersion. Experiments and tests should be continued to determine the effect of different types of sprays.

Discussion

Ralph A. Sherman.[2] All will agree that the discharge of fly ash from the stacks of boiler furnaces, whether stoker- or pulverized-fuel-fired, is one of the most serious problems of power-plant operation. Much study is being given to means of abatement or of elimination of the fly ash and to methods of its disposal. The author's paper is of value in that it shows the limitations of the stack spray rather than its success. Although a recovery of 35 per cent of the solids is a step in the right direction, one cannot be content with that, even though it is true that the remainder is so dispersed that it avoids detection or complaint at present.

One of the first requirements of the problem is accurate knowledge of the amount of ash being discharged from stacks. The author has outlined the principles which should be observed in the determination of the fly ash and has made careful measurements. More similar data are needed, and it would be valuable if some central body would sponsor their collection in a large number of plants, either by the individual operators or, better, by the same observers going from station to station.

It is impossible to calculate, from the data which the author presents, the percentage of ash in the coal fired which was carried up the stack, because the author has not given the ash content of the coal. This value appears to the writer to be a more significant value in such determinations than the percentage of the coal fired, for the ash and carbon are then calculated separately. If we assume that the coal contained 10 per cent of ash, the percentage of the total ash carried as fly ash was about 28 per cent for Test 2 and 70 per cent for Test 4, as shown in Table 1. This is a wide variation for the change in rate of firing and is probably accounted for by other factors.

It may be recalled that some determinations made by the writer[3] at the Toronto, Ohio, Station of the Pennsylvania-Ohio Power & Light Company showed less than 50 per cent of the ash fired carried to the stack with a dry-bottom furnace and not more than 30 per cent when firing similar coal to the same type of furnace except that it was changed to a slagging bottom furnace.

Author's Closure

Referring to the discussion, the author wishes to state that the fuel used during these tests was West Virginia nut and slack coal with a moisture content of about 4 per cent, B.t.u. value of about 13,200 as fired, and an ash content of slightly less than 9 per cent.

Apparently there is some error in the amount of fly ash to the stack as shown in the No. 2 test, as it is noted that this figure does not agree with the rest of the data, whereas the amount of ash recovered by the spray during this test is consistent with the other data. Also, from tests made on the quantity and quality of ash removed from the bottom of these furnaces, it has been found that under normal conditions about 35 to 40 per cent of the ash in the coal remains in the furnaces, which agrees fairly well with the calculated results for the No. 3 and No. 4 tests. The figures shown in the No. 2 test were included, notwithstanding this discrepancy, due to the fact that the per cent recovery with regard to the total coal fired appears to be correct, and the data obtained therefrom are correspondingly valuable.

The author is not inclined to agree with Mr. Sherman regarding the importance of the ratio of the ash discharge from the stack to the total ash in the coal, since it is the total quantity of solids emitted from the stack which must be considered, and in the experience at the Kalamazoo Plant there has probably been more complaint from the surrounding neighborhood when an appreciable percentage of unburned carbon was present in the deposit than in cases when it was made up largely of the lighter-colored ash particles.

[2] Fuel Engineer, Battelle Memorial Institute, Columbus, Ohio. Mem. A.S.M.E.

[3] Sherman, Ralph A.; Nicholls, P.; and Taylor, Edmund. "Study of Some Factors in Removal of Ash as Molten Slag From Powdered-Coal Furnaces." Trans. A.S.M.E., Fuels and Steam Power Divisions, vol. 51, no. 22, Sept.-Dec., 1929, pp. 399–413.

FSP-53-6

Tests of the Resistance to Repeated Pressure of Forged, Riveted, and Welded Boiler Shells

By H. F. MOORE,[1] URBANA, ILL.

Data regarding the performance of welded joints in resisting repeated stresses are very important in any consideration of the adoption of welded construction for power boilers, and it is the object of this paper to present such data. Comparative tests were made on (1) a standard A.S.M.E. riveted drum, (2) manganese-steel riveted drum, (3) forged seamless-steel shell, (4) two forge-and-hammer-welded shells, and (5) eight fusion-welded (metallic-arc) shells, each drum or shell being subjected to applications of pressure well above the allowable working pressure.

The results indicate that under repeated stress properly made welded joints are stronger than the plates themselves when weakened by "stress raisers" such as small holes or surface defects. The tests also give useful data on the strength of drum heads when subjected to intermittent pressures.

THE use of welding for the joints of seams in pressure vessels is of great interest at the present time to the American Welding Society and to the Boiler Code Committee of The American Society of Mechanical Engineers, which has under consideration the revision of its rules bearing on unfired pressure vessels, and which has published "Proposed Specifications for Fusion Welding of Drums or Shells of Power Boilers" and has solicited discussion thereon. About three years ago tests were started at the Barberton works of the Babcock & Wilcox Company on the performance of welded and riveted vessels under repeated applications of pressure. A preliminary report on these tests was made by the author and published in the *Iron Age*, September 5, 1929. There had previously been published some test data of repeated stress tests on small specimens, but there was little knowledge of the performance of fusion-welded joints on full-size pressure vessels.

In making fatigue tests of small specimens of metal, six or more specimens of each metal are ordinarily tested under various stresses, and usually some tests for each metal are run to at least 10,000,000 cycles of stress. Thus for nearly all metals it is possible to determine the "endurance limit" below which the metal will withstand an indefinitely large number of cycles of stress without fracture.

This method of testing six or more specimens of a kind was not feasible for application to full-sized test shells on account of the expense and time necessary for preparing the shells and for making the tests. It was necessary to cut down the number of specimens below that which would be required for making exact determinations of the "endurance limit;" however, the tests which were made to study the relative length of endurance of full-size test shells have given data which are directly applicable to the problem at hand. In the ordinary service of a boiler, the cycle of stress on the shell due to internal pressure is repeated at most a few thousand times, and the stress is not reversed, varying from zero to a maximum with slight fluctuations near the maximum stress. Thus it was feasible to run the tests to numbers of cycles of stress greatly in excess of the number of cycles in the normal service of a boiler or of a pressure vessel.

Test Specimens

Two types of test specimens were used. For the testing of riveted drums the test specimen is of the type shown in Fig. 1.

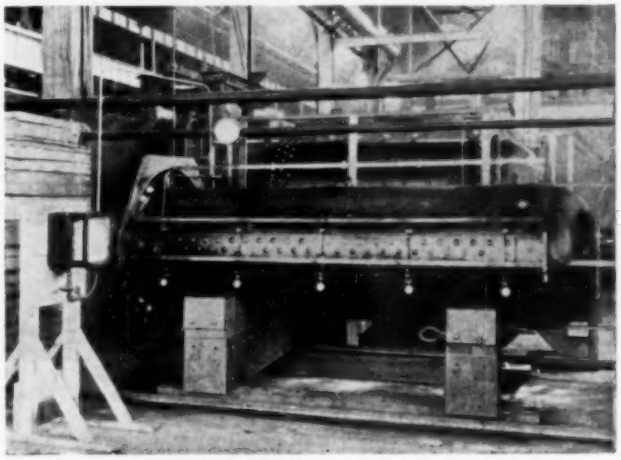


Fig. 1 Riveted Drum Arranged for Tests

This type of specimen is referred to as a test drum. Two test specimens of this type were used, one of A.S.M.E. code boiler plate having a minimum tensile strength of 55,000 lb. per sq. in., and the other of manganese-steel plate having a minimum tensile strength of 105,000 lb. per sq. in.

Fig. 2 shows the second type of test specimen used. This type of specimen is referred to as a test shell. The details of the arrangement are shown in Fig. 3. The test shell is placed around an inner shell, leaving an inner annular space between the two shells. Packing and packing glands close the ends of this annular space and water under pressure is admitted into it. With this type of specimen, the stresses set up are thus limited to hoop tension, and only a small quantity of water is required. Eleven specimens of this type were used: (1) A forged seamless test shell, (2) two forge-and-hammer-welded test shells, and (3) eight fusion-welded test shells, all welded by the metallic-arc method.

Test Apparatus

Fig. 4 is a diagrammatic view of the apparatus. Water under

---

[1] Research Professor of Engineering Materials, University of Illinois. Mem. A.S.M.E. In connection with the tests given in this paper Mr. Moore served as special consulting engineer for the Babcock & Wilcox Company. He has served as instructor in science and mathematics at Colby Academy, instructor in machine design at Cornell University, and mechanical engineer for Riehle Brothers Testing Machine Company and Assistant Professor of Mechanics at the University of Wisconsin. He is the author of many technical papers and bulletins of the University of Illinois Experiment Station.

Contributed by the Power Division and presented at the Semi-Annual Meeting, Detroit, Mich., June 9 to 12, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

a pressure of 1600 lb. per sq. in. was taken from the pressure mains of the Babcock & Wilcox Company's works, and this pressure was regulated by an accumulator. The water passed through a reducing valve and then through the mechanically operated valves, shown diagrammatically in Fig. 4, which by their action, alternately admitted water to the test shell and allowed the pressure to be relieved through an exhaust pipe. This operating valve was driven through link work by an electric motor with variable speed. The link work was fitted with a revolution counter, which indicated the number of cycles of pressure applied. The set-up of apparatus comprised six sets of valves, so that six test shells could be tested at one time.

The pressure applied to each specimen was shown by a Bourdon gage, which was attached directly to the specimen, and was also measured on a recording-pressure gage. The rate of application of pressure averaged about 13 cycles per minute. The

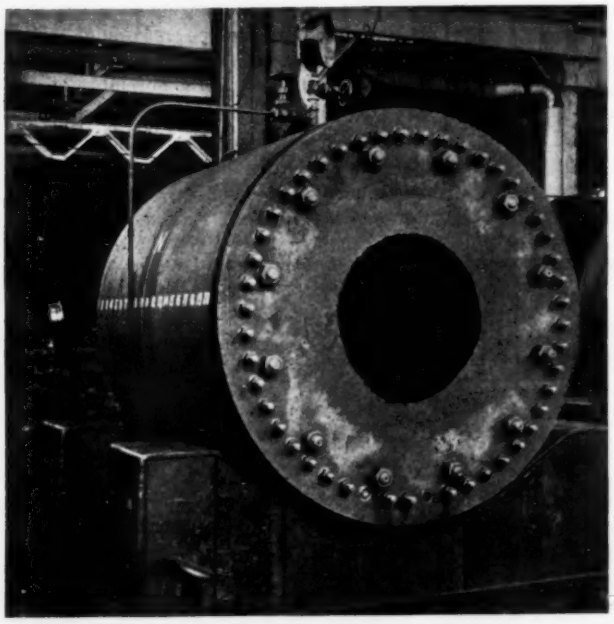


Fig. 2 Test Shell Arranged for Tests

deformation of test specimen was measured at intervals during the test by means of strain gages, which indicated any marked distortion of a specimen during the progress of the test.

A careful watch was kept to detect incipient fracture. Any cracks developed by the repeated pressure started on the inside of the shell, but before failure had extended very far the crack worked through to the outside, and it was detected by the leakage of a fine stream or spray of water. Table 1 gives a summary of the tests up to date, and the numbers assigned to the test shells refer to this table.

Manganese-Steel Test Drum

The manganese-steel drum was constructed of plate having a minimum tensile strength of 105,000 lb. per sq. in., was designed for the working pressure with a factor of safety of five, and the test pressure was alternated from zero to that corresponding to 50 per cent more than the working pressure. The heads were figured in accordance with the Massachusetts standard except that a tensile strength of 105,000 lb. per sq. in. was used in place of 55,000 lb. per sq. in. Fig. 5 shows the principal dimensions of this drum. The deflection of the manganese-steel heads on applying the pressure was greater than if they had been made of A.S.M.E. standard boiler plate, both on account of the high-tensile-strength steel stretching more than the lower-tensile-strength steel for a stress corresponding to a given factor of safety, and on account of the effect of the cross strains in the heads, which are not fully covered by the formula when steel of a higher tensile strength is used.

During the progress of the test, slight leaks appeared around the buttstrap, and after 130,225 cycles of stress, a leak developed near the manhole opening in the head of the drum, which was so large that a fracture was indicated. The test was stopped and examination showed a definite fracture. An attempt was made to patch this fracture by chipping out the fracture to form a double

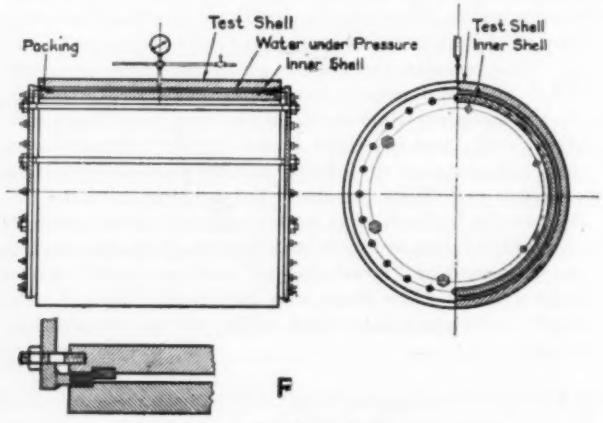


Fig. 3 Test Shell

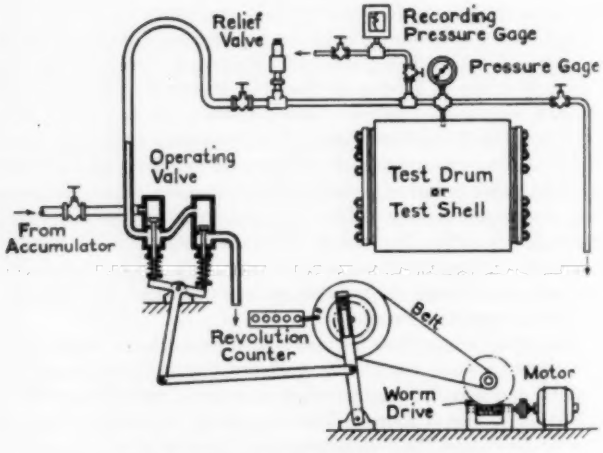


Fig. 4 Diagram of Test Equipment

V and welding. After 143 additional cycles of pressure, the drum head split violently through the crack repaired by welding. The measurements of strain during this test showed no evidence of appreciable distortion of the drum, except in the immediate vicinity of the crack.

A.S.M.E. Standard Drum

The riveted drum of A.S.M.E. code plate was constructed in accordance with the A.S.M.E. standard except that the heads were Massachusetts standard. The reason that the heads were made Massachusetts standard was because the Boiler Code Committee proposed at the time to revise its code to make the rules for heads more nearly in line with those contained in the Massachusetts code. The A.S.M.E. code as revised requires a thicker manhole head than was used in the test drum. Fig. 6 shows the principal dimensions of this drum. It was calked on the inside and not on the outside.

The test pressure was alternated between zero and that corresponding to 50 per cent more than the maximum allowable working pressure given by the A.S.M.E. code. Small leaks developed during the first part of the test, which were stopped by internal re-calking. A total number of 1,013,840 cycles was applied to the drum and no fracture resulted. The measurements of strain gave no evidence of permanent distortion during the test.

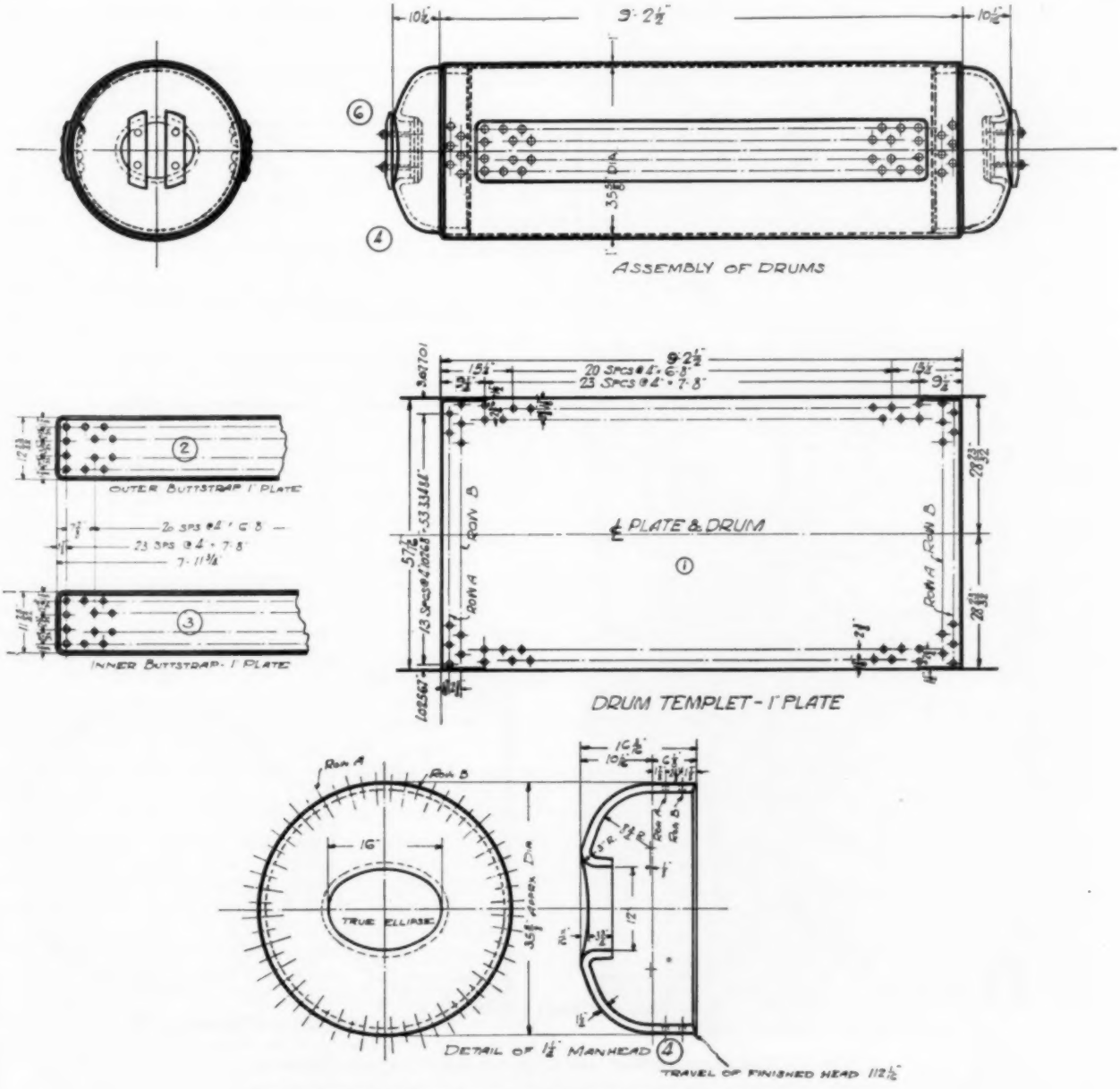


FIG. 5 SPECIAL 36-INCH DIAMETER DRUM OF HIGH-TENSION STEEL PLATE

FORGED SEAMLESS TEST SHELL

The forged seamless test shell was made of steel in accordance with the A.S.M.E. Boiler Code specifications for seamless-steel drum forgings for steel having a minimum tensile strength of 60,000 lb. per sq. in. All the tests called for by the A.S.M.E. Boiler Code for this class of steel were met and the drum was annealed above its critical temperature as called for by the code. The test pressure was alternated from about zero to that corresponding to 50 per cent more than the maximum allowable working pressure given by the A.S.M.E. code.

The seamless forged test shell was subjected to 446,950 cycles of pressure, at which point the test shell fractured at a tapped hole provided for the pressure-gage connection. In this test, as in several others, it was quite evident that this tapped hole acts as a region of stress concentration, where a spreading fatigue crack is liable to start. During the test the strain measurements showed no evidence of permanent distortion.

FORGE-AND-HAMMER-WELDED TEST SHELLS

The first forge-and-hammer-welded test shell (No. 12) was of German manufacture and was constructed of open-hearth quality plate of 35 to 44 kg. per sq. mm. tensile strength with corresponding elongation, the quality of the plate conforming to the pro-

visions of the German steam-boiler laws. The foregoing tensile strength corresponds to a tensile range of approximately 50,000 to 62,500 lb. per sq. in. This test shell had one longitudinal forge-and-hammer weld, made by the so-called gas-welding process prevalent in Germany, the weld being of the same thickness as the plate.

This test shell had no tapped hole in it, the pressure gage being attached to the inner shell of the test apparatus. This 611,184 cycles of stress when it cracked in the plate quite a distance from the weld. This shell had no tapped hole for gage connections, but the failure took place at a well-marked longitudinal surface defect, consisting of a groove about $^1/_{16}$ in. wide, $^1/_{32}$ in. deep, and some 2 ft. long. This fracture is shown in Fig. 8, and it is to be noted that the defect seemed sufficient to locate the start of fatigue failure, although, through being near one end, it was not at the point of maximum stress.

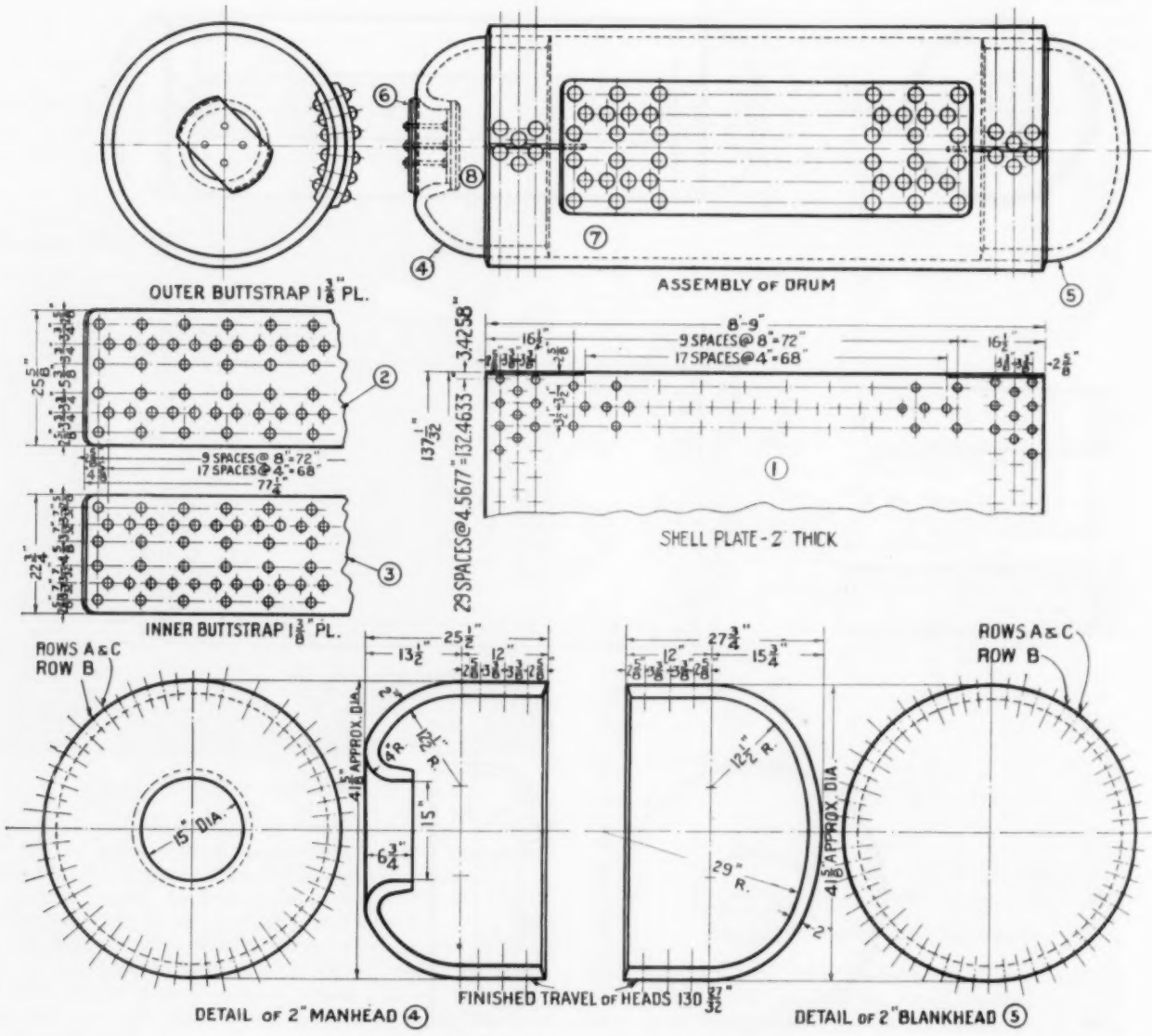


FIG. 6 EXPERIMENTAL TEST DRUM OF A.S.M.E. STANDARD MATERIAL

specimen has withstood 890,000 cycles without failure and the test is still in progress.

The second forge-and-hammer-welded shell (No. 13) was made of A.S.M.E. code plate having a minimum tensile strength of 45,000 lb. and the test pressure was alternated between zero and that which produced a stress of 16,500 lb. per sq. in. in the plate. The thickness at the weld was $2^1/_4$ in. or $^1/_4$ in. more than that of the plate. Strain-gage measurements showed that the stress in the welded seam was 14,000 lb. per sq. in. when the stress in the plate was 16,500 lb. per sq. in.

This second forge-and-hammer-welded shell was subjected to

The A.S.M.E. Boiler Code specifies that the stress shall not exceed 7000 lb. per sq. in. for forged-welded joints for power boilers and 8000 lb. per sq. in. for unfired pressure vessels. The forge-and-hammer-welded shells were tested at a higher pressure than that corresponding to 50 per cent more than the allowable A.S.M.E. code pressure in order that the results might be directly comparable with those secured from the fusion-welded shells.

FUSION-WELDED TEST SHELLS

The fusion-welded test shells were made of A.S.M.E. code boiler plate having a minimum tensile strength of 55,000 lb.

per sq. in. Each test shell had one longitudinal arc-welded joint, this longitudinal joint extending the entire length of the test shell. The excess weld metal on both the inner and outer surfaces was chipped and ground off so that no reinforcement of the weld metal was present. All arc-welded test shells were strain annealed at 1200 deg. fahr. previous to testing. The test pressure was alternated from zero to that which produced a stress of 16,500 lb. per sq. in. in the plate and across the weld. This would correspond to basing the working pressure for a factor of safety of five on a stress of 11,000 lb. per sq. in., which involves the use of a welded joint as strong as the plate. The proposed specifications base the working pressure on 80 per cent of this stress.

Arc-welded test shell No. 1 was welded with ordinary bare welding wire. After 5530 cycles of pressure, fracture developed in the welded seam at the junction of the base metal and weld metal.

Arc-welded test shell No. 2 was the first test shell made in the early stage of the development of the welding technique now perfected by the Babcock & Wilcox Company. A special welding rod was used. After 267,603 cycles of pressure, a fracture $1^1/_4$ in. long developed at the junction of base metal and weld metal near the middle of the length of the longitudinal seam. On examination of the fracture, it was found to extend through some slag inclusions in the weld. Strain-gage readings showed that permanent distortion in the vicinity of the point of failure began to be evident at 240,000 cycles of pressure.

Arc-welded test shell No. 3 was welded using a covered welding wire. After 417,381 cycles of pressure, the test shell fractured, not in the weld, but at the tapped hole where the pressure-gage connection was made. During the test, there were no signs of permanent deformation.

Arc-welded test shell No. 4 was also welded, using a B & W processed electrode. After 561,984 cycles of pressure, the shell cracked at the tapped hole for a gage connection in a manner similar to the failure of the forged shell and of arc-welded test shell No. 3. There were no signs of permanent distortion during the test.

Arc-welded test shell No. 5 was a duplicate of shell No. 4 and the same methods of welding were used. After 435,883 cycles of pressure, failure occurred at the gage connection as in the case of shell No. 4.

Arc-welded test shell No. 6 was welded using a B & W processed electrode, and to study further the effect of the gage connection, an unreinforced tapped hole for the hydraulic-gage connection was put through the weld metal. After 58,842 cycles of pressure failure occurred at this tapped hole. A comparison of this test with tests of shells 3, 4, 5, 7, and 8 shows clearly the disastrous effect of the tapped hole, and shows that such a "stress raiser" is especially dangerous in the weld metal.

Arc-welded test shell No. 7 had no tapped hole for a gage connection. It was subjected to 1,035,037 cycles of pressure without failure. After the test was finished, specimens were cut from the weld and showed no change in physical properties. Other specimens were examined carefully under the microscope. In one of the specimens so studied, a small blowhole was discovered, and what looked like a small fatigue crack about 0.02 in. long leading from it. Whether this crack would have eventually spread to cause failure is an interesting question. In any event, the shell stood many more cycles of stress than would be met with in the ordinary period of service of a boiler.

Arc-welded test shell No. 8 was without a tapped hole for the gage connection. It withstood 1,993,450 cycles of stress without failure, the range of stress being from zero to 16,500 lb. per sq. in. Then the pressure was increased so that the stress ranged from zero to 22,000 lb. per sq. in. At this increased range, the shell has withstood 24,400 cycles of pressure with no sign of failure; the test is still in progress.

In addition to the fatigue tests conducted on the test shells herein reported, there are eight additional test shells made by different methods of welding, which are now being subjected to the application of pressure. At present these test shells have not been subjected to a sufficiently large number of applications of pressure to embody the results in this paper.

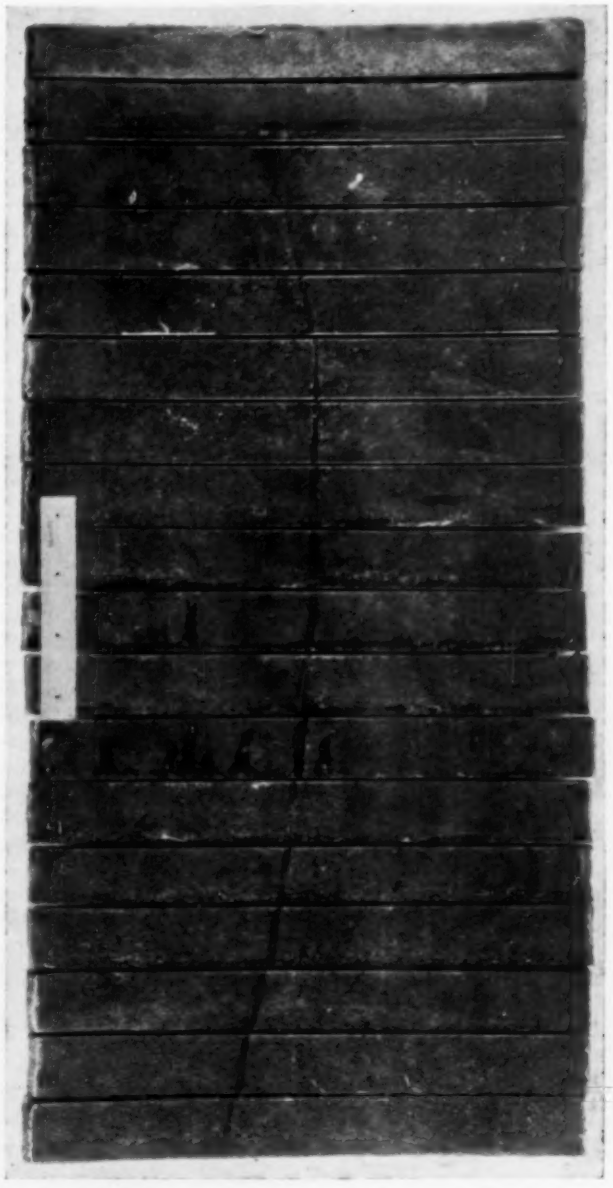


FIG. 7 DEFECT ON SURFACE OF PLATE WHICH CAUSED FAILURE
(Plate was cut into a number of test specimens, which were placed side by side.)

CONCLUSIONS

As noted in the introduction, it is not feasible to make fatigue tests of such large specimens as the full-sized test shell therein described, in the same manner in which fatigue test of small laboratory specimens are made. Endurance limits for the test shells have not, in general, been determined. However, certain

TABLE 1 SUMMARY OF TEST RESULTS

Specimen	Shell dimensions: Inside diam., in.	Plate thickness, in.	Longitudinal seams	Efficiency of joint, per cent	Max. allowable working pressure, lb. per sq. in.[a]	Range of test pressure, lb. per sq. in.	Range of tensile hoop stress	No. of cycles of pressure applied during test	Remarks
Manganese-steel test drum, riveted	35 5/8	1.0	2 at 180°	71	638	0–1260	0–31,500	130,225	Crack in drum head below manhole
A.S.M.E. standard riveted drum	42	2.0	One	80	838	0–1260	0–16,500	1,013,840	No fracture
Forged test shell	42	1 7/8	None	100	1047	0–1570	0–18,000	446,950	Failed at 1/8-in. tap hole for gage
Forge-and-hammer-welded shell No. 12	42	2.0	One	...	665	0–1570	0–16,500	890,000	No fracture; test still in progress
Forge-and-hammer-welded shell No. 13	42	2.0	One	...	665	0–1570	0–16,500 0–14,650[b]	611,164	Failed in plate away from weld at a longitudinal defect on the inside surface
Arc-welded shell No. 1	42	2.0	One	80	838	0–1570	0–16,500	5,530	In long. seam near middle of length at junction of base metal and weld metal. Bare welding wire used
Arc-welded shell No. 2	42	2.0	One	80	838	0–1570	0–16,500	267,603	Through slag inclusions in weld
Arc-welded shell No. 3	42	2.0	One	80	838	0–1570	0–16,500	417,381	Failed at 1/8-in. tap hole for gage
Arc-welded shell No. 4	42	2.0	One	80	838	0–1570	0–16,500	561,984	Same as for shell No. 3
Arc-welded shell No. 5	42	2.0	One	80	838	0–1570	0–16,500	435,883	Same as for shell No. 3
Arc-welded shell No. 6	42	2.0	One	80	838	0–1570	0–16,500	58,842	Tapped hole for gage located in weld; hole very irregular at inner surface; failure occurred at the tapped hole
Arc-welded shell No. 7	42	2.0	One	80	838	0–1570	0–16,500	1,035,037	No failure; no tapped hole in shell
Arc-welded shell No. 8	42	2.0	One	80	838	0–1570 0–2100	0–16,500 0–22,000	1,995,450 24,400	No failure; no tapped hole in shell No failure—supplementary test of shell No. 8; test still in progress

[a] Given by A.S.M.E. Power Boiler Code except for arc-welded shells where the maximum allowable working pressure is that given by the "Proposed Specifications for Fusion Welding of Drums or Shells for Power Boilers."

[b] Stress across welded seam, which is 1/4 in. thicker than plate.

conclusions may be reached as to safe values of stress for boilers and other pressure vessels.

The effect of local defects other than those inherent in the welds is emphasized by the failure of a number of test shells at the tapped hole for a gage connection, and also by the failure of shell No. 13 at a longitudinal defect on the inside surface of the plate well away from the weld. A quantitative value for the effect of such "stress raisers" cannot be given, but they evidently diminish very materially the strength of the plate.

The tests indicate that properly made arc-welded joints in drum shells have greater resistance to repeated stress than the plates themselves have when weakened by rather small surface defects or by tapped holes. The results obtained on arc-welded shell No. 6 indicate that a "stress raiser" such as a hole is particularly dangerous when located in the welded joint; however, the proposed specifications provide against this hazard by stating that no holes shall be located in a welded joint.

The tests also show that wide variations in fatigue strength will result from different types of welding. The necessity for the development of a good welding technique is obvious. The necessity for a rigid system of inspection of the finished welded vessel is also obvious.

In view of the fact that in its normal period of service, a boiler would be subjected to a few thousand cycles of repetition of pressure, together with a larger number of cycles of stress with a very small range, it would seem that all the drums and shells tested (with the exception of arc welded shell No. 1) might be expected to resist the action of repeated stress of the magnitude developed in the tests over a period longer than the normal period of service of a boiler.

In the proposed specifications for fusion welding of drums or shells of power boilers which have been published by the Boiler Code Committee the allowable working stress is give as 1/5 of 80 per cent (16 per cent) of the minimum of the specified tensile range of the plate used. This stress would be 53 per cent of the stress corresponding to the test pressure. A consideration of the test data, wherein properly welded test shells withstood without failure a very large number of applications of a high stress (16,500 lb. per sq. in.), should leave no doubt as to the safety in regard to failure due to service stresses of properly inspected welded boiler drums, designed under the proposed rules, wherein the expected service conditions represent but a few thousands of applications of a relatively low stress which is 8800 lb. per sq. in. as a maximum.

For most constructions the stress would be much lower than this due to the weld being subjected to a lesser unit stress than other parts of the drum.

Acknowledgments

The laboratory force at the Barberton works of the Babcock & Wilcox Company designed and developed the testing equipment described in this article and planned and carried out the tests herein reported. The author served as consulting engineer and cooperated in making the tests.

Discussion

H. E. Jenks.[1] So far as giving information as to the relative strength of plate and weld, tests on arc-welded shells 3, 4, 5, and 6 and on forge-and-hammer-welded shell 13 do not yield much of value. They do serve, however, a useful purpose in making clear the point that regions of stress concentration must be avoided so far as possible in designs where repeated stress is to be encountered.

The weakening effect of a tapped hole is well brought out here. For the case of unidirectional stress such as in these tests, the theoretical coefficient of stress concentration at the circumference of a small circular hole is 3; that is, the stress is three times that at points remote from the hole. Therefore, a failure at the hole instead of at the weld in such cases merely indicates that under fatigue conditions the weld is stronger than one-third the strength of the plate. It does not tell how much stronger.

Taking Professor Moore's figures of 16,500 lb. per sq. in. as the average stress in the arc-welded shells there is 49,500 lb. per sq. in. concentrated at the circumference of the hole. This value must be well over the endurance limit for the material of which the shells were made, as is in fact proved by the failures of shells 3, 4, 5, and 6.

Evidently reinforcement of the plate adjoining tapped holes is necessary to eliminate such points of weakness. The radius of reinforcement should be at least twice the radius of the hole in order to insure a strength in the material around the hole comparable with that of other parts of the design.

The failure of forge-and-hammer-welded shell 13 at an inside surface defect affords another instance of the effect of stress concentration where fatigue stresses exist. It has been shown

[1] Atmospheric Nitrogen Corporation, Hopewell, Va.

TABLE 2 TABULATION OF TEST RESULTS

No.	Type of shell or drum	No. of stress applications	Failure	Ultimate tensile strength, lb. per sq. in. Shell	Joint	Test pressure, per cent of ultimate Shell	Joint	Desirable endurance range for joint, per cent of ultimate
1	Manganese-steel drum, riveted	130,000	Manhead	105,000	....	30	..	..
2	Mild-steel drum, riveted	1,000,000	None	55,000	....	30	..	..
3	Forged seamless shell	450,000	Tapped hole	60,000	....	30	..	..
4	Hammer-welded shell	900,000	None	50,000	50,000	30	30	60
5	Hammer-welded shell	600,000	Surface defect	45,000	45,000	37	33	60
6	Fusion-welded shell, bare wire	5,500	Junction of weld	55,000	42,000	30	40	60
7	Fusion-welded shell, special rod	270,000	Junction of weld	55,000	42,000	30	40	60
8	Fusion-welded shell, covered wire	420,000	Tapped hole	55,000	42,000	30	40	60
9	Fusion-welded shell, "processed"	560,000	Tapped hole	55,000	42,000	30	40	60
10	Fusion-welded shell, "processed"	440,000	Tapped hole	55,000	42,000	30	40	60
11	Fusion-welded shell, "processed"	60,000	Tapped hole in weld	55,000	42,000	30	40	60
12	Fusion-welded shell, "processed"	1,000,000	None	55,000	42,000	30	40	60
13	Fusion-welded shell, "processed"	2,000,000	None	55,000	42,000	30	40	60
14	Fusion-welded shell, "processed"	+24,000	None	55,000	42,000	40	50	60

by experiment that accidental surface scratches may reduce the endurance limit of test specimens as much as 16 per cent, and the reduction of endurance limit in such specimens by V-shaped notches may be as much as 60 per cent. That this defect which was $^1/_{16}$ in. wide and $^1/_{32}$ in. deep must have produced heavy stress concentration is then evident. The need for careful inspection of finished pressure vessels, not only at joints but at all other points both inside and outside, is emphasized.

It is to be hoped that the tests on forge-and-hammer-welded shell 12 and on arc-welded shell 8 have been carried to failure, and that from these shells in which there are no tapped holes more specific information may be obtained as to the relative strength of plate and the two different kinds of welds when properly made.

R. E. Peterson.[5] The test data show that arc-welded shells are capable of withstanding a stress in the weld varying from zero to 16,500 lb. per sq. in. for at least two million cycles. From published test data[4,5] it may be expected that a stress much higher than 16,500 could be carried indefinitely as far as the welded joint is concerned. If the above mentioned data are plotted on a combined stress chart, the result is as shown in Fig. 8. If the line connecting endurance limit (s_e) and the

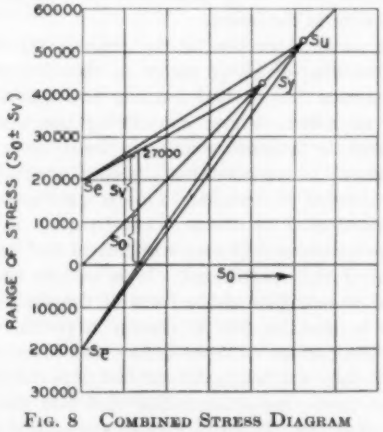


Fig. 8 Combined Stress Diagram

yield point (s_y) is considered a design criterion of failure[6] (which is shown by existing test data to be on the safe side), a stress varying from zero to at least 27,000 is necessary to produce failure. This would indicate that the margin of safety is still larger than indicated in the paper.

One further point may be of interest in this connection. In testing welded fatigue specimens $2^1/_4$ in. diameter, a considerably lower value of endurance limit has been obtained than for small specimens. No satisfactory explanation has been found for this as yet. Inasmuch as the test shells described in the paper were 2 in. thick, further research into the variable of weld size should be of value in interpreting data from tests of boiler shells.

J. M. Duncan.[7] The tests reported in this paper, or any other tests relating to welding and forging methods of steam-boiler fabrication, are of interest to the boiler user in the degree to which they reassure him as to the merits of the newer methods, as compared with the commonly accepted riveted joint or with the solid forged drum.

It is not likely that the results of these tests will contribute greatly to a more general acceptance of fusion welding as a safe practice for power boilers. Consider the accompanying tabulation of the results reported:

In Table 2 the writer has added in the last column what he would regard as a desirable endurance range which should be obtained with consistent uniformity in any weld metal, and in any junction of weld and base metal, before fusion welding can be regarded with confidence as compared with alternative methods available. Endurance range is not to be confused with endurance limit. The endurance limit assumed as attainable and desirable in estimating the endurance range for the particular stress cycle employed in these tests was plus or minus 40 per cent of the ultimate.

Presumably carried out to ascertain roughly the factor of safety of the fusion-welding method when applied to full-scale models, the tests appear to have demonstrated only the following:

a That any of the older techniques using bare electrodes, or any of the techniques which are at present available to the average boiler shop, are exceedingly dangerous.

b That special care must be used in any fusion-welded boiler to avoid surface or hidden defects, holes in the weld metal, or any other feature of design or construction which might result in unforeseen localization of stress.

c That the endurance range of a welded joint using electrodes prepared according to one proprietary formula may be greater than 30 per cent of the ultimate strength of the base metal and may be greater than 40 per cent of the ultimate strength of the weld metal.

While these results of the tests may appear meager and negative as a direct contribution to the knowledge of the art of welding on the part of the general public, they no doubt form a useful contribution toward the perfection of the technique of welding as developed by the company sponsoring the tests, and

[3] Westinghouse Research Laboratories, East Pittsburgh, Pa. Mem. A.S.M.E.

[4] Lobo, "Strength of Butt Welds," Jl. Am. Welding Soc., vol. 8, no. 4, p. 53. April, 1929.

[5] Peterson, "Fatigue Tests of Large Specimens," Trans. A.S.T.M., vol. 29, part II, 1929, p. 371.

[6] Soderberg, "Factor of Safety and Working Stress," Trans. A.S.M.E., 1930, APM-51-21-231.

[7] Petersburg, Va.

in this manner contribute to the benefit of the boiler user.

It is, however, somewhat disturbing to find the Boiler Code Committee putting forward a proposed code covering fusion welding of power boilers with so little published data available to members of the Society or to boiler users in general, technical or non-technical, upon which to form an intelligent opinion as to the dependability of this method. It is even more a matter of remark that no specification appears in the proposed code calling for strain annealing after welding, while the tests of the qualifying work of welders do not include any which would determine even the ductility of the metal.

Any welding method used in power boilers of conventional design should produce a ductile material in the weld and at the junction of weld and base metals. While endurance limits cannot be expressed in terms of ultimate strengths alone, it would be well to assume, in the present state of ignorance concerning fatigue properties of weld metals, that a safe weld metal should have ductility equal to mild steel, and an ordinary endurance limit in air equal to or greater than plus or minus 40 per cent of the ultimate tensile strength. For the test pieces prescribed by the Boiler Code Committee to be made up by fusion welders seeking to qualify for power-boiler welding, this endurance would correspond to plus or minus 16,800 lb. per sq. in., or an endurance range of 33,600 lb. per sq. in. for a purely alternating stress cycle.

Since the stress cycle used in these tests is necessarily a pulsating and not an alternating one, the endurance range for this particular stress cycle should lie above plus 25,000 lb. per sq. in. minus zero inches and probably could be plus 35,000 lb. per sq. in. minus zero inches.

It appears evident, therefore, that in testing these shells through a range of only 16,500 lb. per sq. in. any failures obtained would indicate that the particular technique used in the weld was exceedingly unreliable and dangerous, while the few cases where no failure was produced offer only negative evidence of little actual value.

It is recommended that, before the present or any future proposed code on fusion welding is indorsed, a series of tests, similar in scope and character to those completed by the author of the paper, be carried out. The recent work of McAdam, demonstrating the great influence of certain types of waters in lowering the endurance limit, compels the Boiler Code Committee and the Society to protect the boiler user by including a most searching investigation of the corrosion-fatigue properties of weld metals in any research which is to be the basis for any proposed fusion-welding method for power boilers. The Society and the Committee cannot escape responsibility, if a welding method should be sanctioned which might prove unreliable under any of the numerous boiler-water conditions which may arise in actual operation, whether such operation is conducted with care and intelligence or not.

J. C. Hodge.[8] All of the arc-welded drums were subjected to a stress-relieving operation by heating the finished welded test shells to 1200 deg. fahr., holding them at that temperature for one hour per inch of thickness, followed by cooling in a furnace. In this way all residual stresses set up by the welding operation were fully relieved.

A question has been asked as to the effect of a tube hole within the welded joint on the resistance of a welded drum to repeated applications of pressure. At the present time there are no data available on this condition, but a test shell is now in process in the Barberton shops, this test shell having been constructed to answer this question. Standard tube holes are being placed at standard intervals along the longitudinal welded joint of the test shell and also along an element of the cylinder in the boiler plate. Blind nipples will be rolled into the tube holes several different ways and the test shell then subjected to repeated applications of pressure. The results of this test will undoubtedly be published some time in the near future.

A question has also been asked as to why failure under repeated applications of pressure did not occur through the rivet holes in the riveted test drum of standard A.S.M.E. construction. If failure occurred through the tapped hole for the hydraulic gage connection in several of the test shells after approximately 500,000 applications of pressure, then it should be expected that failure should have occurred through the rivet holes in the riveted test drum after 1,000,000 applications of pressure, whereas no actual failure was found in the riveted test drum after this large number of applications of pressure. Prof. Moore has stated that the absence of failure through the rivet holes was probably due to the pressure on the rivet heads equalizing the stress and distributing the stress around the rivet holes. However, the predominating factor in this connection is the fact that the butt strap and boiler-drum sheets are pressed closely together by the tension in the hot-driven rivets, and a considerable portion of the load on the riveted joint is sustained by the friction between the two plates, thus reducing the stress around the rivet holes.

J. C. McCabe.[9] In view of the fact that there is a high-pressure boiler to each 200 of population the reasonable safety of steam boilers is of vital importance. From the date of the organization of the Manchester Steam Users Association on Feb. 14, 1855, to the present time, much study and speculation has been given to problems relating to steam boilers.

It seems natural to ascribe mysterious reasons for many of the ills of steam boilers. Robert Bewick Longridge, the first Chief Inspector of the Manchester Steam Users Association, attributed the many boiler explosions in and around Manchester to the formation of "an unknown explosive gas which formed within the boiler and exploded with violence and disaster." The questionable quality of the slag-laden iron plate used did not attract attention. Little attention was paid to the manner of using the plate in the boiler.

The paper calls attention to the present-day tendency to ascribe the cracking of boiler plates to chemical action rather than high stresses due to bad forming and loading the plate transversely as rolled without considering the actual tensility of the plate and the fatiguing existing in the structure.

It is customary to consider that the shell or drum of a boiler is uniformly loaded in tension. This is practically true in a correctly formed shell or drum. Unfortunately in the general run of shop practice the importance of correct and true forming is not appreciated and understood. It is seldom that a correct checking and observation of the form of the shell or drum are made. The hydrostatic test is usually a perfunctory gesture without an observation of form before and after test. If the shell or drum does not fail under the test, it is considered a safe article of commerce, notwithstanding that bad forming has as definitely limited the life and utility of the article as disease in the human.

In an investigation made by the writer recently on the failure by cracking at the ligament of the longitudinal joint of drums 60 in. diam., 24 ft. long, and made from $1^{5}/_{16}$-in. plate, radii under the strap were found varying from 28 in. to over 68 in. The normal external radius was $31^{5}/_{16}$ in.

It must be evident that in the region of the 28-in. radius the inside of the plate would be in tension when riveted up and that where the external radius was 68 in. a heavy tension load would

[8] The Babcock & Wilcox Co., Barberton, Ohio.

[9] Chief Boiler Inspector, State of Michigan, Detroit, Mich. Mem. A.S.M.E.

be on the outer fiber and a corresponding compression load on the inner surface with no load at the neutral axis of the plate. In this case the stresses in the outer surfaces would be about 13,000 lb. without internal pressure. It is evident that the internal pressure would greatly increase the already overloaded plate and result in an early and progressive cracking of the plate. The thought of a large shell or drum under high pressure, with 57 varieties of radii at the longitudinal joint and elsewhere, is not pleasant to contemplate, but it is more common than realized.

The A.S.M.E. Boiler Code gives no definite instruction or rule for the proper forming of boiler shells or drums. With the templets ordinarily used by the inspectors a rather "tolerant" view is taken as to how close the plate should be made to conform to the templet. When the hydrostatic test is made it is a rare occasion to make measurements before and at the time of the hydrostatic test to determine the evidence of bad forming and the inevitable internal stresses. The A.S.M.E. Boiler Code should definitely require the systematic observation of the forming and the behavior of the various areas under pressure.

The fact that rivets along calked joints and the straps are stressed to the yield point is well understood and needs no comment. When a plate is loaded transversely as rolled the tension tests should be made in the same direction. The cracking at the gage-pipe hole indicates that a deflection occurred at this point which threw the outer fiber into tension with the inner surface in compression. This behavior changed the affected section from direct and uniform tension to a loading comparable to a beam.

The writer favors the approval of fusion welding under proper control as to procedure. This will of course include correct and definite limits as to forming and annealing. This will be a worthwhile achievement for the engineer. The same consideration should be given the welded boiler that is given the riveted structure. The welded boiler can be made much safer and desirable than the riveted boiler.

A. B. Kinzel.[10] The tests in this paper are most illuminating, and the author and his collaborators in this work are to be both congratulated and thanked for presenting these results. There are several items of interest, particularly if the work is considered from the point of view of the proposed specifications for fusion welding of drums or shells of power boilers. It would indeed be very interesting to know whether or not any or all of these shells had been subjected to non-destructive tests before, as well as after, the repeated-pressure test. It would also be interesting to know the cause of the failure of the bare-wire arc-welded shell 1. Was there a defect in the weld? Was there lack of fusion at the base, or was the phenomenon connected with the physical characteristics, that is, tensile strength and bend ductility of the weld metal in question? In other words, could this failure have been predicted?

In connection with the manganese-steel test drum, was the head and manhole reinforcement designed according to more recent ideas on this subject?

The conclusions to be drawn from the paper, aside from the very important ones which the author has stressed regarding the seriousness of unreinforced openings in the shell and defects in the plate, would seem to bear on the application of this test to service. As the author very rightly points out, this is probably a great deal more severe than a similar stress variation in service. For example, arc welded shell 2, with 267,000 cycles and a fiber-stress variation of 16,500 lb. per sq. in., would probably have shown a life in service far beyond the expected average of 20 years. It seems that any test of this kind, if applied as a routine test, would merely have to differentiate between the type of construction which fails in 5000 cycles as against some hundred thousand cycles. If the cause of failure at a low number of cycles can be determined by other tests, such as the non-destructive test or analysis of the physical properties of the material, the use of such a test in routine application would not be warranted. This is particularly true if it be considered that many believe that the repeated-stress test may have a deleterious effect on the drum.

C. S. Gladden.[11] Ordinary riveted construction, when calked on the outside only and when in service with boiler water having low ratios of sulphate to carbonate alkalinity, is the source of many failures from embrittlement of the steel. If the riveted joints are calked on the inside and the spaces between the butt straps and the main plate can be definitely sealed, most of the difficulty from embrittlement may be eliminated. However, if boiler drums can be constructed without riveted joints and butt straps, that is, by the method of forge-and-hammer welding or electrical-fusion welding with the joint equally as strong as the plate, nearly all embrittlement troubles would be eliminated, regardless of the water condition, and the ideal boiler drum would be developed.

There is one point, however, concerning which additional information is needed; that is, concerning the effect of rolling tubes into holes in the drum. Have any tests been made of repeated pressure on drums in which blank-ended tubes have been installed? A number of the fractures, especially in the arc-welded shells as given in the table of this paper, occurred at the tapped hole for the gage or in the region where the "stress-raiser" was located. Does the nature of the tapped hole with sharp grooved edges influence these failures more than a round hole of larger size and subjected to the additional stress due to the rolling in of the tube?

No drum, however perfect it may be, is of practical use without having tubes rolled into tube holes cut through the sides of the drum. The writer is therefore interested particularly in what results are obtained with shells under these conditions.

Author's Closure

Several of the discussions of the paper point out important questions which have not been settled by the tests herein reported. That there are many things as yet unknown about fusion welds is of course quite true. The author would like to comment on the statement of Mr. Duncan that "It is not likely that the results of these tests will contribute greatly to a more general acceptance of fusion welding as a safe practice for power boilers." Whether these tests will contribute "greatly" to the acceptance of fusion welding is a matter for the future to decide. However, it seems that in his discussion Mr. Duncan has overlooked the fact that for boiler shells subjected to only a few thousand cycles of stress during their period of service, the *practical limit of strength* may be appreciably higher than the endurance limit based on tests of tens of millions of cycles of stress. As the author has frequently pointed out, the numbers of cycles of stress above the endurance limit which a metal will endure without failure is no reliable criterion of its fatigue strength under millions of cycles of working stress. However, when a boiler shell withstands 100 times the number of cycles of stress it will be called upon to stand in actual service, it would seem to be reasonably safe so far as mere fatigue strength is concerned. Of course, conditions of boiler water which may lead to corrosion bring in a very different problem, both for the weld material and for the plate steel. The author believes that the tests do show that, so far as fatigue strength is

[10] Union Carbide and Carbon Research Laboratories, Long Island City, N. Y.

[11] Director, Power and Maintenance Section, General Motors Corporation. Mem. A.S.M.E.

concerned, boilers welded by modern, closely controlled methods will stand the cycles of stress which they meet in ordinary service under ordinary allowable working stresses. The author does believe that, as pointed out in discussions, the tests show that there is a wide range of strength for different techniques of welding.

Some of the figures given in Table 2, which was prepared by Mr. Duncan, and some of his statements are not correct. The tensile strength of the joints which were fusion welded by the "processed" electrodes is given in the table as 42,000 lb. per sq. in., as compared to 55,000 lb. for that of the shells, whereas the strength of these fusion welded joints was greater than that of the shell. His statement that it is even more a matter of remark that no specification appears in the proposed Code calling for strain annealing after welding is also incorrect, as can be seen on examining the Proposed Specifications for Fusion Welding of Drums or Shells of Power Boilers published in the March, 1930, issue of "Mechanical Engineering." These Proposed Specifications also include the requirement that the minimum elongation of the outside fibers of the weld metal in bend test specimens shall be 30 per cent, whereas his remarks might make it seem that there are no requirements for ductility.

Mr. Duncan's principal criticism is that the testing of the shells through a stress range of 16,500 lb. per sq. in. did not give conclusive results. This stress is 50 per cent higher than that which would exist in the most highly strained parts of a boiler drum constructed in accordance with the A.S.M.E. Boiler Code, which stress comes in the ligaments between tube holes and between rivet holes. As the proposed Code does not sanction the placing of holes in the welds, the stress in the welds in a boiler drum would be considerably lower than the maximum stress in other parts of the drum. The allowance of 80 per cent for the efficiency of the joint given in the proposed rules for fusion welding results in the tests being made at pressures of from two to over three times the strain on the welded joint for boiler drums of the usual construction when pierced with holes, which has been criticised as being too high rather than too low for a test pressure. As all but one of the failures with the shells welded with the "processed" electrodes were in the solid part of the plate and not in the weld, Mr. Duncan's criticism, if valid, would apply to the plate much more than to the welds. The plate was firebox steel which met the requirements of the A.S.M.E. Code so that his criticisms in reality apply to a material which is generally recognized as being suitable for the use to which it is put and not to fusion welding.

Concerning the effect of stress concentration of holes, fatigue tests show that the actual weakening of steel by such "stress raisers" is less than the theoretical, but that the weakening due to the holes, nicks, and scratches is very pronounced, sometimes producing an effective strength concentration of twice the computed stress.

Mr. Gladson asks an interesting question about the effect of rolling tubes into the holes in the drum, and the author would refer him to Mr. Hodge's discussion of this paper.

Some of the discussions emphasize the need of fatigue tests of welds to insure the proficiency of the operator. Fatigue tests of large specimens, either on flat plates bent back and forth or on turned round specimens, can be made without difficulty. However, the author believes there is still greater need for the development of some non-destructive test to detect incipient fatigue cracks in actual seams. He hopes that the simple stethoscope tests proposed by Mr. Kinsel, the heating test used in some welding shops, and the elaborate Sperry test and X-ray tests will all be studied. A test which has occurred to the author, but which he has had as yet no opportunity of studying, consists in measuring the minute change in electrical resistance in a piece of metal between a stressed condition and an unstressed. Possibly an incipient fatigue crack might cause abnormal change in this resistance.

FSP-53-7

The Dry Quenching of Coke

By WILLIAM O. RENKIN,[1] NEW YORK, N. Y.

This paper describes the process of dry quenching or cooling of coke with automatically formed inert gases, the sensible heat in the glowing coke being used to heat the cooling gases, which in turn pass through boilers, producing steam at useful pressure and economic cost without the consumption of any other fuel. Formulas, curves, and data regarding heat available and method of calculation are given, together with descriptions of installations and illustrations of plants in operation.

A detailed economic study is given which shows the savings to be expected and the approximate cost of a dry-quencher plant to cool 1500 net tons of coke per day; and composite charts giving actual operating records as well as output data and the cost of power for one plant by months for the year 1929 are included.

Analyses of coal and coke are given which show conclusively that dry-quenched coke carries less dust than wet-quenched coke.

Data giving actual results obtained by using dry-quenched coke in blast furnaces, and the advantages of substituting dry-quenched coke fines for high-priced bituminous gas coal in gas producers, are discussed in considerable detail.

A dry-quencher plant at a power station in Flint, Mich., the latest to be installed, is described in some detail and the principal technical data given.



WHILE there have been many articles published on this subject and much literature is available describing the operation of dry-quencher installations, this paper will present some new economy data, as well as a description of the latest plant, located at Flint, Michigan.

Dry quenching of coke, or more correctly speaking, the cooling of coke without the use of water, has been brought about by the need of coke plants to eliminate the operation difficulties and dust-laden vapor clouds so common to water-quenching installations, as well as by their desire to reduce operating expenses and make use of the sensible heat in the glowing coke.

The process of dry quenching of coke may be briefly described as a circulation of cooled inert gases by means of a fan, upwardly through a container intermittently charged with hot coke at the top, and from which the cooled coke is intermittently withdrawn from the bottom.

The rising inert cooling gases absorb heat from the slowly intermittently descending coke mass, and these heated gases are thence passed through a waste-heat boiler for cooling, and then through the fan for recirculation to the coke container again, the process being continuous as well as remarkably simple.

Dry quenching of coke is as great an improvement over wet quenching as the modern by-product coke plant is over its predecessor, the beehive oven.

Dry quenching not only produces a better coke, but reduces the quantity of fines and produces them in a more commercial form. Dry-quenched coke is stronger, resists further handling better without degradation, screens more easily, and gives a cleaner product than wet-quenched coke, and gives superior results in producer and water-gas machines.

Dry Quenching an Economic Necessity to the By-Product Coke Industry

Throughout all industry there is a strong trend toward conservation and the elimination of waste. Almost every one who has witnessed the great clouds of steam produced by the wet quenching of coke has wondered if it were not possible to stop this seemingly needless waste of heat. A few figures are sufficient to demonstrate clearly what a tremendous waste of energy wet quenching represents.

In the United States alone 53,475,481 net tons of by-product coke were produced in 1929. Assuming an average temperature of the hot coke going to the quencher as 1800 deg. fahr., there can be recovered approximately 480 B.t.u. per lb. of coke by dry quenching, which means the possible production, utilizing the available sensible heat in the coke, of practically fifty-two billion pounds of steam per year from and at 212 deg. fahr. In other words, a waste of fuel equivalent to two and one-half million net tons of 13,500-B.t.u. coal in boiler plants operating at an overall efficiency of approximately 75 per cent is the result of wet quenching.

The entire steam consumption of a standard coke-oven installation may be secured from the waste heat of the glowing coke alone, and in most instances an excess of energy is available which may be diverted to other departments. An analysis of this situation provided the incentive to seek a solution of the problem.

The dry-quenching process developed by Sulzer Brothers represents a most efficient and reliable method of reclaiming the sensible heat of glowing coke. Numerous installations have proved the correctness of the process by years of continuous satisfactory service.

Description of the Process

In the Sulzer system of dry quenching, the cooling is accomplished by circulating the inert gases in a closed cycle, first through the glowing mass, where the gases extract the sensible heat from the glowing coke, thence through a steam boiler where the hot gases transfer the heat to the water in the boiler circulation and convert it into steam.

Referring to Fig. 1, the glowing coke is conveyed to the sealed cooling chamber, and the air of the system, including that en-

---

[1] Engineer, Dry Quenching Equipment Corporation. After leaving school Mr. Renkin was connected with various iron and steel plants, and was resident engineer for the organization and construction of the steel works for Tata Iron and Steel Company, Sakihi, Bengal, India. Returning to the United States he spent a year in special work and some time as chief engineer for A. M. Byers Company, Pittsburgh. He had done much work in the development of pulverized-coal equipment in the metallurgical field. Mem. A.S.M.E.

Contributed by the Fuels Division and presented at the Semi-Annual Meeting, Detroit, Mich., June 9 to 12, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

trained in the interstices of the coke mass, is put into circulation by means of a fan.

When the system is first started, a small portion of the coke is converted into a mixture of carbon monoxide and carbon dioxide, the percentage of each depending on the temperature at that particular time. However, the oxygen of the encased air is quickly consumed, and a mixture of inert non-combustible gases remains. This mixture of gases, in passing through the glowing coke, absorbs its heat, which is carried to the adjoining steam generator. The cooled gases leaving the boiler are conducted back to the fan and again forced through the glowing coke. The coke is drawn off from below, which is the coolest zone, and fresh charges of glowing coke are introduced from above.

In this manner continuous performance is established and, by regulating the fan speed, a relatively close relation may be maintained between the cooling of the coke and the generation of steam.

The possible loss of coke through burning has been determined

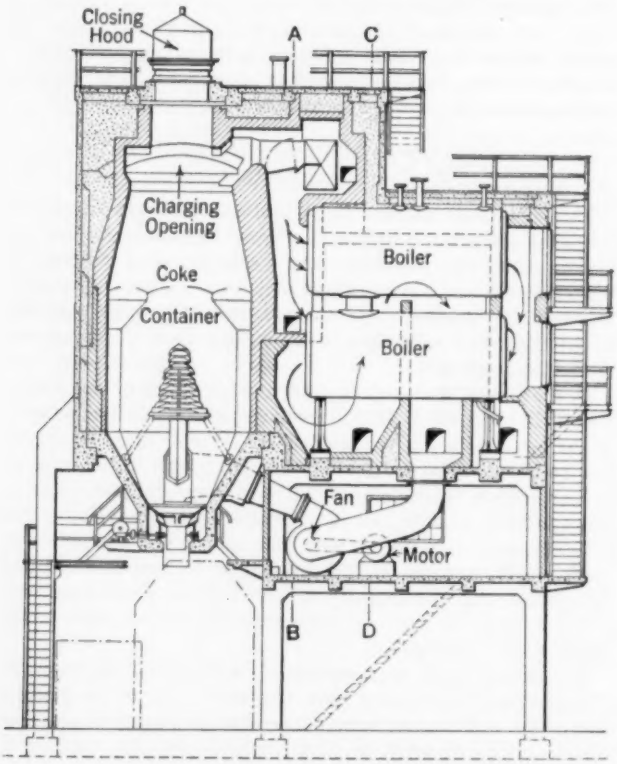


FIG. 1 EARLY TYPE OF INSTALLATION COMPOSED OF COOLING CONTAINER UNIT WITH TWO FIRE-TUBE HORIZONTAL BOILERS

by numerous tests to average 0.04 per cent, a quantity practically negligible, even if greatly increased.

PHYSICAL ELEMENTS OF THE PROCESS

Having adopted the idea of utilizing the waste heat of glowing coke, the question arises as to the actual amount of heat contained therein. This quantity of heat is determined by the weight, temperature, and mean specific heat of the coke. Through the investigations of Otto,[2] Schlapfer and Debrunner,[3] *Stahl und Eisen*,[4] Wöhler,[5] *Glückauf*,[6] Terres and Schaller,[7] *Gas und Wasserfach*[8] the specific heat of coke has been established as shown in Fig. 2.

The divergences of the different values are due to differences in methods of measuring and to the variance in the ash content. In the chart, therefore, the curves of the mean specific heat of graphite,[9] i.e., practically pure carbon, and of quartz[10] have

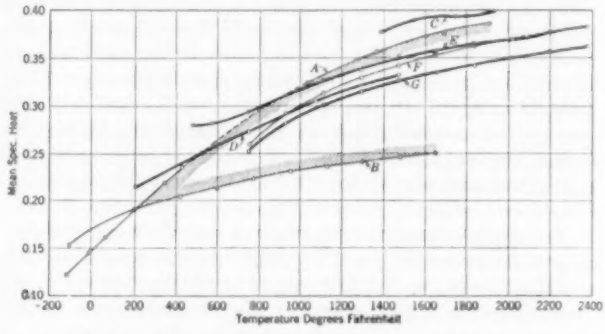


FIG. 2 MEAN SPECIFIC HEAT OF GRAPHITE, QUARTZ, AND COKE

A—Graphite referred to 32 deg. fahr. with range to 523 deg. fahr. and referred to approximately 72 deg. fahr. with range above 1034 deg. fahr.[9]
B—Quartz referred to 68 deg. fahr.[10]
C—Coke[2]
D—Coke with 10.5 per cent ash referred to 64 deg. fahr.[4]
E—Coke with 9.45 per cent ash referred to 68 deg. fahr.[6]
F—Coke with 15 per cent ash referred to 70 deg. fahr.[5]
G—Coke with 30 per cent ash referred to 70 deg. fahr.[6]

been plotted, assuming that the greater part of the coke ash is composed of silica. The lines for graphite and quartz therefore represent value limits for coke with 0–100 per cent of ash, between which are to be found the values of the other species of coke.

TABLE 1 MEAN SPECIFIC HEAT OF COKE
(According to Wöhler)

Range of temperature		Mean specific heat
Deg. cent.	Deg. fahr.	C_m
20– 250	68– 482	0.2798
20– 300	68– 572	0.2810
20– 400	68– 752	0.2938
20– 500	68– 932	0.3107
20– 600	68–1112	0.3269
20– 700	68–1292	0.3381
20– 800	68–1472	0.3528
20– 900	68–1652	0.3582
20–1000	68–1832	0.3652
20–1100	68–2012	0.3695
20–1200	68–2192	0.3762

TABLE 2 MEAN SPECIFIC HEAT OF COKE
(According to Terres and Schaller)

Range of temperature		Mean specific heat
Deg. cent.	Deg. fahr.	C_m
21– 250	70– 482	0.230
21– 422	70– 792	0.267
21– 506	70– 943	0.292
21– 609	70–1128	0.315
21– 708	70–1306	0.331
21– 811	70–1492	0.345
21– 912	70–1674	0.354
21–1009	70–1848	0.361
21–1107	70–2025	0.369
21–1297	70–2367	0.376

Schlapfer and Debrunner[3] have found that the specific heat of coke depends also upon the content of volatile constituents, and they have given the following formula to calculate the mean specific heat of coke of different composition:

$$C_m = \frac{X}{100} C_a' + \frac{Y}{100} C_k' + \frac{Z}{100s} C_g'$$

where X = content of ash in per cent
Y = content of fixed carbon in per cent

[2] C. Otto, Dissertation, Breslau, 1914.
[3] Schlapfer and Debrunner.
[4] *Stahl und Eisen*, 1922, p. 1270.
[5] Wöhler.
[6] *Glückauf*, 1922, nos. 14 and 15.
[7] Terres and Schaller.
[8] *Gas und Wasserfach*, 1922, pp. 732 and 821.
[9] Weber, *Poggedorfs Annalen*, 1874, vol. 154, p. 408.
[10] Stierlin Quarterly bulletin of Naturforschende Gesellschaft, Zurich, 1907, p. 411.

Z = volatile constituents in per cent by weight
C_a' = mean specific heat of ash (quartz)
C_k' = mean specific heat of graphite
C_g' = mean specific heat of gas per unit volume
S = specific gravity of gas (about 0.45).

Assuming the initial temperature of glowing coke to be 1800 deg. fahr., and the final temperature after cooling to be 400 deg. fahr., the amount of heat in B.t.u. utilized equals

$$1800 \times C_{m1800} - 400 \times C_{m400}$$

The mean specific heat, according to Table 2, equals 0.361 between 70 and 1800 deg. fahr. and 0.230 between 70 and 400 deg. fahr. Thus $1800 \times 0.361 - 400 \times 0.230 = 558$ B.t.u. gained per pound of coke, which is equivalent to approximately 1,120,000 B.t.u. per ton of coke.

These calculations show that steam is generated by utilizing the sensible heat of the hot coke and not by burning the coke as fuel.

Fig. 3 gives the specific heat and heat content in B.t.u. per pound at various temperatures, and has been made up from the data available and found very useful. It will be noted from this chart that the heat content in coke of an average analysis (using curve 2) at 1800 deg. fahr. is 640 B.t.u. per lb. As experience has shown that the most advantageous temperature for discharging the dry quenched coke is 700 deg. fahr., it will be found from the chart that at this temperature there is left in the coke 194 B.t.u. per lb., or a heat available for useful recovery of 446 B.t.u. per lb.

However, it has been found that there is a certain additional amount of heat secured, from combustion of H_2 yielded by green ends and incompletely distilled coke, often amounting to more than 60 B.t.u. per lb. It is therefore conservative to add this to the heat recoverable, which gives a total heat available for recovery amounting to 506 B.t.u. per lb. of coke dry-quenched.

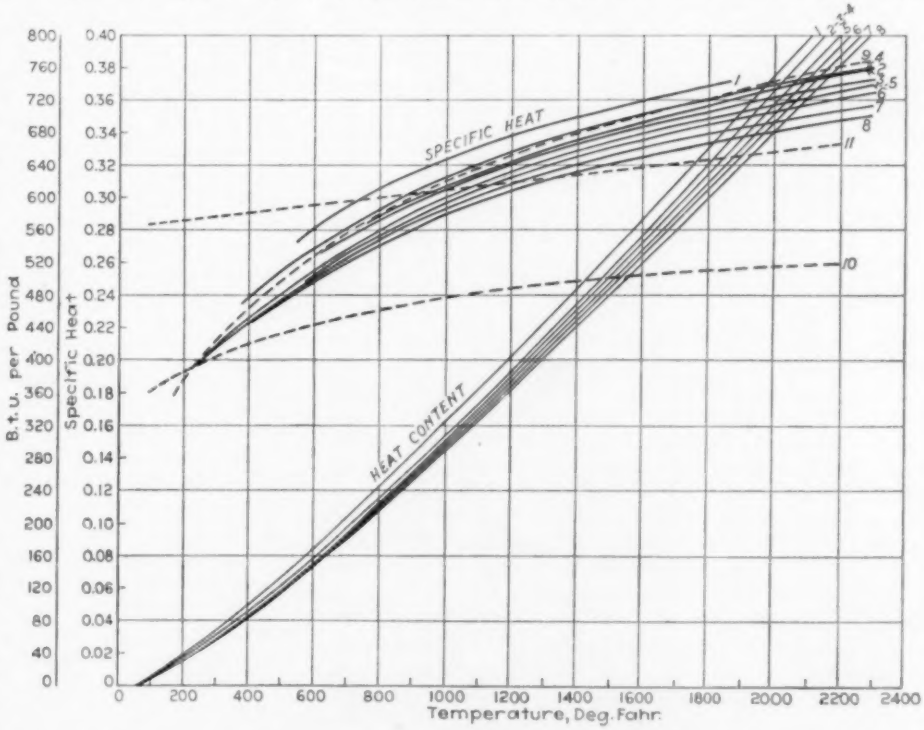


FIG. 3 SPECIFIC HEAT AND HEAT CONTENT OF GRAPHITE, QUARTZ, ASH, AND COKE FOR VARIOUS TEMPERATURES

1—Durham coke—Ungassed—Schlapfer and Debrunner
2—Coke with 87.3 per cent C., 10.6 per cent ash, 1.05 per cent gas } computed
3—Coke with 81.94 per cent C., 15.75 per cent ash, 2.31 per cent gas } by
4—Coke with 86.37 per cent C. 10.45 per cent ash, 3.28 per cent gas } formula
5—Coke with 5-10 per cent ash
6—Coke with 15 per cent ash
7—Coke with 20 per cent ash
8—Coke with 25 per cent ash
9—Ceylon Graphite—Carbon
10—Quartz—Ash
11—Gas—H_2 sp. ht. per cu. ft. per 1°F., at atmos schule

Determination of the mean specific heat (C_m) of coke at temperature (t) from proximate analysis:

$$C_{tM} = KC_{tK} + AC_{tA} + G/SC_{tG}$$

C^t = sp. ht. at given temperature of element
K = per cent fixed carbon
A = per cent ash
G = per cent volatile matter—gas
S = sp. gr. of gas about 0.45.

VALUE OF DRY COKE

The greater efficiency of a fuel free from moisture is becoming more and more appreciated. If a coke contains 10 per cent of water, this water has to be evaporated before it can be ignited. The coke is thus not only inferior because of the smaller percentage of carbon, but its heat is absorbed in evaporating the water and raising the vapor to the temperature of the waste gases.

For instance, if the coke is used to heat retorts and the products leave the regenerator at 600 deg. fahr., we obtain the following calculated results per kilogram of coke containing 15 per cent of ash on the dry sample:[11]

	Dry coke	Wet coke
Carbon, per cent	85	76.5
Ash, per cent	15	13.5
Water, per cent	..	10.0
	100	100.0
Heat of combustion of carbon, kg-cal. per kg	6868	6181
Less to evaporate water from 60 deg. fahr.	..	62
Less to raise vapor to 600 deg. fahr.	..	16
Available heat, kg-cal. per kg	6868	6103

This shows a depreciation of 11 per cent due to the presence of the water.

It may therefore be roughly stated that the heating value of coke increases about 10 per cent more than the percentage of moisture removed, or in other words, that the moisture-free coke provides 11 per cent more available heat than coke containing 10 per cent of moisture.

[11] John P. Leather, M. Inst. C. E., in *Gas Journal*, June, 1927.

Coke in Blast Furnaces

Dry-quenched coke being of more uniform size and devoid of moisture, carries with it less fines and breeze.

Experience shows that fine coke and dust reduce the capacity of the blast furnace, since more resistance is offered to the blast. Further inconvenience arises from the formation of dust whirls and dust pockets. The steel works of the Compagnie des Forges et Aciéries de la Marine et d'Homécourt, in Homécourt, France, reported that by the use of dry-cooled coke more regular and improved action of the furnace was evident, and that trouble caused by hanging of the load in the furnace was greatly reduced. The saving of coke in this furnace after the introduction of dry-cooled coke was 4.17 per cent.[12,13] The dry-cooled coke is of more uniform size. The large pieces, already split on account of the contraction which takes place during the process of distillation, break up to a certain extent when passing through the container. The remaining pieces are of somewhat smaller size, but very much stronger and stand further handling better than wet-quenched coke. Therefore, in going down the cupola of the blast furnace, dry-quenched coke does not break up under the heavy load of the charge as is the case with wet-quenched coke. The formation of dust and breeze pockets in the blast furnace is thus prevented, and a more regular distribution of the blast through the charge is secured.

It is well known that blast-furnace gas contains a considerable quantity of fine dust, both of ore and fuel, which must be removed by expensive gas-cleaning machinery for most needs; and as dry-quenched coke carries less dust and fines, a considerable reduction in the dust content of the gas produced can be expected.

Long-period tests on blast furnaces have been made in German steel works with coke that was purposely subjected to mechanical handling and then was screened. The best results were obtained with coke of medium and uniform size.

Edgar C. Evans[14] in a lecture in England before the National Federation of Iron and Steel Manufacturers, showed that after the elimination of 6 per cent of small coke and dust, the capacity of the furnace increased by 20 per cent. This fact is now well known, and the relation between the percentage of fines and furnace capacity has been established.

The advantages of dry coke for this work are evident, as dry-quenched coke, being devoid of all water, shows a higher thermal value; therefore the use of dry-quenched coke is equivalent to a saving of fuel.

In blast-furnace operation a considerable amount of cooling water is required for the condensation of steam in the blast gases, which under certain conditions must be filtered and recooled. As the water content of the blast gases reduces the heating effect, due to lowering the flame temperature, it should be eliminated, although sometimes water is added for special ores.

It is also a well-known fact that variations in the percentage of moisture of wet-quenched coke cause difficulties in the blast furnace, or in any other type of furnace where coke is used in steady operation. As dry-quenched coke is free of moisture, there is no variation in the quality of the fuel.

The water contained in the wet-quenched coke effects a precooling of the hot blast gas. This difficulty has not been observed at any furnace where dry-quenched coke is used. If a special condition be encountered where the absence of the water should cause faulty conditions, it can easily be remedied by sprinkling the coke with water according to requirements at the place where it is used.

It is becoming more and more important to have the accurate weight of coke charged to blast furnace, and dry quenching eliminates the inaccuracies caused by variation in moisture content so common to wet-quenched coke, and therefore makes possible the accurate records desired.

Sulphur Content of Coke

The question of sulphur content has been explicitly answered by analyzing coke at the mines of Mathias Stinnes, Germany.[12,13] The average of ten samples of coke showed a sulphur content of 0.914 per cent for dry-quenched coke and 0.857 per cent for water-quenched coke, a difference of 0.057 per cent. With water quenching, a desulphurizing action amounting to 6.4 per cent was obtained.

This difference is negligible in the practical use of coke, as it is noticeable in the second decimal only. It may be added that only a portion of the sulphur content of the coke is absorbed by the charges—for instance, only 30 per cent in the foundry and only 50 per cent to 60 per cent in the blast furnaces, which further proves the actual insignificance of this small difference. It was further shown in tests at the Stinnes Mines that only that part of sulphur was extracted which exists in the coke in the form of sulphides and which after quenching evaporates as sulphureted hydrogen or during combustion as sulphurous acid. It is, however, a well-known fact that in the course of time the vapors from wet quenching have a marked destructive effect on iron and concrete, aside from the decided hygienic disadvantage and inconveniences which they cause.

Capacity for Reaction

The capacity for reaction (combustibility) of dry-quenched as compared with water-quenched coke has also been carefully tested. The results showed, it is true, that the combustibility of dry-quenched coke was somewhat less than that of wet-quenched coke; this difference was noticeable, however, only in the finer granulations (up to $^1/_{16}$ in.) of both kinds of coke, and became less and less apparent with the increasing size of the granules. Since the difference diminished with the increased size of granulation, it may be assumed that in practice this difference will be negligible and that it will have no bearing on combustibility, whether the coke be dry- or wet-quenched.

In the laboratory tests both the influence of granulations and temperatures were examined, but for a final judgment of the capacity for reaction, the influences of the kind of furnace, air velocity, chemical composition, porosity, etc., will have to be investigated. The solution of the problem of combustibility by laboratory tests only cannot be expected. It is significant that no difficulty in this respect has been experienced by furnaces using dry-quenched coke.

Coke Fines

In general, the coke fines are used as boiler-plant fuel by burning on Coxe or other similar-type specially built stokers.

Very little successful work has been done along the lines of burning the wet-quenched coke fines for other purposes, owing to the difficulty of packing and choking, although some success has been attained by using small coke in gas producers, but this has not proved very commercial due to the necessity of using only sizes over a $^5/_8$-in. screen.

However, the dry-quenched coke fines have proved a very satisfactory fuel in gas producers and are being substituted for high-grade bituminous gas coal or larger sizes of salable coke.

A comparison of bituminous-coal producer gas and dry-quenched coke fines gas gives the following.

---

[12] Dr. Mueller, "Comparative Investigation of Dry- and Wet-Quenched Coke."

[13] *Glückauf*, 1926, no. 35.

[14] *The Gas World*, June, 2, 1926.

From the report of many different tests, the following bituminous-coal gas analysis may be quoted:

	Combustible				Non-combustible			
	CO	CH_4	C_2H_4	H_2	CO_2	O_2	N_2	B.t.u. per cu. ft.
Max.	30.00	4.20	1.00	14.70	6.20	0.40	58.00	177.30
Min.	23.80	1.90	0.60	11.30	3.50	0.00	48.10	157.30
Avg.	27.50	3.40	0.80	12.70	4.60	0.24	50.80	167.10

Another investigator[15] found that his particular plants secured the best results from gas having the following analysis:

	CO	CH_4	C_2H_4	H_2	CO_2	O_2	N_2	B.t.u. per cu. ft.
Per cent	24.30	3.50	0.70	12.90	4.90	0.00	53.70	168.81

The following results are being obtained with dry-coke fines having the following analysis:

	Volatile	Fixed carbon	Ash	Sulphur	B.t.u. per lb.
Per cent.....	2.80	83.40	13.80	0.73	12,200

(NOTE: Sulphur separately determined.)

The gas delivered from producers using this coke has the following analysis:

	Combustible				Non-combustible			
	CO	CH_4	C_2H_4	H_2	CO_2	O_2	N_2	B.t.u. per cu. ft.
Per cent	26.40	0.70	0.00	12.60	6.30	0.10	53.90	134

The gas delivered from each pound of coke has 10,334 B.t.u., thus indicating a cold-gas efficiency of 84 per cent.

From an examination of these two fuels, it will be found that the coke-fines gas is devoid of the illuminant C_2H_4 (ethylene) and gives the very mellow, soft flame desired in heating processes.

The production of a gas devoid of C_2H_4 permits the use of less excess air, thus giving a better and wider range in flame control.

For the open-hearth and melting processes requiring an illuminating flame this gas should be ideal as it permits the addition of illuminants, such as tar or oil, in the correct quantity under separate control by parallel firing directly at the furnace.

Bituminous-coal producer gas for most processes contains too high a percentage of illuminants, thereby necessitating the use of too much excess air.

The C_2H_4 in such gas usually contains 5 to 6 per cent of the total heat units, while there is hardly any process requiring more than 2.8 to 3 per cent.

This excess C_2H_4 also has a destructive effect on brickwork, causing high refractory maintenance, which can be greatly reduced by substituting the dry coke fines for bituminous coal.

ADVANTAGES OF DRY QUENCHING

The advantages of dry quenching of coke are notable, and experience has shown the following very definitely:

1 Using of the sensible heat in the coke to produce approximately 1000 lb. of steam (from and at 212 deg. fahr.) at the gage pressure required, including superheat where necessary. This means the production of this steam without any additional fuel.

2 Improvement in the quality of the coke. Since dry-quenched coke was first introduced, many efforts have been made to compare the characteristics of dry-quenched with water-quenched coke. These have all shown that the dry quenching produces coke of a more uniform size with greater strength, and when the water-quenched coke has been handled to the same extent as the dry-quenched, the difference in size of the larger pieces is negligible. This is shown clearly in tests that have been run.

It is evident that when the glowing coke at a temperature of about 1800 deg. fahr. is suddenly chilled with water, the structure must be affected. Even though the expansion of coke by heat be small, certain stresses must follow this sudden change of temperature. These stresses, due to the brittleness of the material, form cracks and splits, which, with the sudden developing of steam, cause the breaking apart of the coke. Dry-quenched coke is not subject to these stresses, and is only reduced in size by handling.

The mechanical strength and friability of coke are factors of definite interest, as the formation of coke fines or breeze represents a loss regardless of whether the crumbling occurs in the loading or unloading, by shaking during transit, or in the furnace proper under the weight of the charge. Many investigations of the percentage of dust and coke breeze have determined the superiority of dry-quenched coke.

3 Dry quenching produces a coke devoid of moisture, which insures cleaner coke—that is, less fines and dust adhering to the large pieces. Better screening and preparation of coke is insured by dry quenching, and this is a very important difference in its favor. When coke is to be transported, a reduction in the moisture content of only 5 per cent means frequently a saving in freight of 50 cents per ton of coke delivered. Dry-quenched coke permits the accurate weighing of coke burden in metallurgical-furnace operations, as well as insuring cleaner and more uniform operation.

4 Dry quenching insures a higher sale value for the coke fines, as they form a very satisfactory substitute for bituminous coal in producer-gas plants. This is especially true at coke plants operating in connection with open-hearth and steel plants, as well as for gas producers in connection with the by-product coke ovens, as the dry-coke-fines producer gas takes the place of by-product coke-oven gas in firing the ovens.

5 Dry quenching eliminates the pumping of water necessary for all wet-quenching stations, as well as the dust-laden vapor clouds from wet-quencher stacks which are so destructive to surroundings and unpleasant for the plant personnel. Reduction in maintenance as well as increased life of steel structures are other valuable items to be given consideration.

6 Reduction in labor cost, as frequently one car can, depending on plant layout, do the work with a dry-quencher installation that required two cars for wet quenching.

7 Reduction in cost of conveyor operation, as the elimination of water due to dry quenching insures more continuous operation and lower maintenance cost.

ECONOMIES AND SAVINGS TO BE EXPECTED FROM THE INSTALLATION OF A DRY-QUENCHING PLANT

		Per net ton of coke cooled
1	Production of approximately 1000 lb. of steam (from and at 212 deg. fahr.) per net ton of coke cooled at a saving of 20 cents per 1000 lb. of steam.....	$0.20
2	Reduction in labor operating car and wharf...	0.01
3	Reduction in coke-car and plant maintenance..	0.02
4	Elimination of water pumped to wet-quencher installation.....	0.03
	Total direct savings per net ton.....	$0.26
On basis of a 1500-ton-per-day plant, or, say, 525,000 net tons of coke cooled per year the savings are.....		$136,500
Assuming that this coke plant is operating in connection with a steel plant where there are blast furnaces as well as producer-gas-fired heating furnaces. Then		

[15] F. E. Leahy, Assoc. I.&S.E.E. Convention, June, 1929.

5	Blast furnace assumed as now using 486,720 tons of wet-quenched coke will require with dry quenched coke, say, 467,250 tons, or a saving of 19,470 tons. The value of the coke saved at $4 per ton is	$77,880
6	As the quantity of dry-quenched coke fines available will only be approximately 11 per cent, or 57,750 net tons per year, this can be successfully used in the producer-gas plant in place of high-priced bituminous coal, with the following saving:	
	Coal per net ton delivered	$4.00
	Coke fines per net ton delivered	$2.50
	Gasification charge in each case	$0.50
	57,750 net tons of coke fines per year, cost	$144,375
	As the heat content in gas delivered from each net ton of coke fines is	20,668,000 B.t.u.
	and from each net ton of 14,000 B.t.u. bituminous coal is	19,600,000 B.t.u.
	the quantity of bituminous coal required to deliver the same heat units in the gas as produced from 57,750 net tons of coke fines will be approximately	60,890 net tons
	Value of this gasified	$274,005
	Value of coke fines gasified	$144,375
	Savings per year	$129,630
	At the value given, a dry-quencher installation cooling 1500 net tons of coke per day should conservatively show the following yearly saving:	
1	Direct savings in operation of plant	$136,500
2	Blast-furnace fuel	77,880
3	Dry coke fines in gas producer	129,630
	Total	$344,010

In the foregoing savings report such matters as interest on investment, profit, and amortization of plant have not been

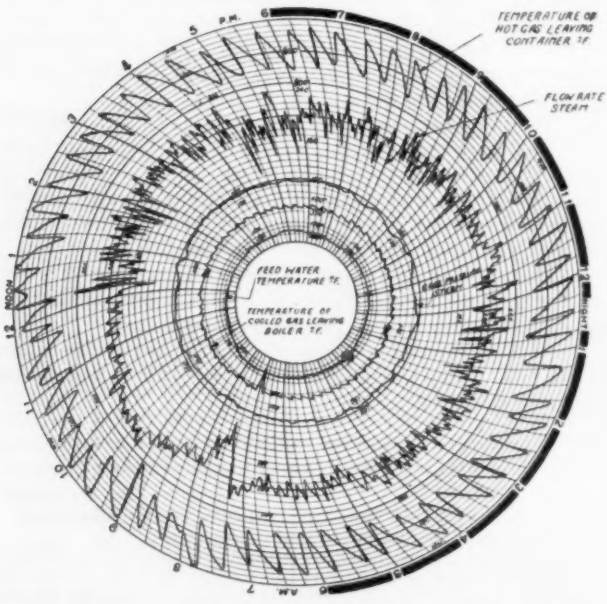


FIG. 4 COMPOSITE CHART GIVING STEAM AND GAS DATA FOR PLANT OPERATING 24 HOURS A DAY

worked out, as the installation assumed shows a complete return of the investment in less than one year.

However, there are many plants not so favorably situated as the one in question and where only the direct steam saving can be counted on plus the increased sales value of coke fines which amounts to from $1.50 to $2.00 per net ton, depending on plant location.

On the basis of figures given by A. M. Beebee in his paper read before the American Blast Furnace & Coke Association, November 26, 1929, the entire installation cost of a 1500-ton-per-day plant should not exceed $300,000.

In this same paper the following statement is made referring to the wet-quenching installation; "We have always had it and have grown to accept it as a necessary evil," and referring to dry-quenching installations such as theirs, where they cannot count on other savings: "It is easy to see that such a plant can earn yearly from 30 to 70 per cent on the investment, depending on the sales price of the steam."

DESCRIPTION OF DRY-QUENCHER INSTALLATION AT THE COURT STREET STATION OF CONSUMERS POWER CO., FLINT, MICH.

This plant has recently been put in operation, and as it differs considerably from installations at other plants, a description following the normal course of operation, with technical data, should prove of interest.

The hot coke is pushed from the coke ovens into a hot car of the usual type (except that it is much shorter, to suit the elevator cage), which is moved by the locomotive car on to a cage hoist where the locomotive is uncoupled and the hot car locked to the cage rails and hoisted up by means of an electrically driven entirely automatically controlled hoisting engine. As the load is hoisted the container charging door automatically opens, and when the cage reaches the top position the car discharge doors, electrically operated, automatically open and permit the coke to flow through the charging chute and door into the container. A time-limit switch then starts the empty car down, and in due time closes the charging door.

When the car again reaches the lower or ground position, the electrical equipment cuts out and the operator couples on the locomotive, releases the car locks, and pulls the car back to the battery in position to receive another charge when the oven is ready.

However, before another load of hot coke is put into the container, the operator withdraws an equivalent amount of cooled coke from the bottom of the container, loading it on to the coke wharf, whence it is taken by a wharf conveyor and transported to the screening station.

The dry quenching or cooling of the coke takes place in the firebrick-lined container which holds about seven loads of hot coke.

A circulating fan drives an automatically formed inert gas into the bottom of the container through a gas distributor and up through the bed of coke, which gas absorbs heat in cooling the coke and passes out near the top of the container through a superheater chamber and into a fire-tube boiler of the vertically inclined type. The gas passing through the boiler gives up heat and generates steam.

Dust carried over with the circulating gas is either precipitated in the superheater chamber or caught by the dust collector at the bottom of the boiler.

The cooled gas is drawn from the boiler into the suction side of the circulating fan and is discharged and driven into the bottom part of the container.

This process is continuous, and the inert gas is used over continuously as the entire system operates on a closed cycle.

To maintain this closed cycle, not more than one point in the gas-circulation system should be open at a time.

The charging door and discharging doors are only opened alternately so as to maintain the closed cycle.

The coke container is provided with a manually operated coke-level indicator so that the operator can readily determine the quantity of coke in the container and also gage the quantity discharged each time.

The coke-discharging device consists of a coke rake, operated

from the outside of chute, and gastight shut-off doors with latching devices to prevent infiltration of air.

The container, superheater chamber, and boiler are lagged and insulated to reduce radiation losses.

The boiler is provided with a Copes feedwater regulator and Reliance water column.

The hoisting machine and motor are placed in a separate brick

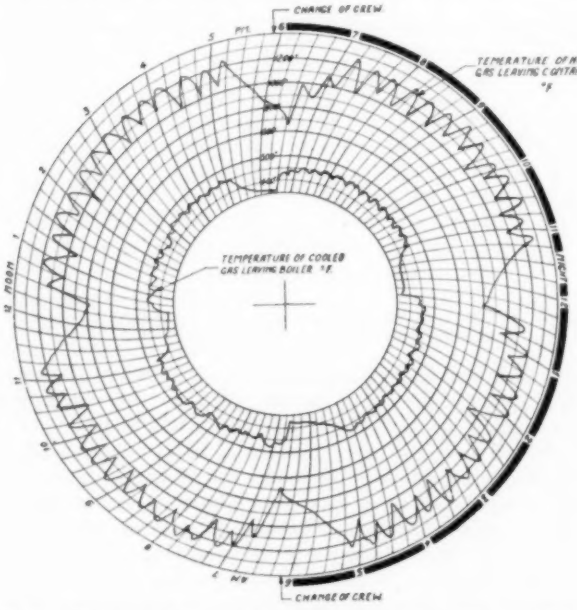


FIG. 5 COMPOSITE CHART GIVING GAS TEMPERATURES FOR PLANT OPERATING TWO TEN-HOUR TURNS IN EACH 24-HR. DAY

building with the operating controls and switch panels placed in a separate room.

MAIN TECHNICAL DATA OF FLINT, MICH., DRY-QUENCHER INSTALLATION

Number of coke ovens	29
Type of ovens	Horizontal by-product
Coke produced per charge	9000 lb.
Plant operates	Two (2) 10-hour turns per day
Capacity of dry quencher per 20-hour day	280 net tons
Capacity of dry quencher per hour	14 net tons
Temperature of glowing coke, approx.	1800 deg. fahr.
Temperature of cooled coke, approx.	700 deg. fahr.
Steam pressure	250 lb. gage
Steam superheat	100 deg. fahr.
Total temperature of steam	506 deg. fahr.
Feedwater temperature (about)	200 deg. fahr.
Time interval between loads (approximately)	19 min.
Mean gas temperature leaving container	1150 deg. fahr.
Outlet gas temperature (assumed)	450 deg. fahr.
Expected steam production per net ton of coke cooled from and at 212 deg. fahr.	1000 lb.
Expected steam production per hour from and at 212 deg. fahr. (approx.)	14,000 lb.

The steam produced is used in steam turbines and as process steam in the by-product plant.

COMPOSITE CHARTS OF PLANT OPERATION

Fig. 4 is the composite chart of a plant operating 24 hours per day. The diagram between the 800 and 1400-deg. lines shows the temperature of the gases leaving the coke container, from which the effect on temperature of each hot charge can be readily seen.

When a load of hot coke is put in the container the temperature rise is almost instantaneous, and then as the coke cools this temperature slowly drops until the next load is put in.

The diagram between the 160 and 240 lines shows the rate of steam flow and is a composite of steam production and demand. In general the fluctuations follow the hot-gas temperature line.

The next diagram shows the steam gage pressure in pounds per square inch, and this is succeeded by one showing the temperature of the cooled gases after passing through the boiler, ducts, and fan; in other words, the temperature of the gas going to the container. This latter diagram shows plainly the effect of each hot load and indicates that the discharging of cooled coke has very little effect on the gas temperature in this installation.

The innermost diagram shows the temperature of the feedwater in degrees fahrenheit.

This composite chart is not a record of test conditions, but was made up from actual operation charts of a plant operating 24 hours per day.

Fig. 5 shows diagrams of the hot and cooled gas temperatures in a plant operating on two 10-hour turns in each 24-hour day. The four delay periods of approximately one hour each show clearly that there is sufficient reserve heat in the stored coke to keep producing steam and quickly pick up the operating load when the hot-coke supply restarts.

In this plant the steam is produced at 250 lb. per sq. in. gage

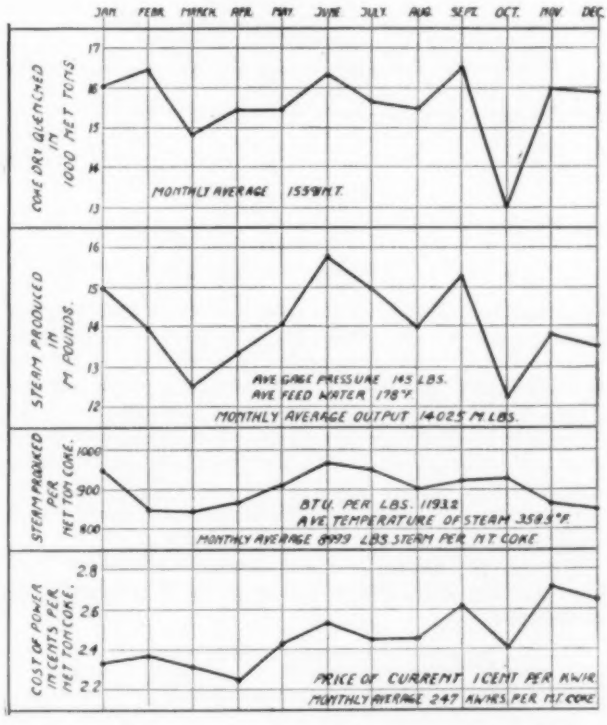


FIG. 6 OPERATING RECORD, BY MONTHS, FOR YEAR 1929

pressure and delivered with 100 deg. fahr. superheat, making a total steam temperature of 506 deg. fahr.

Fig. 6 gives the operating record of one plant for the year 1929. It shows:

1 The quantity of coke dry quenched per month and the monthly average for the year
2 The quantity of steam produced per month and the monthly average for the year
3 The quantity of steam produced per net ton of coke quenched for each month and the average for the year. The steam quantities are given in actual output at the pressure produced

4 The cost of power for the entire dry-quencher installation, including circulating gas fans and hoist.

All of the data given in Fig. 6 are taken from actual operation and are not test figures.

Cooling Gas

In the operation of a dry-quencher plant it has been found that sometimes hydrogen is formed, and in the early plants it was burned off at the opening of the charging doors. These latter were small, and therefore created a restriction which required the use of a small pilot flame.

However, improvements in later installations seem to have

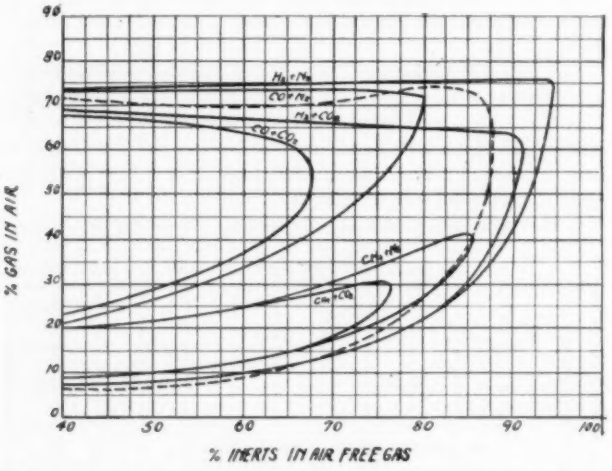


Fig. 7 Showing Composition of Various Mixed Gases in Relation to Dry-Quencher Cooling Gas
(From data by J. W. Jones in *Ind. & Eng. Chem.*, vol. 20, 1928.)

eliminated this difficulty, at the same time giving some additional heat recovery from the burning of the small amount of hydrogen.

As previously mentioned, a mixture of CO and CO_2 is formed in the apparatus at the very start of operation. To this mixture is sometimes added a certain amount of hydrogen, which may increase to about 20 per cent if the cooling chamber is charged with badly distilled coke. The suggestion has been made that such a mixture of gases might lead to explosions. An analysis of the conditions, together with the experience of actual installations, proves that these fears are unfounded.[16]

The following table shows percentage gas analyses from the dry-quenching installation at the gas works of Keilehaven, near Rotterdam:

	Time between observations							
	13 min.		20 min.		19 min.		16 min.	
	Sample numbers							
	I	II	III	IV	V	VI	VII	VIII
CO_2	6.5	7.0	8.6	8.8	13.3	12.4	12.5	11.7
O_2	0.2	0.2	0.3	1.0	0.3	0.7	0.3	1.2
CO	6.7	6.6	6.9	6.4	4.5	4.5	7.0	6.5
CH_4	2.6	2.2	2.0	1.1	0.2	0.2	0.2	0.1
H_2	18.4	11.9	11.4	6.7	0.4	1.3	3.8	1.7
N_2	65.6	72.1	70.8	76.0	81.3	80.9	76.3	78.8

The samples numbered I, III, V, and VII were taken shortly after refilling the container, while those numbered II, IV, VI, and VIII were taken shortly before refilling the container with glowing coke.

In investigating the possibility of explosion, 60 gas analyses taken at this coke-quenching plant were considered. It was found that the proportion of carbon monoxide, as well as that of hydrogen, reached at most a maximum of about 20 per cent.

[16] M. C. Sissingh, *Gas und Wasserfach*, no. 51, 1923.

TABLE 3 ANALYSES OF DRY-QUENCHER-GAS SAMPLES

	No. of samples
Less than 10 per cent H_2 and less than 4 per cent CO:	
Due to presence of air in quencher	10
Due to nearly complete combustion	58
Less than 10 per cent H_2 and 4 per cent or more of CO	218
Between 10 per cent and 20 per cent H_2	154
Between 20 per cent and 30 per cent H_2	69
30 per cent H_2 or more	25
Total number of samples	534

TABLE 4 CORRELATION OF GAS-ANALYSIS DATA

No. of samples	Lower limit	Upper limit	CO	H_2	CH_4	Inerts	B.t.u.
10	12	64	6	33	2.5	58	153
3	15	65	8	35	2.3	54	166
8	13	68	10	32	1.4	57	150
4	15	69	13	25	1.3	62	134
1	18	72	12	17	1.6	69	113
5	20	71	8	19	1.2	72	98
9	23	71	6	18	0.9	75	88
2	24	73	14	14	1.0	71	101
7	27	73	8	13	1.0	78	78
11	28	73	6	< 10	...	> 84	< 62
14	30	72	5	< 10	...	> 85	< 50
15	32	72	4	< 10	...	> 86	< 47
6	31	73	4	< 10	...	> 86	< 45
12	31	74	9	< 10	...	> 81	< 61
16	32	74	8	< 10	...	> 82	< 60
13	36	74	8	< 10	...	> 84	< 57
17	None	None	2	< 10	...	> 88	< 39

This would result in a gas consisting of a mixture of 20 per cent of carbon monoxide and carbon dioxide, and 20 per cent of hydrogen diluted with 60 per cent of carbon dioxide and nitrogen.

The explosion limit for a mixture of water gas and air at average temperatures is between 12.5 and 66.6 per cent of water gas. If air is allowed to enter the mixture of cooling gases it serves to dilute the mixture, and the concentration of combustible material is lowered. Laboratory tests and actual experience in plant operation have shown that no apprehension is to be felt for the safe performance of Sulzer System dry-quenching plants.

At one plant 534 samples of dry-quencher gas were analyzed during the year. These analyses were made primarily to indicate the presence of possible leaks in the system and not as another check on operating conditions, and the results are summarized in Table 3 in terms of the main inflammable constituents in the gases.

The analyses were made by an experienced operator and are considered accurate with 0.1 per cent for each constituent. The B.t.u. was calculated at 60 deg. fahr. and 30 in. pressure.

Consideration of the Data

In order to study the composition of dry-quencher gas, it is necessary to correlate the data and bring the essential figures into a more compact form, as in Table 4. The lower limit is taken as a basis for the following classification. Since oxygen and carbon dioxide seem to affect the limits very slightly, if at all, these have been added to the percentage of N_2 or inerts. Carbon dioxide, as is seen, varies, from about 2 per cent to 10 per cent, and generally the higher percentages are found among those analyses having less than 10 per cent H_2, as would be expected. The other figures are rounded off to the nearest unit except in the case of methane.

Fig. 7 shows curves plotted from the data given by J. W. Jones of the Bureau of Mines,[17] for $H_2 + CO_2$, $H_2 + N_2$, $CO + CO_2$, $CO + N_2$, $CH_4 + CO_2$, and for $CH_4 + N_2$.

These curves differ slightly from those given by other authorities as to curve for $H_2 + CO_2$, but the differences can be accounted for by the type of apparatus used. Jones used upward propagation, i.e., his flame started at the bottom and traveled upward, while most others ignited the mixtures in the center, or upper center, by means of a fuse wire.

Upward propagation was used in these tests, and the results obtained compare favorably with the work of Mr. Jones.

[17] *Industrial and Engineering Chemistry*, vol. 20, 1928.

Dry-quencher gas is continually changing, so that it is therefore hard to get an accurate average, and the curve given in Fig. 7 covers but one condition.

The following few analyses of the circulating gas, taken at the latest installation, will give an idea of the composition of this cooling medium, which is found immediately after starting up with air, as the oxygen is quickly consumed when the air circulates through the first load of coke.

CO_2	O_2	CO	
—Per cent—			
13.80	2.20	3.00	After charging
12.40	0.00	5.40	Cooling
9.40	2.20	4.00	After charging
12.50	0.00	4.90	Cooling

An Installation of the Latest Type

Fig. 8 shows a modern-type installation similar to the Flint plant, except that the glowing coke is transferred from the by-product oven by means of a standard hot-coke car and discharged at the dry quencher into a bucket which is hoisted and dumped into the top of the coke container automatically.

The circulating cooling gases are fed to the discharge chute and bottom portion of the container, as well as through the distributor, thus taking advantage of all the space containing coke for cooling purposes.

This installation includes a superheater and vertically inclined fire-tube boiler with simple type of dust collector at the bottom. The dust collected in the superheater chamber is intermittently discharged into the boiler dust collector, from which all dust is drained approximately once in each turn.

Shut-off dampers are provided to shut off boiler units when necessary. Regulating dampers are installed in the ducts leading to the distributor and container hopper so that the distribution of cooling gas can be adjusted as required.

This type of installation discharges the cooled coke back to the hot-coke car, which delivers it to the regular wharf for distribution by belt conveyor to point of storage or use.

It will be seen from the figure that the container, superheater chamber, and boiler head are well insulated, while the boiler is lagged on the outside to reduce loss by radiation to the minimum.

The charging door, discharging coke rakes, and discharge doors are all mechanically operated and provided with push-button controls.

Sizes of Installations

Local conditions, capacity in net tons of coke per day, and oven output affect the dry-quencher-plant designs, and the illustration referred to below should give a fairly good idea of the equipment required.

Fig. 9 shows a plant of small capacity where the time interval between charges is relatively great; it consists primarily of one or more cooling chambers in which the special coke container is stored for the predetermined period needed to bring the coke down to handling temperature.

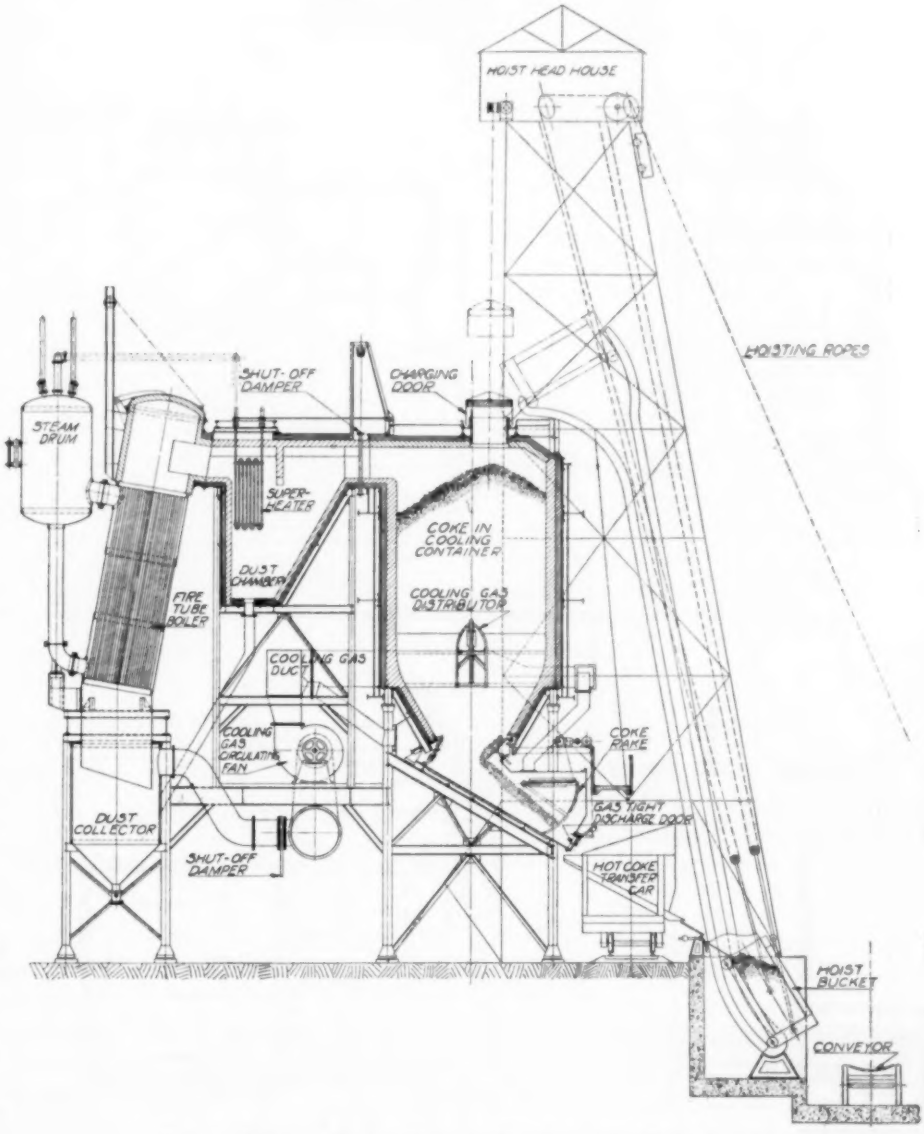


Fig. 8 Showing Latest Type of Dry-Quencher Installation

Fire-tube waste-heat boilers of suitable size are installed in connection with the cooling chambers so that the circulating fan will cause a current of automatically formed inert gas to pass through the coke, into and through the boiler, thence back to fan, ad lib., in a similar manner to that previously described for the process.

The quantity of steam produced by such a plant is usually slightly less than with a larger installation.

Fig. 10 shows a slightly different type of small-capacity installation where the boiler, of the water-tube type, forms the cooling chamber and the coke charge in a special container is stored inside what is practically a water-wall-cooled furnace. The cooling

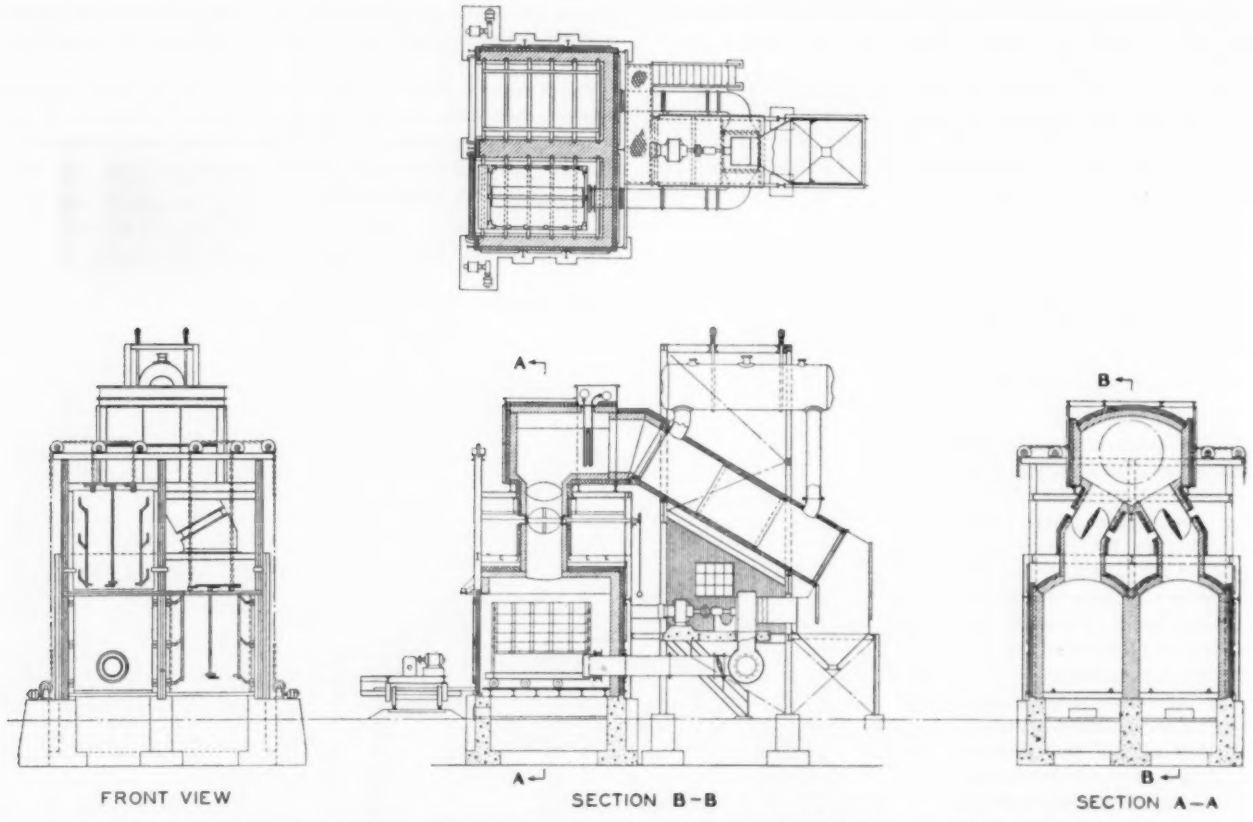


FIG. 9 SHOWING SINGLE-CHARGE COOLING-CHAMBER-TYPE INSTALLATION WITH INCLINED FIRE-TUBE BOILER

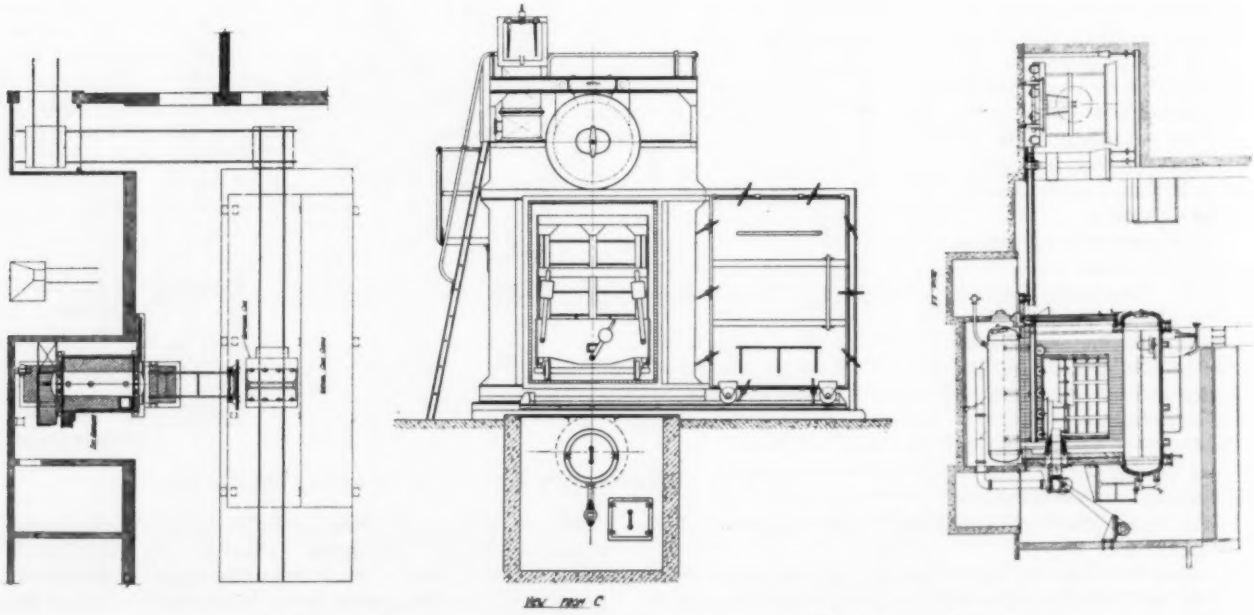


FIG. 10 SHOWING ARRANGEMENT OF SMALL-CAPACITY INSTALLATION WITH WATER-TUBE BOILER

gases pass up through the coke, then down through the boiler-tube interstices, and out at the bottom to the circulating fan. Reproduction of photographs of a plant of this type are given in Figs. 16 to 23.

Fig. 11 shows a plant similar to the installation shown in Fig. 8, but having a hoist tower and bucket to receive the coke from the hot-coke car, and then elevate and discharge it into the top of the coke container. The capacity of the plant is 530 net tons of coke per day, a charge of hot coke being received approximately

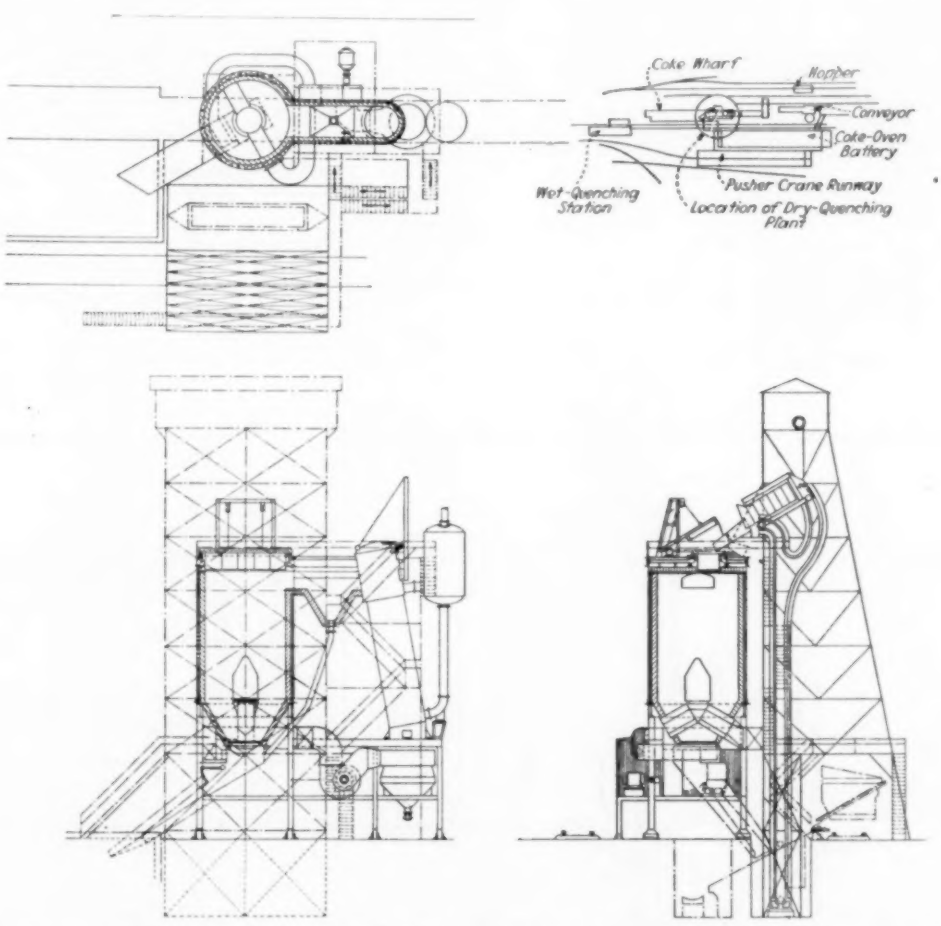


FIG. 11 SHOWING INSTALLATION OF 530 NET TONS PER DAY CAPACITY WITH SINGLE CONTAINER UNIT AND SINGLE VERTICALLY INCLINED FIRE-TUBE BOILER

every 20 min. Fig. 12 shows a similar plant having a capacity of 1000 net tons of coke per day, receiving a charge of hot coke approximately every 15 min. These two installations are arranged to discharge directly on to existing coke wharfs.

Fig. 13 shows a plant of 2200 net tons daily capacity which has a rectangular coke container and three boilers so arranged that any one may be taken out of service when required. This installation discharges the cooled coke into the hot-coke transfer car, which in turn delivers it on the existing wharf, the machinery being designed to handle a charge of coke approximately every 7 min.

Fig. 14 shows a plant designed to handle 3800 tons of coke per day, located over a screening station and built in two units, each equipped with a hoist tower, bucket, and three boilers. It is arranged to discharge the cooled coke on to an intermediate dust-settling wharf, from which it is fed through chain feeders uniformly to belt conveyors.

This system releases the hot-coke car immediately after it has discharged a load of glowing coke into the hoist bucket, which permits one car to do the work where ordinary wet quenching would require two hot-coke cars. It also provides gas burners for use in auxiliary firing of boilers in the event that the coke output runs below that required to produce all the steam required to operate turbines, etc. The time interval between charges for this plant is approximately 5 min.

Fig. 15 shows the 1000-ton-per-day installation at Homécourt, France, which has been in successful operation for some years. This, being one of the early installations, is equipped with water-tube boilers and was designed on the basis of a four-hour cooling time; later experience in this country, however, has demonstrated that only two hours' cooling time is all that is economically needed.

Figs. 16–23 are reproduced from photographs of a 15-ton-per-day plant that has been in successful operation for the past two years. This installation showing sectional elevation in Fig. 10 is at a plant fitted with vertical retort ovens, each oven being of 1.5 net tons coke capacity.

The cooling period in this installation is approximately $1^1/_2$ hours and the steam production is about 800 lb. at 200 lb. gage pressure, the power consumption being approximately 2.5 kw-hr. per net ton of coke cooled.

From an inspection of these views it will be seen that the hot-coke container is moved into and out of the boiler chamber

mainly by means of a hand winch. This of course can be done by power where necessary.

The glowing coke comes to the boiler chamber at an average temperature of 1800 deg. fahr., and the cooled coke is discharged at approximately 450 deg. fahr.

Fig. 24 is a photographic view of the dry-quenching plant at the gas works of the Rochester Gas and Electric Corporation, Rochester, N. Y. This plant was built on the basis of four hours

Coal and Coke Analyses

At both the Rochester and Flint plants high-volatile coals are used, the relative coke and coal analyses being as follows:

Coal Charged to Ovens:

B.t.u. per lb	14,300
Volatile, per cent	36.47
Fixed carbon, per cent	58.97
Ash, per cent (calculated on moisture-free basis)	4.56
Sulphur separately determined, per cent	0.75

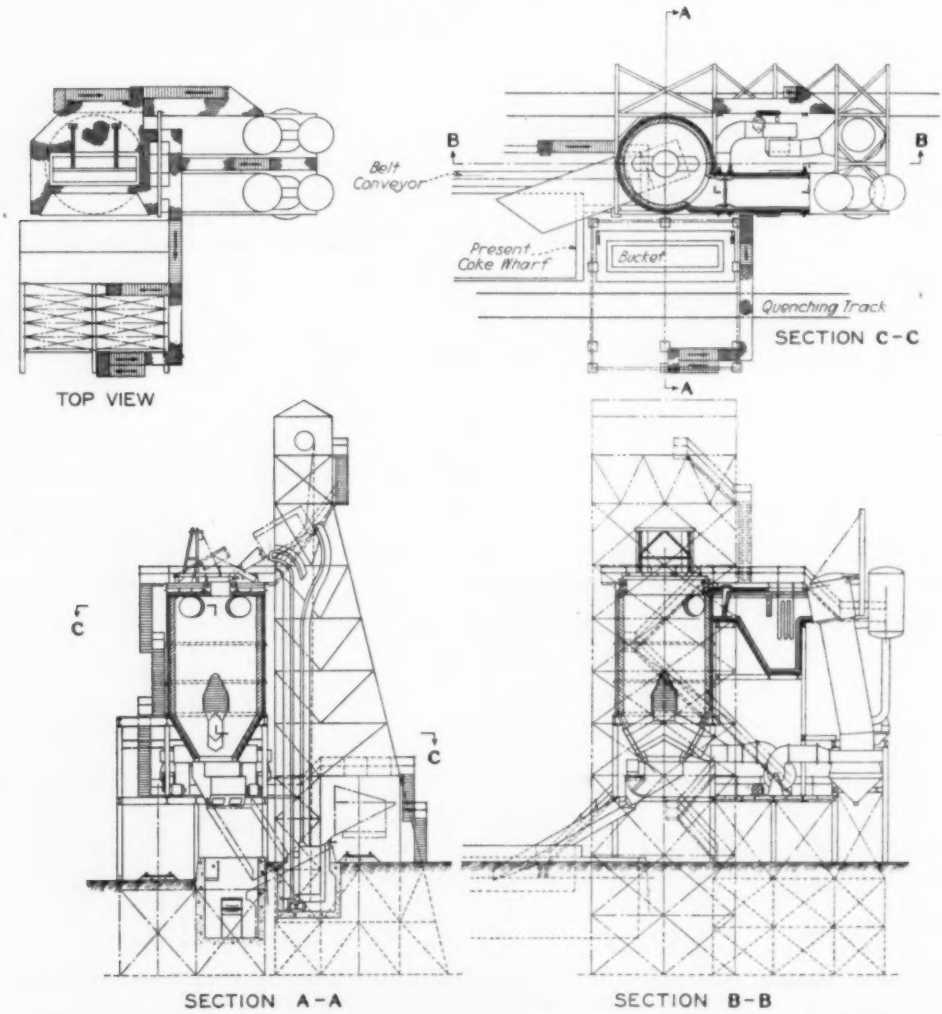


Fig. 12 Showing Installation of 1000 Net Tons per Day Capacity With Single Container Unit and Two Boilers

cooling time for a capacity of 425 net tons of coke cooled in 24 hours. Tests and experiments have demonstrated that two hours cooling time is sufficient, so that new plants designed on this basis will naturally represent a considerably lower investment.

The plant consists of two complete container and boiler units, and over 500 tons per 24 hours have been successfully cooled in one unit on occasions when required. Steam is produced at approximately 140 lb. gage pressure, without superheat.

Fig. 25 shows the dry-coke quenching installation at the Court Street Station of the Consumers Power Company, Flint, Michigan. This installation has a capacity of cooling 280 net tons of coke in 20 hours as the plant operates on two 10-hour shifts in each 24-hour day. Steam is produced at 250 lb. gage pressure with 100 deg. superheat and a total steam temperature of approximately 506 deg. fahr.

Coke Produced, Wet-Quenched:

	Moisture free	As received
B.t.u. per lb	13,039	11,463
Volatile, per cent	1.71	1.49
Fixed carbon, per cent	91.29	79.51
Ash, per cent	7.00	6.10
Moisture in coke as received, per cent		12.90
Sulphur separately determined, per cent		0.75

Coke Produced, Dry-Quenched:

	Moisture free	As received
B.t.u. per lb	13,088	13,023
Volatile, per cent	1.16	1.15
Fixed carbon, per cent	91.48	91.03
Ash, per cent	7.36	7.32
Moisture in coke as received, per cent		0.50
Sulphur separately determined, per cent		0.68

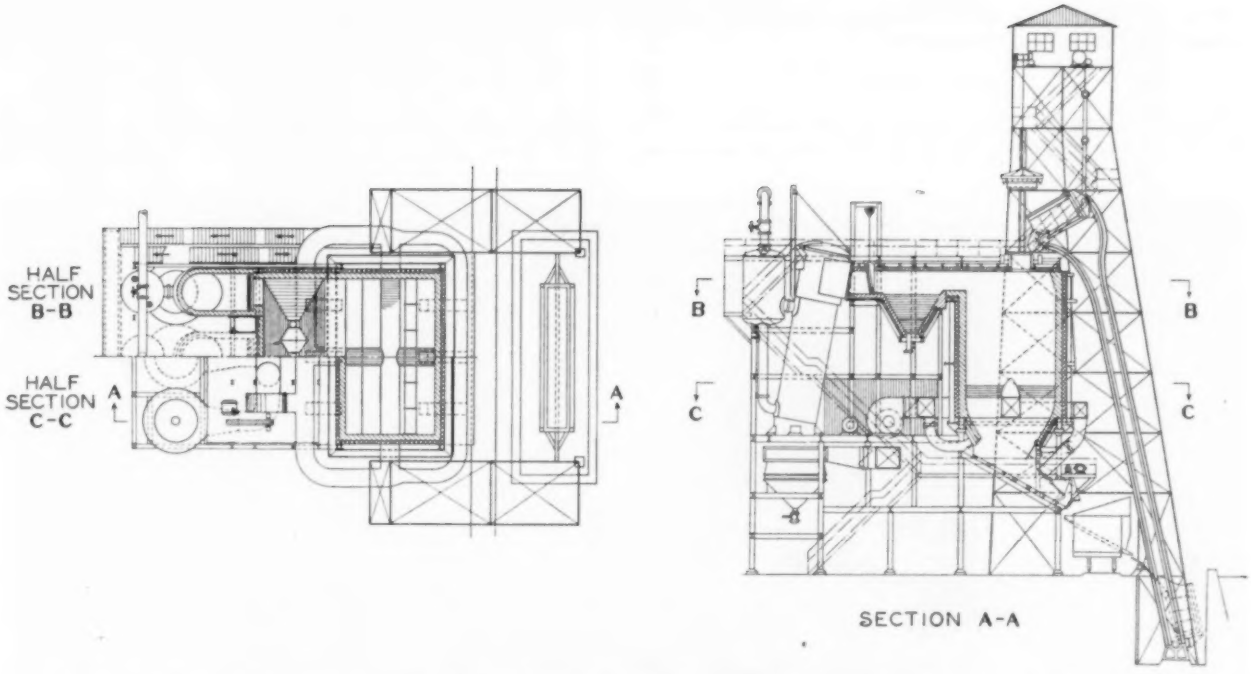


FIG. 13 SHOWING INSTALLATION OF 2200 NET TONS PER DAY CAPACITY WITH SINGLE CONTAINER UNIT AND MULTIPLE BOILERS

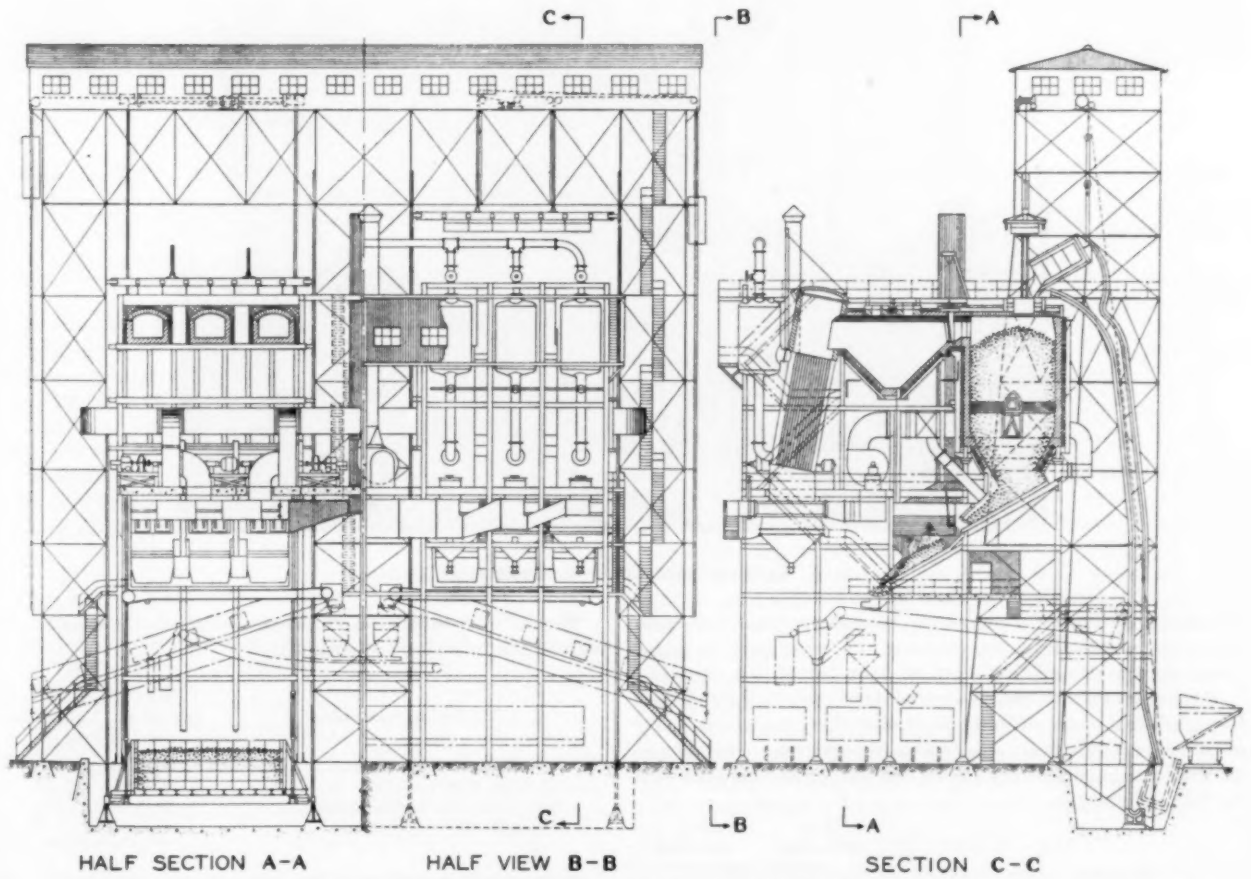


FIG. 14 SHOWING INSTALLATION OF 3800 NET TONS PER DAY CAPACITY WITH TWO CONTAINERS AND MULTIPLE BOILERS

From the foregoing it will be seen that dry-quenched coke has a higher heat value than wet-quenched.

Screen Tests

One of the items of special interest to coke manufacturers is the size of coke produced, and as the Rochester dry-quenching installation apparently handled the coke more roughly than do the usual water-quenching methods, improvements in plant design were developed that seem to have helped the situation.

While plants of new design have not been operating a sufficient length of time to get definite, conclusive data, the results so far obtained seem to indicate a noticeable improvement. Some comparable tests have been run on wet- and dry-quenched coke with the following results:

	Wet-quenched	Dry-quenched
	—Per cent—	
Over 2 3/4-in. × 2-in. screen	63.6	55.3
Through above screen	11.5	15.5
Over 2-in. × 2-in. screen	11.5	11.5
Through above screen	19.4	25.6
Over 3/4-in. × 3/4-in. screen	19.4	25.6
Through above screen	5.5	3.6

From this it will be seen that the quantity of fines through 3/4-in. screen is less in dry-quenched than in water-quenched coke.

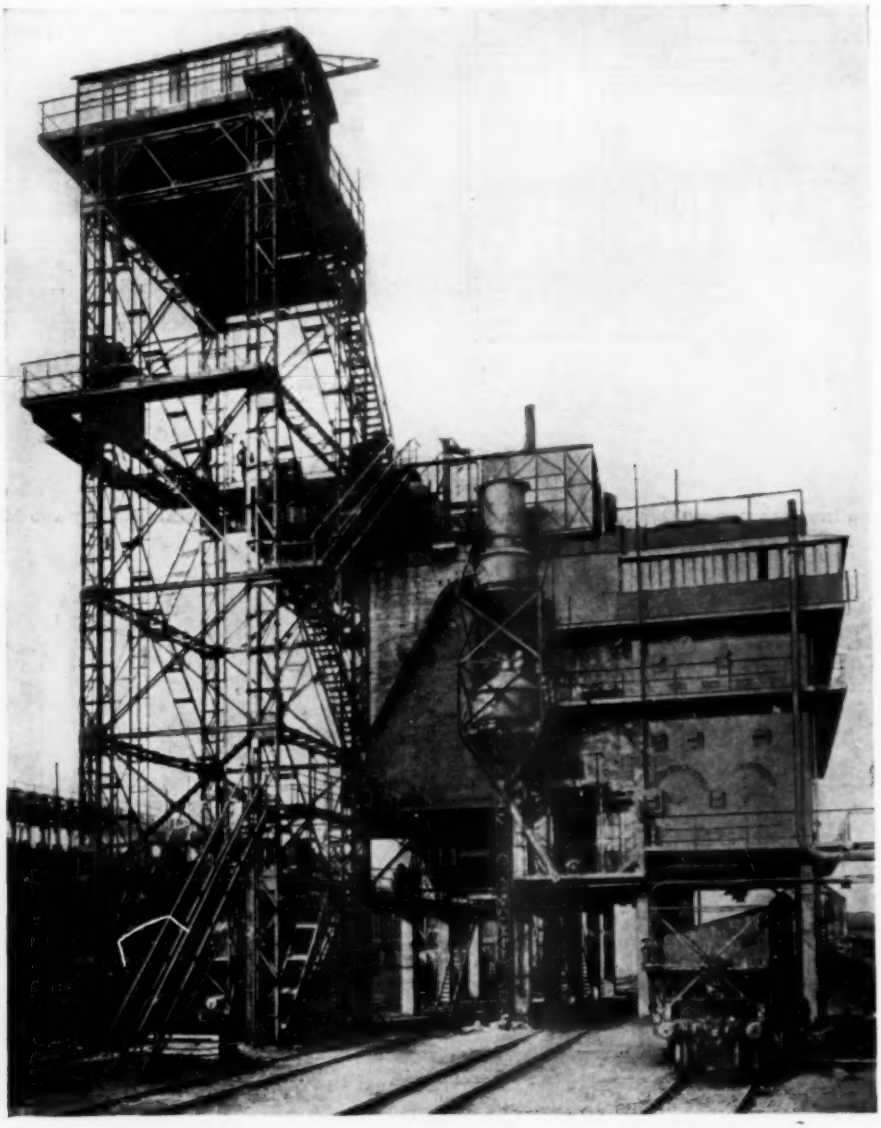


Fig. 15 View of Installation at Homécourt, France

Shatter tests carried on at the same time as the screen tests showed the following comparison:

Method of quenching	Wet-quenched	Dry-quenched
	—Per cent—	
Over 2-in. × 2-in. screen	50.0	66
Through above screen	39.5	26
Over 3/4-in. × 3/4-in. screen	39.5	26
Through above screen	10.5	8

From this it would seem that dry-quenched coke is stronger on 2-inch screen than water-quenched.

This bears out the results of tests run by C. C. Russell[18] in 1927 at Rochester, as we find it stated in his report that—

Dry-quenched coke (through 3-in. on 2-in. screen) when subjected to the standard shatter test (four 6-ft. drops) still contains 63 per cent of coke on a 2-in. screen, whereas wet-quenched coke of the same original size contains 50.1 per cent on a 2-in. screen after the same treatment.

	Per cent		
Over 2¾-in. × 2-in. screen	62.04	60.71	38.79
On 1½-in. × 1-in. screen	25.05	21.17	43.77
On 1-in. × ½-in. screen	2.35	2.50	4.98
On ½-in. × ½-in. screen	3.47	2.86	6.86
Through above screen	7.09	12.76	5.60
Total	100.00	100.00	100.00

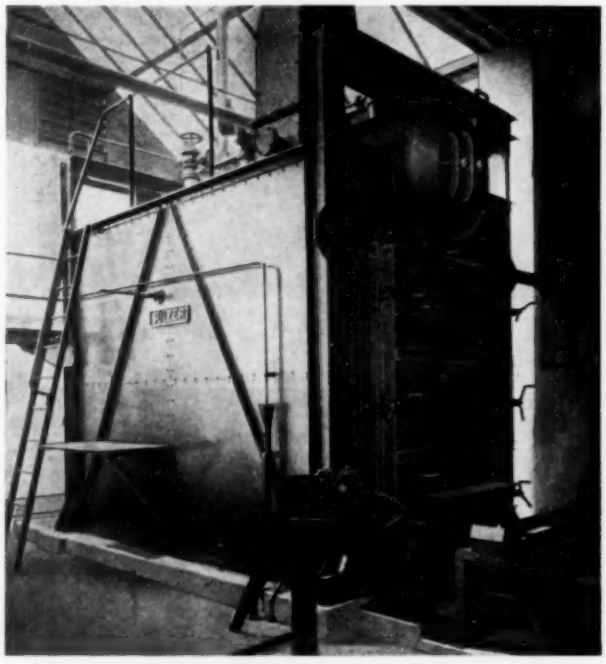


Fig. 16

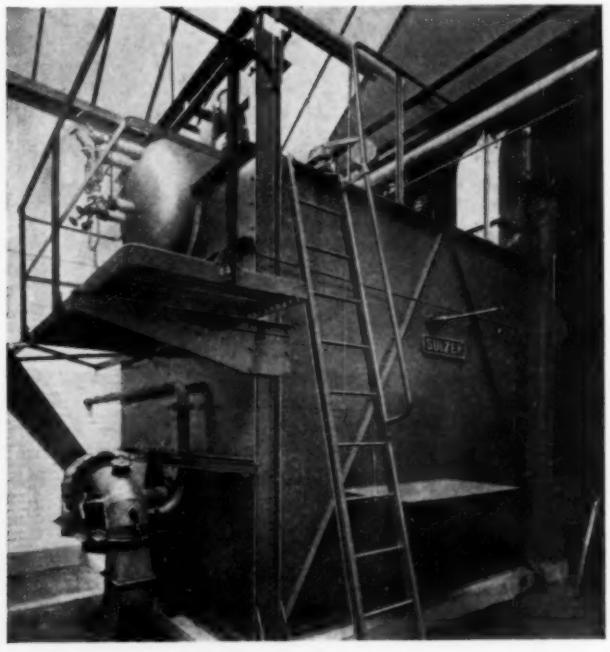


Fig. 17

FIGS. 16 AND 17 VIEWS OF SELF-CONTAINED UNIT INSTALLATION AT SCHAFFHAUSEN, SWITZERLAND

Repeated shatter tests on the same samples of dry- and wet-quenched coke show that any given size of dry-quenched coke resists breakage better than the same size of wet-quenched coke, no matter how much rough handling they are given.

In every case, then, the dry-quenched coke is stronger than the wet-quenched coke of the same size.

At the Flint plant it was observed that the water-quenched and dry-quenched coke as delivered to the wharf differed somewhat in appearance, although some lumps as large as 8 and 10 in. were discerned coming from dry quencher.

Additional samples were then taken and comparable screen tests run as follows. The first column gives the results from a wet-quenched sample taken from wharf; the second gives results from the same sample dried to a moisture-free condition; and the third, similar data from a dry-quenched sample.

	Weight in pounds		
Weight of sample	310	280	281
Weight over 2¾-in. × 2-in. screen	192½	170	109
Weight through above screen	117½	110	171½
Weight over 1½-in. × 1-in. screen	77½	59¼	123
Weight through above screen	40	50¾	49½
Weight over 1-in. × ½-in. screen	7¼	7	14
Weight through above screen	32¾	43¾	35
Weight over ½-in. × ½-in. screen	10¾	8	19¼
Weight through above screen	22	35¾	15¾

Reducing these data to a percentage basis, the following comparison is secured:

[18] Koppers Co., "A Comparative Study of Coke at the Plant of Rochester Gas and Electric Corporation Made December, 1926."

A further test was run to determine, if possible, the amount of fines adhering to the coarse in wet-quenched coke, as no doubt the fines so adhering would be released in furnace, or stock bins, as the moisture evaporated.

A sample of 730 lb. of wet-quenched coke was therefore screened, the moisture partially eliminated through air drying, and again screened, with the following results:

		After drying
Coarse, over ½-in. screen, lb	730	589
Fines through ½-in. screen, lb	73	88
Fines through ½-in. screen, per cent	10	15
Moisture in fines, per cent	23	5

This showed clearly that dry coke is much cleaner than the usual run of moisture-laden coke, and that there is actually more fines with the wet-quenched than the dry-quenched, although the usual screening method does not show it.

It should be noted that the difference in weight of sample cannot be construed as all moisture, because a portion of the fine coke was lost in handling so that the weights given represent the actual quantity put on the screen, and the figures giving the amount of fines after air drying are therefore conservative.

From the foregoing it will be noted that the fines are very much greater in the wet-quenched coke after drying than in the dry-quenched coke. This simply indicates the condition that arises when the water-quenched coke is put in a furnace, and proves that the dry-quenched coke is cleaner and more desirable.

HEATING VALUE OF COKE

Most wet-quenched coke contains a large percentage of moisture, while dry-quenched coke is moisture-free when it leaves



Fig. 18

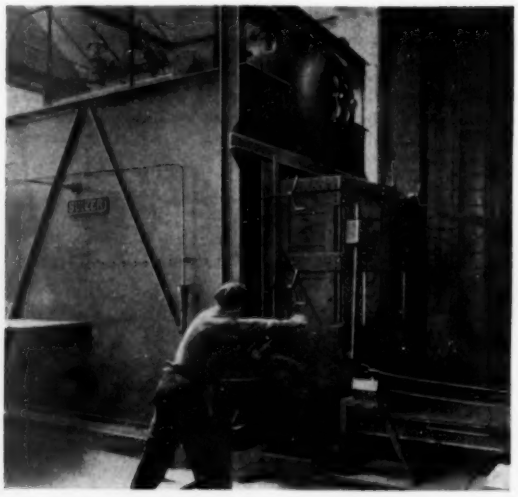


Fig. 21

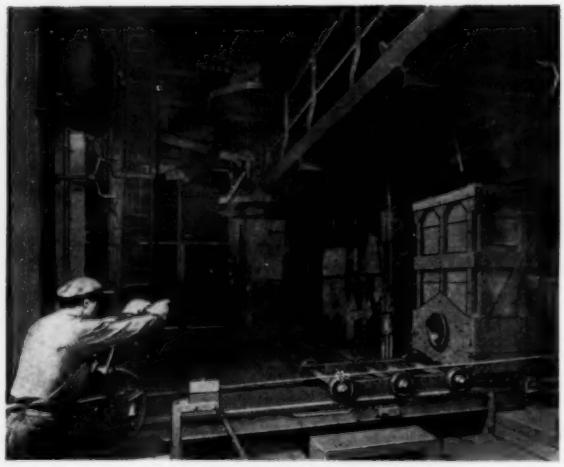


Fig. 19

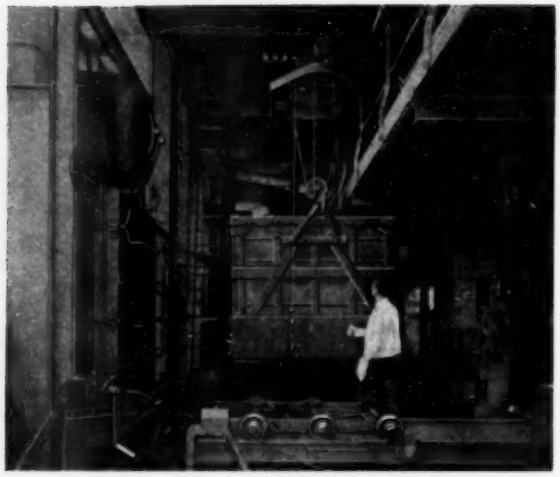


Fig. 22

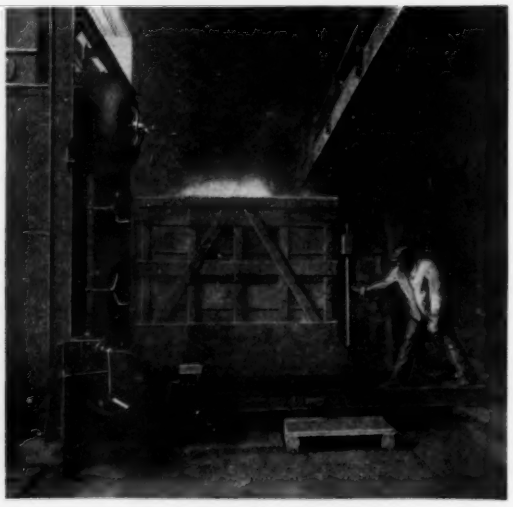


Fig. 20



Fig. 23

FIGS. 18-23 VIEWS OF SELF-CONTAINED UNIT INSTALLATION AT SCHAFFHAUSEN, SWITZERLAND

the quenching unit; sometimes, however, it is sprayed or absorbs moisture from the air, which, it has been found, amounts to from four-tenths to sixth-tenths of 1 per cent.

The actual heating value per pound of coke as received is naturally affected by this moisture content, so the following comparisons will no doubt be of interest.

Tests of wet-quenched coke as received by consumer show 8.10, 11.65, 14.20, and 19.79 per cent moisture. Dry-quenched-coke tests show 0.41, 0.50, and 0.60 per cent, so an average of 0.50 per cent is conservative.

The comparative heating value of water-quenched and dry-quenched coke, assuming an average of 8.10 per cent moisture for the former, is therefore as follows:

	B.t.u. per lb. as received
Water-quenched coke	11,983
Dry-quenched coke	13,023
Actual increase in heating value	1,040

or, on a percentage basis, the consumer receives 10.87 per cent more heat units per pound of dry-quenched coke than per pound of water-quenched coke.

In the case of coke fines it seems that the moisture is concentrated, and it has been found that wet-quenched coke fines through a $^3/_4$-in. screen contained from 20 to 30 per cent of moisture, while the dry-quenched maximum was but 0.8 per cent. Assuming, therefore, an average for wet-quenched of 23 per cent and for dry-quenched a maximum of 0.8 per cent, the following comparison is conservative:

	B.t.u. per lb. as received
Water-quenched coke	10,525
Dry-quenched coke	12,974
Actual increase in heating value	2,449

or on a percentage basis, the consumer receives $23^1/_4$ per cent

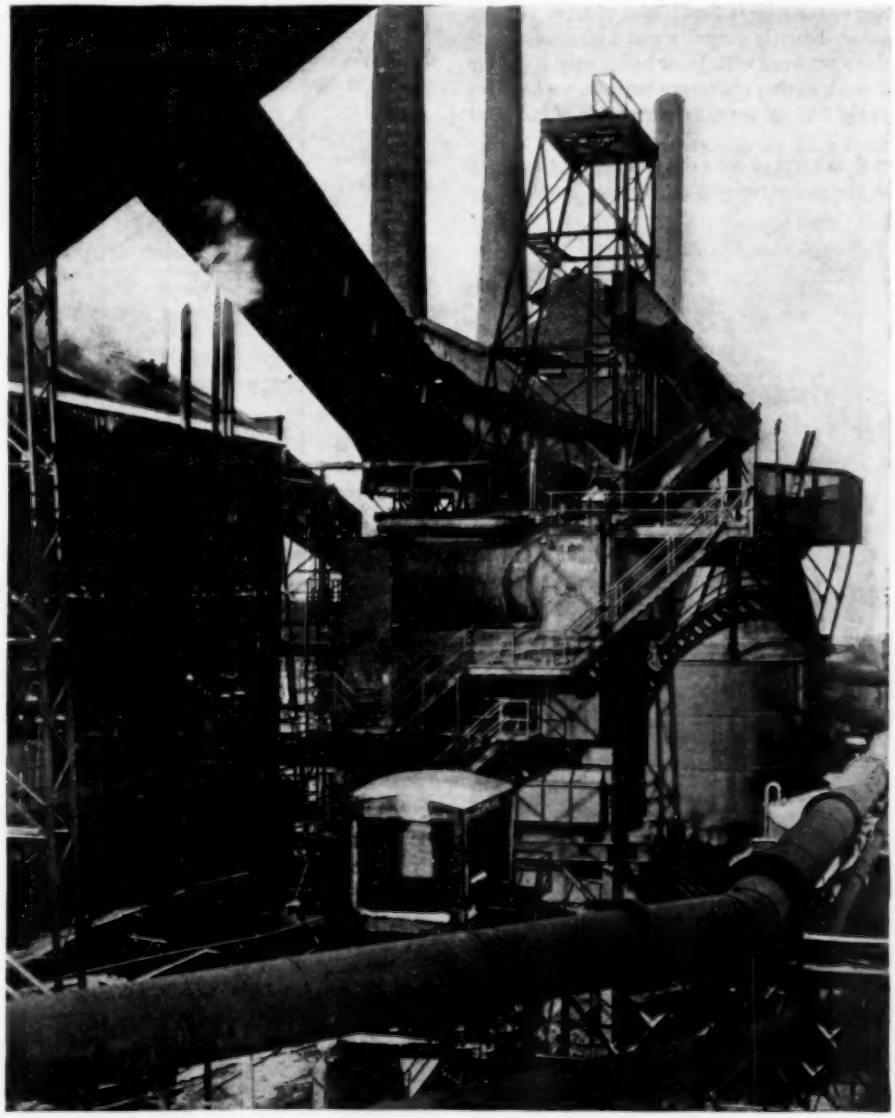


FIG. 24 VIEW OF INSTALLATION OF PLANT OF ROCHESTER GAS & ELECTRIC CORPORATION, ROCHESTER, NEW YORK

more heat units per pound of dry-quenched coke fines than per pound of wet-quenched coke fines.

These figures indicate that the consumer should insist on dry-quenched coke wherever it can be used.

Fig. 26 shows the relative heat units in dry-quenched and wet-quenched coke as received by consumer at various moisture contents.

Return on Investment

Assuming an average overall investment cost of $240 per net ton of installed dry-quencher daily capacity and a steam-producing operation of 350 days per year with labor being paid 365 days per year due to repair work during shutdown periods, we have an average output per year of 350 net tons of coke per ton of installed quencher capacity.

From the heat in each net ton of coke dry quenched, there will be produced steam at the pressures and superheats required by

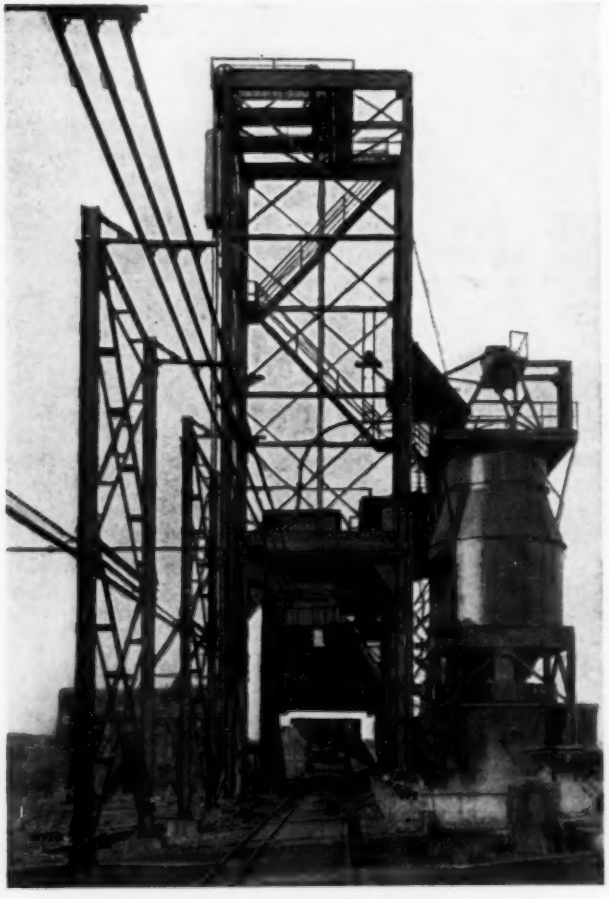


Fig. 25 View of Installation at Consumers Power Company, Court Street Station, Flint, Michigan

different plants in varying amounts, and the chart in Fig. 27 has been made up on the following basis:

1 Steam produced from and at 212 deg. fahr. at rates of 800, 900, and 1000 lb. per net ton of coke cooled
2 Steam values of 20 to 60 cents per 1000 lb. from and at 212 deg. fahr.
3 Return on the investment in dollars per year per net ton of coke installed daily quencher capacity, for each of the steam outputs at money values per 1000 lb.

This latter figure is made up as follows:

1 Value of steam produced
2 Less cost of steam, including fixed charges amounting to 16 per cent on the investment cost

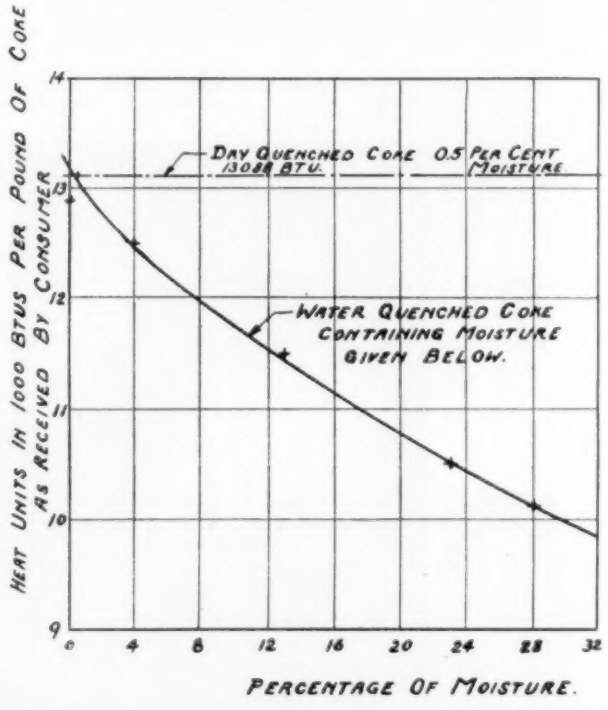


Fig. 26 Chart Showing Relative Heating Value of Dry-Quenched and Wet-Quenched Coke

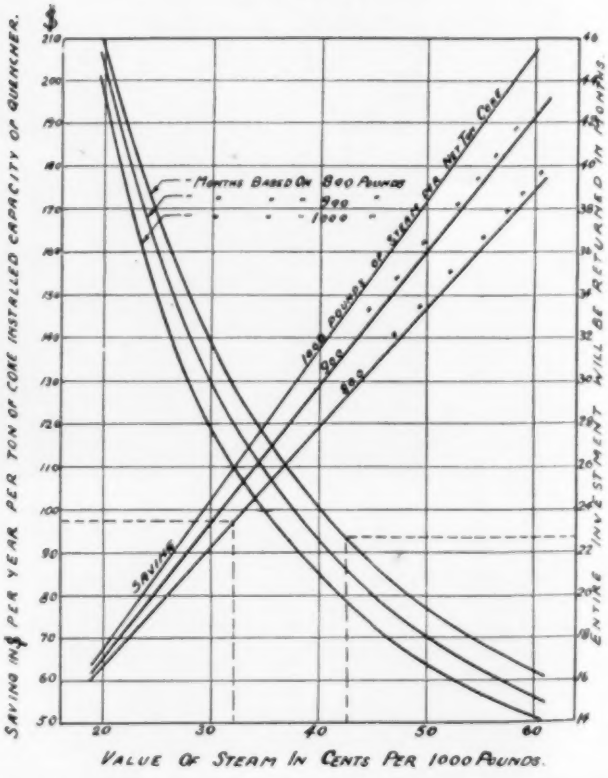


Fig. 27 Chart Showing Savings and Return on Investment Obtainable With a Dry-Quencher Installation

3 Less cost of pumping water in existing wet-quenching plant—assumed at 2 cents per net ton of coke
4 Less an allowance for reduced maintenance due to elimination of corrosive action of wet quenching—assumed at 1 cent per net ton of coke
5 Less increased sales value of dry coke fines, assumed at $1 per net ton, on basis of this being 11 per cent of the total coke cooled.

From the chart it will be noted that an installation producing 1000 lb. of steam from and at 212 deg. fahr. per net ton of coke cooled with a steam value of 60 cents per 1000 lb., shows a saving of $206.50, at which rate it will equal the investment cost in less than 15 months.

Correspondingly, a plant producing but 800 lb. of steam at a value of 20 cents, will show savings of only $52.50 and will require approximately 4½ years to amortize the investment cost.

Conclusion

With our present need for fuels conservation, it would seem that the waste and discomfort caused by water quenching of coke are economically indefensible.

As the savings to be secured and advantages to be gained by dry quenching are so notable, there should be no doubt as to its desirability, and the author hopes the facts he has presented will be of assistance in furthering this economic improvement.

Discussion

R. J. Bender.[19] The writer would ask whether the necessity of using some kind of a steam accumulator has been proved in connection with these dry-quenching plants so as to control the steam output, which is apparently irregular; or are the boilers of sufficient volume to store enough steam?

H. P. Reid.[20] This paper gives an interesting detailed discussion on a simple but novel method of reclaiming heat in the form of steam as a by-product of a large-scale commercial operation, viz., coke production. One interested in conservation of by-products cannot but be impressed by the seemingly extravagant methods of removing heat from coke by the wet-quenching method; likewise, the cooling of other high-temperature products by water spraying.

While the data given in the paper cover the operation from both the coke-cooling and steam-generation angles, it is lacking in some other data which would be interesting. To enumerate, could the author answer the following:

(1) What is the total static pressure drop across the system as indicated by pressures at the inlet and the outlet of the fan? Of course this pressure drop would increase greatly with the increase in percentages of smaller sizes of coke. Are there any data available on these pressure drops with higher percentages of these smaller sizes present in the coke? If so, what are they?

(2) What is the time period required for coke to pass the system?

(3) Have any plants been operated long enough to determine the abrasive effect of the coke dust on (*a*) the superheater elements, (*b*) the boiler tubes, (*c*) the dust collector, and (*d*) the fan blades and housing? If so, what is the life or service period of each of these items?

(4) What average gas velocities are used across the superheaters elements and through the boiler tubes?

(5) The flow of material downward through a hopper or bin such as the coke container of this installation is rather difficult to control so that the flow shall be uniform across any horizontal cross-section of the container. For best boiler efficiency and effective cooling it is evident that this flow should be uniform. Has this system been so designed as to give such uniformity; and if so, how?

(6) Is it possible to admit a constant stream of hot-material charge by a conveyor of some type, rather than by batches, granting that such hot material could be obtained in stream flow rather than by batch, without infiltration of cold air to the system?

R. W. Parkinson.[21] The author shows savings due to the use of the dry-quenching process which will pay the cost of the installation in from one to two years' time. It is quite probable that such large savings can be effected only in more or less ideal cases where there is a continuous demand for moderate pressure steam which will be equal to or in excess of the rated steam output of the proposed dry-quenching plant.

It would of course be possible in the case of an isolated coking plant where there is only a small demand for steam to dry-quench only a portion of the product and thus make only such steam as is required. In most cases, however, this would result in a very small installation, and the economies effected would be small. It would seem that one of the broadest fields for the application of the dry-quenching process would be in large coking plants owned and operated by steel companies when these plants are located on or adjacent to the mill properties. In such cases there is generally a large demand for medium-pressure steam to be used for process and for heating purposes throughout the mills. Ordinarily such a dry-quenching plant would be called upon only to furnish a portion of the steam required, so that the entire output could be used to advantage in this case, and the deficiency in steam could be made up from other sources perhaps in fuel-burning boiler plants using coke breeze. When applied to steel mills, the dry-quenching process must necessarily compete with other sources of waste heat, such as the heat in the exit gases from the open-hearth furnaces and blast-furnace gas from the blast furnaces. In one case comparisons of various methods of producing steam from waste heat have shown that all the heat for the moderate-pressure steam required could be recovered to at least equal advantage by utilizing other waste-heat sources. It is quite clear that in such a case, if the dry-quenching plant can be justified at all, the justification must be based mainly on other advantages claimed by the author, such as the superiority of the dry coke and the lower percentage of coke breeze produced. The author states that the dry coke will have a heating value approximately 1000 B.t.u. per lb. higher than that of wet-quenched coke. This of course will result in higher efficiencies due to the lower moisture losses when burning the dry fuel. This will be a net saving, although the total B.t.u. value of the coke produced per ton of hot coke quenched will of course be little in excess of that for the wet-quenched coke.

With regard to the sizing of the coke, it appears that the coke made by the dry-quenching process runs somewhat smaller in size than the wet-quenched product, and this is considered to be a disadvantage by the majority of blast-furnace operators. From an inspection of drawings presented with the paper, it would appear that this objection could be largely overcome by the manufacturer by means of improvements in the design of the conveying equipment which dumps the hot coke into the top of the dry quencher. This could probably be arranged so that the coke would slide in smoothly and would not have to be

[19] Whiting Corporation, Harvey, Ill. Mem. A.S.M.E.
[20] Special Engineer, Universal Atlas Cement Company, Chicago, Ill.
[21] Allied Engineers, Inc., Jackson, Mich. Jun. A.S.M.E.

dumped twice, first into the skip bucket and then into the coke container through the charging doors. If the rough handling of the coke were eliminated, there is no apparent reason why the dry-quench coke should not be equal in size to the wet-quenched product.

While it is believed that the author is a little optimistic in his expectation that the savings to be realized with a dry-quenching plant will ordinarily pay for the installation in a period of one or two years, it is no doubt true that there are very material savings to be made with the use of this process, and all should feel very much indebted to the author for the detailed account of the progress being made along these lines.

G. R. Strimbeck.[22] Our installation has been in operation only a short time, and any comments must be viewed in the light of that fact. Our experience to date using a mixture of Kentucky King coal from Lisle, Ky., and Wharton coal from Wharton, W. Va., does not show any increase in strength of the coke that would clearly point to a gain inherent to the process of dry quenching. Our shatter tests show only a slight increase, which could be explained by the extra handling the coke receives in our plant due to use of the dry-quenching unit. We find no difference in the sulphur content and very little change in amount of fines produced. The coke does have a better appearance since the volatile matter in any "black" ends is burned out and the silvery appearance is retained due to dry-quenching.

Our biggest problem and one that will be encountered by all users of this process is the matter of dust. When the container is charged, some dust blows over the yard, and a large cloud of dust rises when the container is dumped. Due to the temperature of the coke leaving the unit, it is difficult to eliminate this by any spraying system, and we are trying a dust-collector arrangement. Due to peculiarities of our layout, this dust causes some trouble in our screening system. The writer feels that where new units are installed this trouble could be overcome by allowing time for the coke to cool after leaving the container and then treating with calcium-chloride solution before screening. We expect to try out dust collectors over our screens with final spraying of coke with calcium chloride before it enters trucks or cars for our customers.

In the paper a statement is made that a saving on labor operating car and wharf will result. This is difficult to understand as the hot-car operator would be retained and the man formerly on the wharf would run the dry-quenching unit. That is our arrangement.

The statement is also made that some steam is produced during the four one-hour layover periods. Our experience indicates that this is negligible and that we must keep our regular plant boiler under fire to take care of these drops in steam production. On continuous operation of the battery the steam production fluctuates some, and the writer doubts very much if it would be possible to operate on a dry-quencher boiler only, without some means of controlling the variations in pressure and steam production. At Flint we run our regular boiler, which is equipped with automatic controls, in connection with the dry-quencher boiler and obtain very good charts at the point of steam consumption. To obtain a load sufficient to secure good operation of our regular boiler we have added enough steam-using units to increase our steam consumption by about 15 per cent over last year. These units replace electrically driven ones.

The other data given on the Flint plant are correct. We have not been able to find any hydrogen in our recirculating gases, and we have obtained a steam production which meets the guarantees.

The difficulties we have had in operating the unit are those to be expected from any new equipment, and our costs cannot yet be taken as indicative of long-time operation results. Our power costs are considerably higher than those quoted on the seventh page, due to the fact that we are hoisting the entire hot car. We will undoubtedly save in maintenance of the hot car and the steel work in the vicinity of the quenching station due to the elimination of wet-quenching, but it is difficult to state this saving in dollars.

[22] Superintendent, Consumers Power Company, Flint, Mich.

Author's Closure

In regard to Mr. R. J. Bender's question, the direct answer is "No," as we have not so far felt the need of an accumulator, as the boilers are generally of sufficient size to store steam.

In the smaller-sized plants the steam-generating unit is made oversize and the steam is generated at a fluctuating pressure generally higher than the pressure at which steam is to be used. A regulating valve being installed on the steam outlet permits the steam to flow only at the pressure required. In plants having boilers of the type installed at Flint the steam drum is made of sufficient size to generally act as an accumulator.

Fluctuation in steam pressure is generally affected by the charging, or coke-oven pushing, interval, and the boiler-steam space or reservoir is designed to suit each particular installation. For instance, in a plant having a 25-min. charging interval, the total steam temperatures fluctuation is approximately 8 per cent, while in a plant having a 7-min. charging interval, the fluctuation is approximately $6^1/_2$ per cent.

Answering the questions of Mr. H. P. Reid: (1) The total static pressure drop across the system is approximately 7 in., of water as this has been found to be economic from a fan-operating standpoint. Unless accurate data are available on the percentage of voids in the coke, the calculations are based on an average of 50 per cent voids with not more than 10 per cent breeze or fines through $^1/_2$-in. screen. Data on pressure drops for other conditions are being collected and will be available later.

(2) For such plants as that installed at Flint, the cooling time is 2 hours; that is, the coke container has effective volume to hold 2 hours' production of coke.

(3) Plants have been in successful operation for some years and the question can be answered in detail as follows: (*a*) The superheater elements have so far shown no signs of abrasion or wear due to the abrasive effect of the coke. (*b*) In installations having horizontal fire-tube boilers no wear has been noted in one plant after more than four years of operation. In the vertically inclined type of boiler, as installed at Flint, no wear has so far been noted. (*c*) The dust collectors have so far shown no sign of wear or cutting. (*d*) The fan blades at the Rochester plant are renewed at the time of the yearly plant overhauling, although at other plants the fan blades have run two years. The fan housing has not been replaced at Rochester, and it is in the fifth year of operation.

(4) In general the mean gas velocities through the superheater space and boiler tubes are 60 to 80 ft. per sec. Each installation is calculated separately.

(5) The cooling-gas distributor and the container-discharge opening are designed with a certain amount of adjustment to cause a fairly uniform flow of the coke through the container.

(6) Yes, in special cases by using a rotary or other type of sealing discharge mechanism. One plant is now under development to receive the hot coke from a conveyor. Cold air does not enter the system; there are never two points open to atmosphere at the same time.

In answer to the question of Mr. R. W. Parkinson, the author would say that there are many factors entering into the savings to be effected by dry-quenching of coke, not the least of which is the reduction in plant upkeep due to the elimination of the wet-quenching, very frequently phenol-laden, vapor that is being found very destructive of steel work as well as of the neighbors' property.

Referring to Mr. Parkinson's comment on the sizing of the coke, it will be well to bring out the fact that dry-quenched coke has noticeably less cross-fracture and is stronger size for size than the wet-quenched coke. Uniformity of usable sizes rather than more large pieces is really what the blast-furnace operators want; in fact, crushing and screening to uniform size is now being very extensively demanded by the blast-furnace men. One plant visited by the author is arranging to screen all blast-furnace coke in 3 sizes through $3^1/_2$ in. on $2^1/_2$ in., through $2^1/_2$ in. on $1^3/_4$ in., and through $1^3/_4$ in. in $1^1/_4$ in., charging these different sizes separately.

In conclusion, the author desires to thank the gentlemen who discussed this paper, and to add here that the brick lining of the coke container in one plant is now in its fifth year of operation without one cent of repair or expense and at another plant the steam produced has been more than 1050 lb., F. & A. 212 deg., per net ton of coke cooled.

FSP-53-8

The Performance Rating of Steam-Generating Equipment

By H. L. SOLBERG,[1] LAFAYETTE, IND.

The demands for high capacity, high efficiency, and flexibility have resulted in the development of modern stokers and pulverized-coal burners, adequate furnaces with steam- and water-cooled walls, air heaters, steel-tube economizers, etc., and have completely changed the equipment which was used for steam generation ten years ago. Consequently the author believes it is time that the methods of expressing the performance of such equipment should be revised to conform with new conditions. Considering the problem of steam generation from the viewpoint of the functional purpose of the equipment, it appears to the author that the logical method of expressing the performance of such equipment should be on the basis of the total heat absorbed by all water- and steam-cooled surfaces per unit of time, such a method being general and applicable to all types of steam-generating equipment, and expressing the performance, not in terms of arbitrary dimensions, but of actual results produced.

THE output of steam-generating equipment is generally expressed in terms of boiler horsepower or pounds of steam produced per hour. The second term is indeterminate and unscientific inasmuch as feedwater temperatures, steam pressures, and steam temperatures vary so widely that there may be a difference of 25 per cent in the quantity of heat required to produce a pound of steam in two neighboring plants operating under entirely different conditions. Consequently the term "pounds of steam produced per hour" should be avoided.

The term "boiler horsepower" is unfortunate in that no power is developed in the boiler, nor is there any relationship between boiler horsepower and the power obtainable from the steam in a prime mover. The term originated in 1876 when the Committee of Judges of Steam Boiler Trials at the Centennial Exhibition in Philadelphia defined a boiler horsepower as the evaporation of 30 lb. of water per hour at 100 deg. fahr. into steam at 70 lb. gage, 30 lb. being the average water rate of engines in use at that time. The term was introduced into the A.S.M.E. boiler test code of 1884, and was finally redefined as the evaporation of 34.5 lb. of water per hour at 212 deg. fahr. into steam at 212 deg. fahr. As thus defined, a boiler horsepower is equivalent to the absorption of 33,479 B.t.u. per hr. The term is a misnomer, and the only justification for its continued use is the fact that it has a definite numerical value and is in common use.

Rated Capacity and Per Cent of Rating

The performance curves of steam-generating equipment are ordinarily plotted on a common abscissa of "per cent of rating" or "pounds of steam evaporated per hour." As previously mentioned, the second term is dependent upon several variables and should be avoided. "Per cent of rating," since it is based on an arbitrary "rated capacity," is equally unsatisfactory.

In the A.S.M.E. boiler test code of 1899, the rated capacity of a boiler was defined as follows:

> A boiler rated at any stated capacity should develop that capacity when using the best coal ordinarily sold in the market where the boiler is located, when fired by an ordinary fireman, without forcing the fire, while exhibiting good economy; and further, the boiler should develop at least one-third more than the stated capacity when using the same fuel and operated by the same fireman, the full draft being employed and the fires being crowded; the available draft in the flue just beyond the boiler, unless otherwise understood, being not less than one-half inch water column.[2]

On the basis of this extremely vague definition, boilers were rated at from 8 to 16 sq. ft. per boiler horsepower. The custom gradually developed of rating water-tube boilers on the basis of 10 sq. ft. per boiler horsepower. Consequently, a water-tube boiler, not equipped with a superheater, is operating at 100 per cent of rating when the mean rate of heat transfer is 3348 B.t.u. per sq. ft. per hour. If a boiler is operated at 200 per cent of rating, it does not mean that the boiler is overloaded or subject to excessive thermal strains, but merely that the mean rate of heat absorption is 6696 B.t.u. per sq. ft. per hour. As heat absorption rates in excess of 100,000 B.t.u. per sq. ft. per hour (approximately 3000 per cent of rating) are encountered under favorable conditions in certain parts of boilers, the rated boiler horsepower is not a measure of the maximum capacity of the unit in the same sense that the rating of a turbo-generator is a measure of the power which it can produce continuously. In fact, when units can be run continuously at five times rated capacity and there is no relation between rated capacity and maximum capacity, then the term "rated capacity" is little else than an arbitrary and meaningless one. However, the term "per cent of rating," when applied to boilers not equipped with superheaters, does express in a definite though clumsy way the mean rate of heat transfer, and gives some indication of the conditions under which the boiler is being operated.

Superheaters and Per Cent of Rating

In the case of an installation composed of boiler and superheater, the developed boiler horsepower is based on the total heat absorbed in both the boiler and superheater and may be calculated as follows:

Developed boiler horsepower = (total heat absorbed per hour in boiler plus total heat absorbed per hour in superheater) ÷ 33,479.

The rated horsepower is based on the boiler surface *only*, although the superheater surface is obviously additional heat-absorbing surface. Consequently the per cent of rating does *not* indicate the mean rate of heat transfer in a boiler when a superheater is included in the setting.

Economizers

Curve *A* of Fig. 1 represents the temperature gradient through the water-cooled surfaces of a conventional boiler. It will be noticed that 50 per cent of the heat is absorbed in less than 25 per cent of the surface, and that the last 50 per cent of boiler surface

[1] Associate Professor of Mechanical Engineering, Purdue University. Assoc-Mem. A.S.M.E. The author received his degree of Bachelor of Science in Mechanical Engineering from Purdue University in 1921 and the Master's degree in 1923. Since graduation in 1921, he has been connected with that university.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] Trans. A.S.M.E., vol. 21 (1900), p. 37.

is relatively ineffective. If the last 40 per cent of boiler surface is replaced by economizer surface as shown by curve *B*, Fig. 1, then the rated capacity of unit *B* is only 60 per cent of that of unit *A*. Nevertheless, with the same stoker and furnace, the steam-generating capacity of the two units would be approximately the same. However, the boiler horsepower developed at identical steaming rates would not be the same, since, according to the A.S.M.E. boiler test code, the heat absorbed in the economizer is not considered in figuring the developed boiler horsepower. Why should the economizer be left out

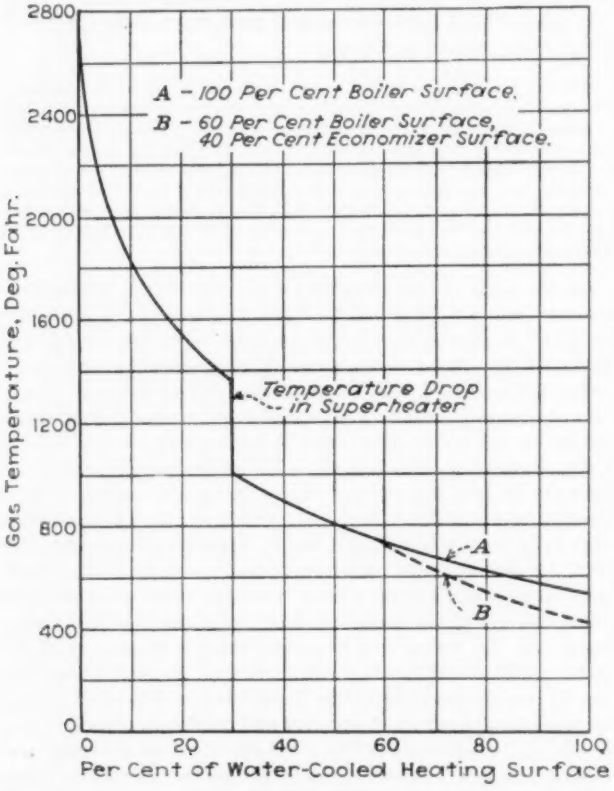


FIG. 1 RELATION OF GAS TEMPERATURES TO PER CENT OF WATER-COOLED HEATING SURFACE PASSED OVER
(Based on Fig. 27, Prime Movers Committee Report No. 25–82.)

of consideration when it is an essential part of unit *B* and one square foot of economizer surface is more effective as a heat absorber than the same amount of surface which it replaces in the conventional boiler? For the same steaming rates, with unit *B* rated at 60 per cent of the capacity of unit *A*, the boiler and superheater are developing about 85 per cent of the boiler horsepower developed in unit *A* and are being operated at a percentage of rating which is about 1.4 times that of unit *A*. The higher "per cent of rating" developed in unit *B* does not indicate that it is being forced more than unit *A*, since for the same steaming rate the mean rate of heat absorption in the boiler of unit *B* is the same as that in the first 60 per cent of unit *A*. The higher percentage of rating in case *B* is obtained merely by replacing relatively ineffective boiler surface by economizer surface. Consequently, the term "per cent of rating" is no indication whatever of the rate of forcing a boiler unless complete data are at hand concerning the arrangement of the heat-absorbing surfaces.

Fig. 2 shows a cross-section of the 11,400-sq. ft. boiler at the 59th Street Station of the Interborough Rapid Transit Company. If this boiler, which is 24 tubes high, were replaced by a 14-high boiler and an economizer of sufficient size to result in about the same flue-gas temperatures as are obtained with the 24-high boiler (assuming that feedwater temperatures will permit this to be done), the unit would be rated at 684 hp. instead of 1140 hp., but could produce the same amount of steam if set over the same furnace and stoker. The larger boiler would be operating at 300 per cent of rating, and the smaller at about 450 per cent when producing the same amount of steam, with identical combustion rates on the stokers and in the furnaces, and with the same heat-transfer rates in the first 6800 sq. ft. of boiler surface. "Per cent of rating" ceases to mean anything under such circumstances unless the entire arrangement of heat-absorbing surfaces is specified.

WATER- AND STEAM-COOLED FURNACE WALLS AND AIR HEATERS

The development of steam- and water-cooled furnaces has resulted in a large increase in capacity brought about by the addition of a relatively small amount of surface. This is especially true in cases where air heaters are employed to maintain the furnace temperature and permit of the absorption of a maximum amount of radiant heat. Fig. 3 shows the remodeled boiler in the Fordson plant of the Ford Motor Company. By doubling the number of pulverized-coal burners and adding air heaters and water-cooled surfaces equal in area to 12 per cent of the original boiler surface, the capacity of the unit was practically doubled. Yet the per cent of rating upon which the performance curves are plotted in Fig. 4 is based upon the original surface of the boiler. In determining the rated capacity of this unit, why is the additional saturated-steam-making surface in the furnace walls not included when it is the most effective surface in the boiler? If it is included, should the rated capacity be figured on the basis of 10 sq. ft. per boiler hp. when the rate of heat absorption is often over five times that of the mean heat-transfer rate of the convection boiler? As yet there is no accepted method of determining the number of square feet of heating surface in the furnace walls and screens. Under such circumstances the rated capacity can be figured in several different ways to give different results which change the "per cent of rating" developed by corresponding amounts. What is the value of "per cent of rating" and "rated capacity" when the unit is operated continuously at five times its rated capacity?

In determining the per cent of rating of a unit such as the Calumet boiler shown in Fig. 5, what is the rated capacity? If based on the boiler surface only, it is 594 hp.; if the furnace surface is included, it is 840 hp. Why should the economizer surface be ignored when it is practically as extensive as the combined furnace and boiler surface? If the economizer were replaced by equivalent boiler surface, then at the same steaming rate the "per cent of rating" would be much lower, although the heat release per cubic foot of furnace volume and the heat transfer per square foot of radiant-heat-absorbing surface remained the same. What is the value of the term "per cent of rating" when applied to units of this type where most of the boiler surface is exposed to radiant heat?

Fig. 6 shows a cross-section of reheat boiler No. 1 on unit No. 3 at the Crawford Avenue Station of the Commonwealth Edison Company. What is the rated capacity of this unit? How is the per cent of rating to be determined? The rating of such units as this on the basis of conventional boiler surface is not only worthless but misleading.

Fig. 7 shows a set of performance curves which was recently published by a prominent stoker manufacturer. In examining these curves, two things are very noticeable: (1) the increase in maximum efficiency of the newer installations, due to better stokers and furnaces and improved arrangements for absorbing the heat, and (2) the higher ratings at which the newer units are

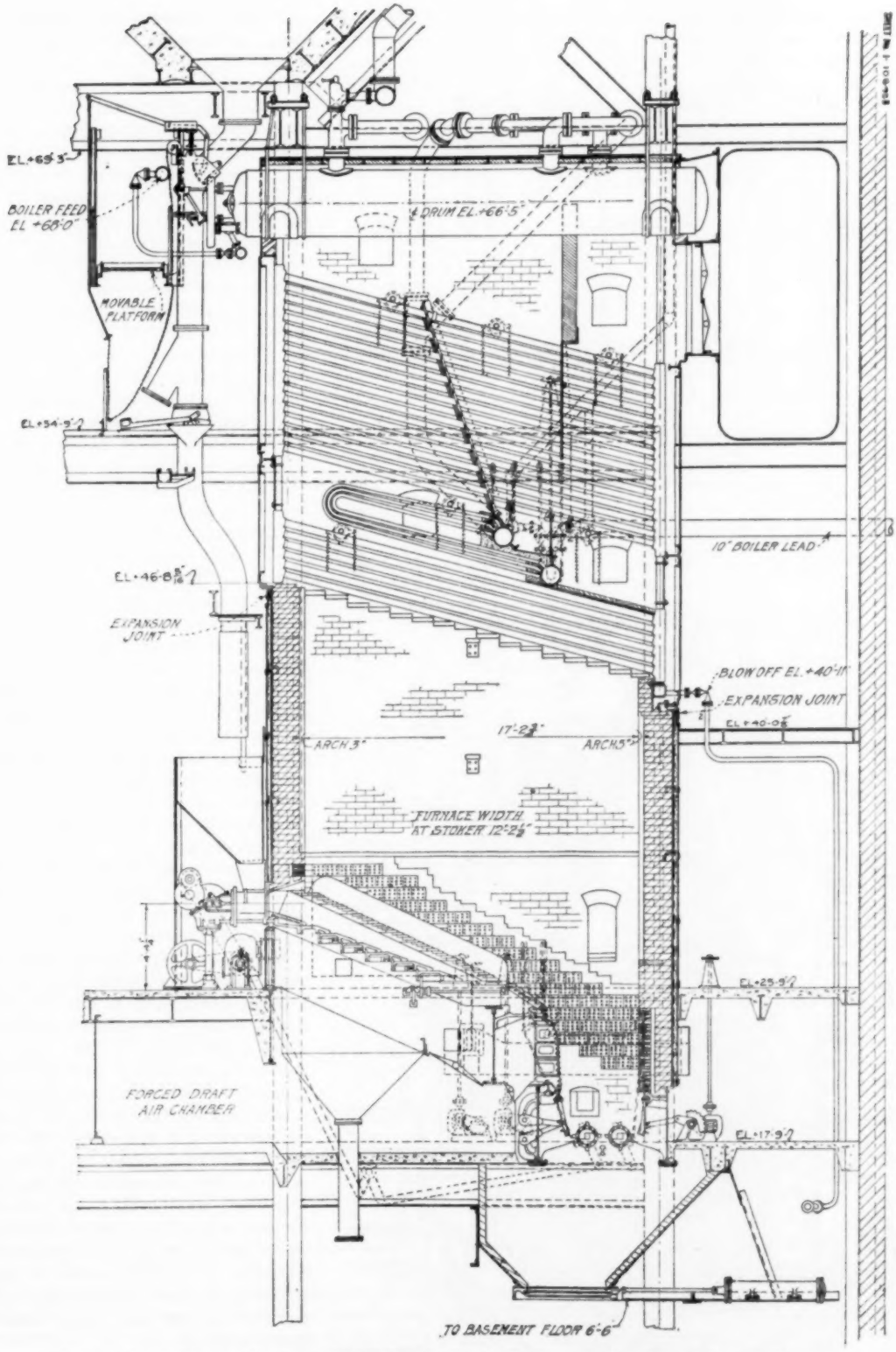


FIG. 2 CROSS-SECTION THROUGH 11,400-SQ. FT. B. & W. BOILER, 59TH STREET POWER STATION, INTERBOROUGH RAPID TRANSIT COMPANY

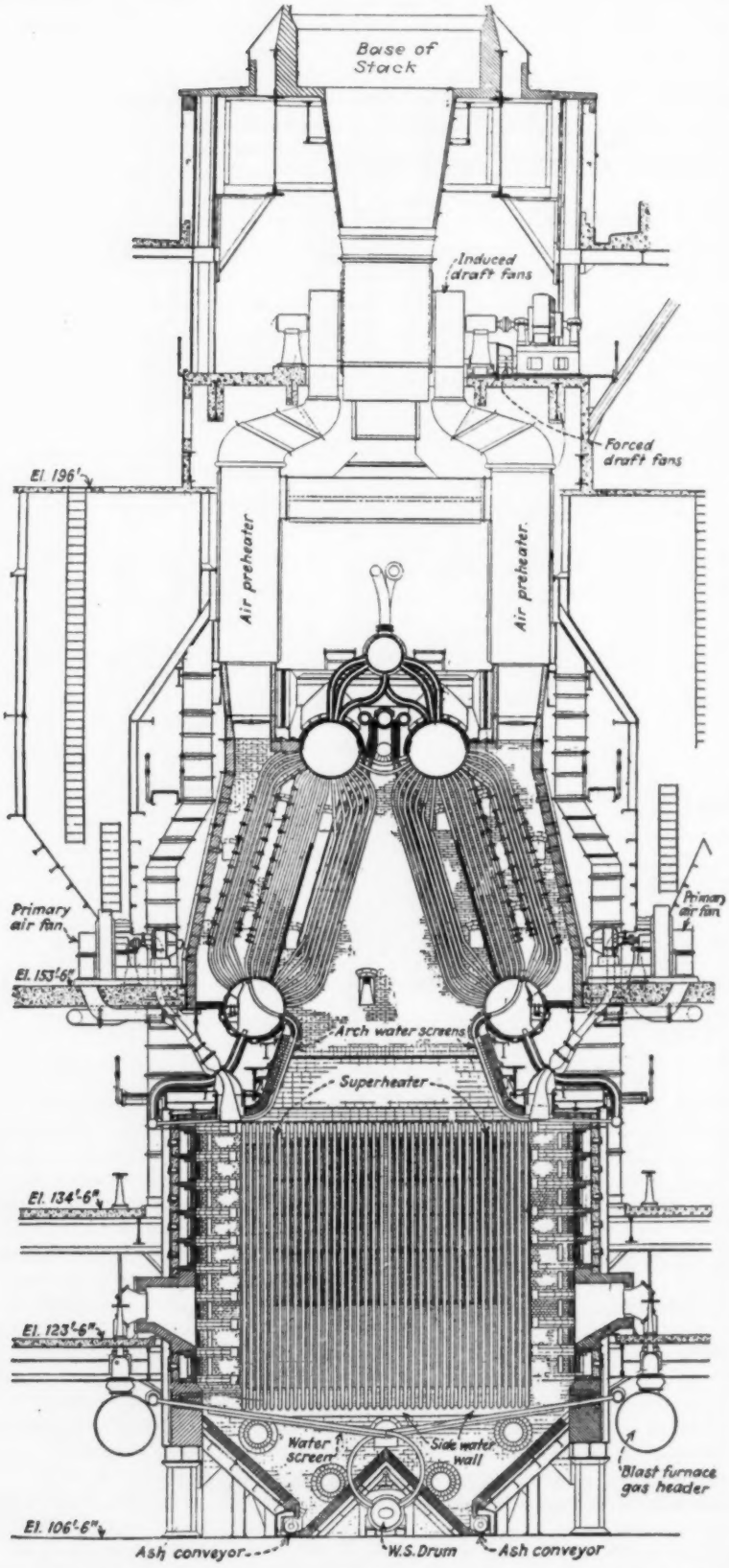


FIG. 3 CROSS-SECTION OF SIDE ELEVATION OF THE REMODELED STEAM-GENERATING UNITS OF THE FORDSON PLANT, FORD MOTOR COMPANY

operating. However, the abscissa upon which these curves are plotted is meaningless. The coal burned per square foot of grate, the heat release per cubic foot of furnace volume, and the maximum rate of heat transfer are no greater in some installations operating at 500 per cent of rating than in others when operating at half of this figure. These high ratings are obtained by eliminating ineffective boiler surface, placing a maximum amount of boiler surface where it is exposed to direct radiation from the fuel bed and flame, and utilizing air heaters and economizers to maintain a high gas temperature throughout the saturated steam-making surfaces. In other words, with adequate fuel-burning capacity, the per cent of rating is a function of the type and arrangement of heat-absorbing surfaces, and curves such as those in Fig. 7, which do not take into account this factor, are misleading.

To summarize, the present system of expressing the performance of steam-generating equipment is open to the following serious objections:

1 The "rated capacity" bears no relation to the maximum capacity and is arbitrarily based on the extent of heat-absorbing surface of one type only without regard to the position of this surface, its effectiveness per unit area as a heat absorber, or the influence of heat-absorbing surfaces of other types.

2 The output, expressed in terms of equivalent evaporation or developed horsepower, as defined in the boiler test code, does not include the heat absorbed in the economizer, although economizer surface serves the same general purpose as boiler surface and is more effective per unit area than the boiler surface which it replaces.

3 "Per cent of rating" is not an indication of the mean rate of heat absorption in a boiler where superheaters are used. Nor does a high mean rate of heat transfer indicate that a boiler is being forced, since it is generally obtained by replacing relatively ineffective boiler surface by other types of heat-absorbing surface.

Modern steam-generating equipment is composed of several different types of heat-absorbing surfaces, combined in so many ways and proportions that the performance can no longer be based upon boiler surface alone. The boiler, superheater, economizer, and reheater are designed and operated as a unit, and must be so considered in determining the performance of steam-generating equipment. The function of the boiler, superheater, and economizer is to absorb heat from the products of combustion in order to generate steam. The function of the air heater is to return to the furnace as much as possible of the heat not absorbed by the steam- and water-cooled surfaces. Therefore, the logical method of expressing the performance of steam-generating equipment is in terms of total heat absorbed by water- and steam-cooled surfaces.

THE UNIT OF CAPACITY

There are three units of capacity used in the A.S.M.E. boiler test code: (1) the unit of evaporation; (2) equivalent evaporation from and at 212 deg. fahr.; and (3) boiler horsepower. The

unit of evaporation is the absorption of 1000 B.t.u. per hour; the unit of equivalent evaporation is the absorption of 970.4 B.t.u. per hour, based on Marks and Davis steam tables; the unit of boiler horsepower is the absorption of 33,479 B.t.u. per hour. Of the three, the unit of evaporation is the simplest, easiest to calculate and understand.

The output of steam-generating equipment can be most easily expressed as follows:

$$\text{Units of evaporation developed} = \frac{\text{Weight of water evaporated per hour} \times \text{total heat absorbed per lb. in economizer, boiler, superheater, and reheater}}{1000}$$

This differs from the present test-code method in that the output is expressed in terms of total heat absorbed in all steam- and water-cooled surfaces. The output could be expressed in boiler horsepower by dividing the numerator by 33,479 instead of 1000. However, the name is a misnomer and the unit is clumsy, so that it would be better to eliminate the term entirely. The output could also be expressed in terms of equivalent evaporation by dividing the numerator by the latent heat of steam at standard atmospheric pressure, instead of by 1000. However, this is an unwieldy unit and, as generally computed, requires the calculation of a useless term called "factor of equivalent evaporation." The simplest method of expressing output is in terms of thousands of B.t.u. absorbed per hour by all steam- and water-cooled surfaces.

Rated Capacity

The rated capacity should be based upon the maximum continuous load which a unit can carry. Otherwise the term is arbitrary and meaningless. The maximum capacity of a steam-generating unit supplied with good feedwater is generally limited by the amount of fuel that can be burned rather than by the heat-absorbing capacity. On account of the flexibility of such equipment, the numerous factors which may limit the capacity, and the manner in which characteristics of the fuel and operating conditions affect performance, it is necessary to consider the entire fuel-burning, draft-producing, and heat-absorbing equipment as a unit and rate its capacity on the basis of the quantity of heat that can be transferred to steam- and water-cooled surfaces per hour under conditions of maximum constant-load operation when supplied with the fuel which is normally burned in the plant. In other words, rated capacity should be based upon actual performance and not upon arbitrary, empirical standards which bear no relation to the actual capacity of the unit.

It may be urged that the rated capacity of a unit when figured in this manner depends upon the kind of fuel burned and other operating conditions and cannot be determined very closely until the unit has been placed in operation. However, steam-generating equipment is generally purchased on the basis of the quantity of steam which it can produce per hour when burning coal of known characteristics. This result, expressed in thousands of B.t.u. absorbed per hour in order to eliminate local operating conditions, should constitute the rating of the unit. It might be necessary to revise the rating somewhat after the unit has been placed in operation because of conditions not foreseen by the designer. However, as data continue to accumulate on the performance of modern steam-generating equipment, it should be possible to estimate in advance, with reasonable accuracy the maximum continuous capacity of a unit. A rating so determined is certainly more satisfactory and indicative of the actual capacity of a steam-generating unit than a rating determined by present arbitrary methods which gives results which are often not within 500 per cent of the actual capacity of the unit.

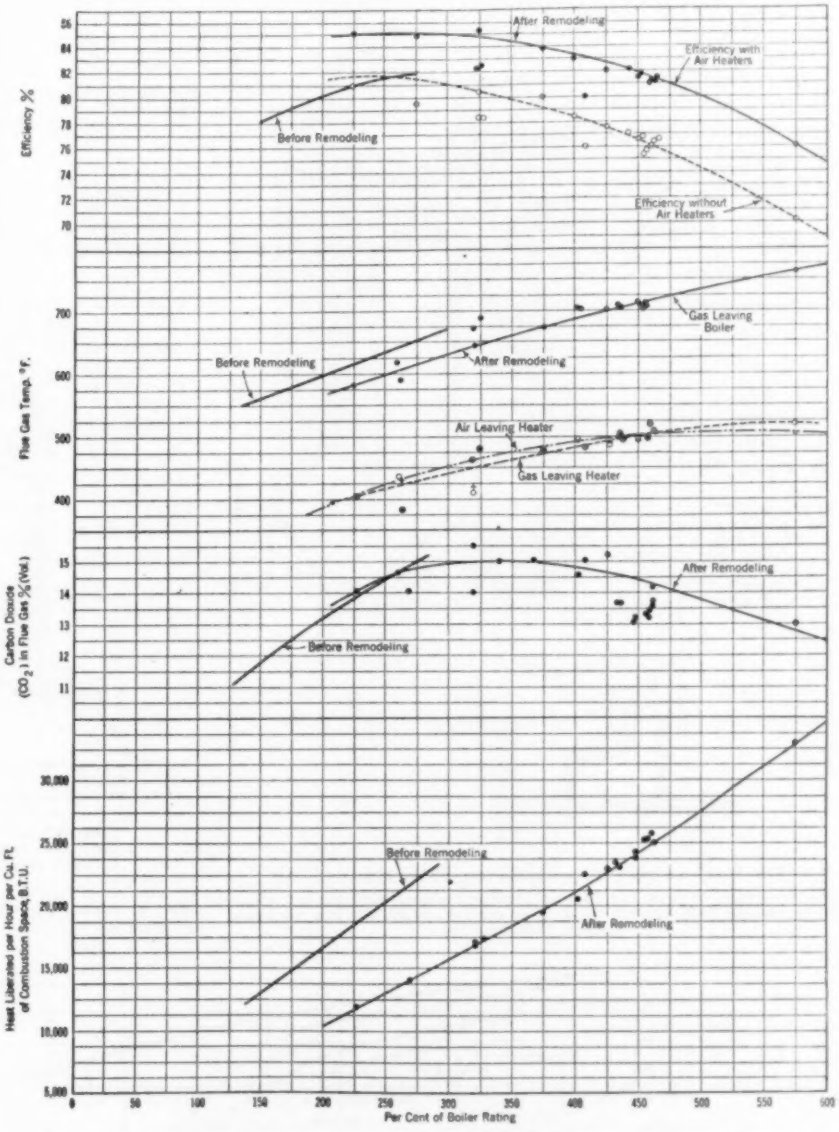


FIG. 4 COMPARATIVE PERFORMANCE OF NO. 2 UNIT OF FORDSON PLANT BEFORE AND AFTER REMODELING

(From publication 1 C-2 of Combustion Engineering Corporation; 500,000 lb. of steam per hour from one unit.)

Performance Curves

Performance curves should be plotted on a common abscissa

which has a definite numerical value, is readily understood, and leaves no opportunity for misinterpretation or false comparisons. The objections to "per cent of rating" and "steam produced per hour" have been considered. Fig. 8 includes a set of performance curves plotted on a common abscissa of "total heat absorbed per hour in thousands of B.t.u. by water- and steam-cooled surfaces." Other data such as draft losses, gas temperatures, heat absorbed in various parts of the system, etc., can be plotted on the same abscissa. The essential advantage of a set of curves plotted in this manner over the conventional curves plotted on per cent of rating is that the abscissa has a definite numerical value and

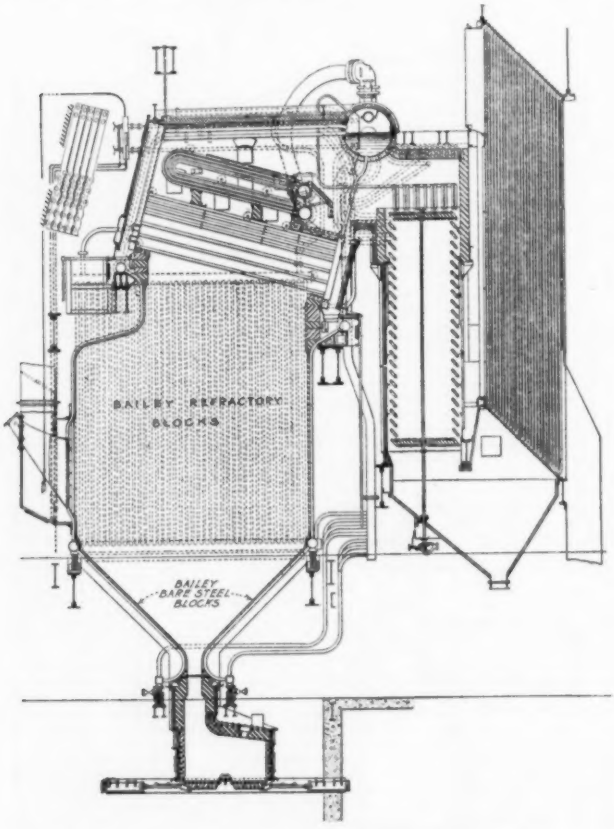


FIG. 5 CALUMET PULVERIZED-FUEL-FIRED BOILER UNIT
(Boiler surface, 5938 sq. ft.; furnace heating surface, 2460 sq. ft.; economizer surface, 8365 sq. ft.; total water surface, 16,763 sq. ft.; air-heater surface, 41,700 sq. ft.)

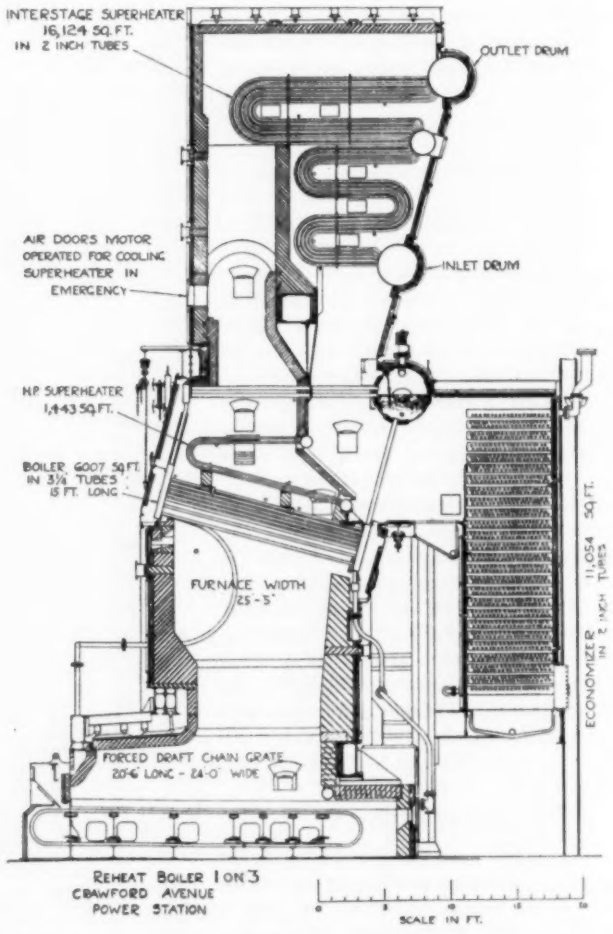


FIG. 6 CROSS-SECTION OF REHEAT BOILER NO. 1 ON UNIT NO. 3, CRAWFORD AVENUE STATION, COMMONWEALTH EDISON COMPANY

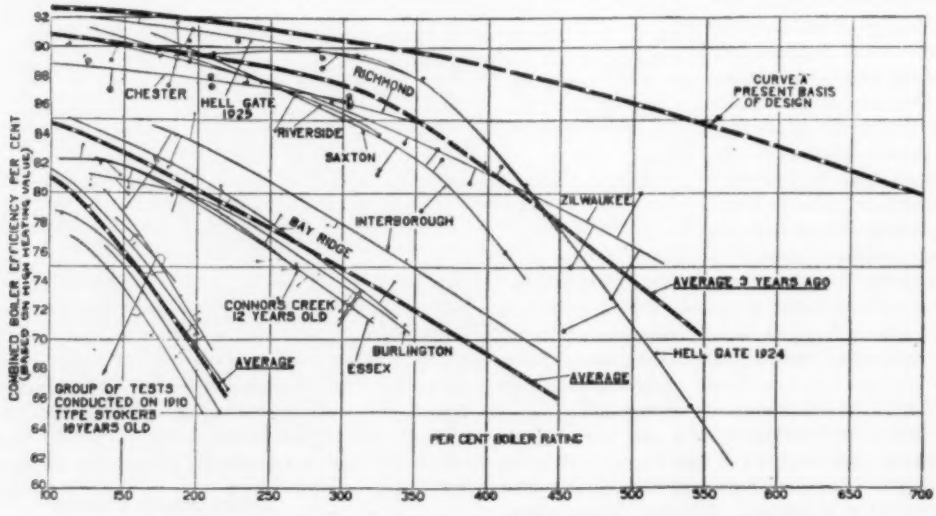


FIG. 7 PERFORMANCE CURVES OF STEAM-GENERATING EQUIPMENT

shows not only the range of capacity but the actual capacity developed.

The curve of mean rate of heat transfer is plotted for the boiler only. It is possible, from this curve, to estimate the extent to which the unit is being forced if the amount and position of each type of heat-absorbing surface is known and a drawing of the setting is available. However, without this supporting information, the curve is of little value as an indication of the rate of forcing of the unit. Similarly, the curve showing the heat release per cubic foot of furnace volume is of questionable value until the kind of coal, particularly its fusing temperature, the excess air supplied for combustion, and the position and extent of all radiant-heat-absorbing surfaces are known.

Few fields of mechanical engineering have witnessed so many progressive changes in the past ten years as have taken place in the field of steam generation. The demands for high capacity, high efficiency, and flexibility have resulted in the development of modern stokers and pulverized-coal burners, adequate furnaces with water- and steam-cooled walls, air heaters, steel-tube economizers, etc., and have completely changed the equipment which was used for steam generation ten years ago. Consequently it is time that the methods of expressing the performance of such equipment, particularly in regard to rated capacity and graphical representation of results, should undergo a revision to bring them into conformity with new conditions. Considering the problem of steam generation from the viewpoint of the functional purpose of the equipment, it appears that the logical method of expressing the performance of such equipment should be on the basis of the total heat absorbed by all water- and steam-cooled surfaces per unit of time. Such a method is general and applicable to all types of steam-generating equipment, and expresses the performance, not in terms of arbitrary dimensions but of actual results produced.

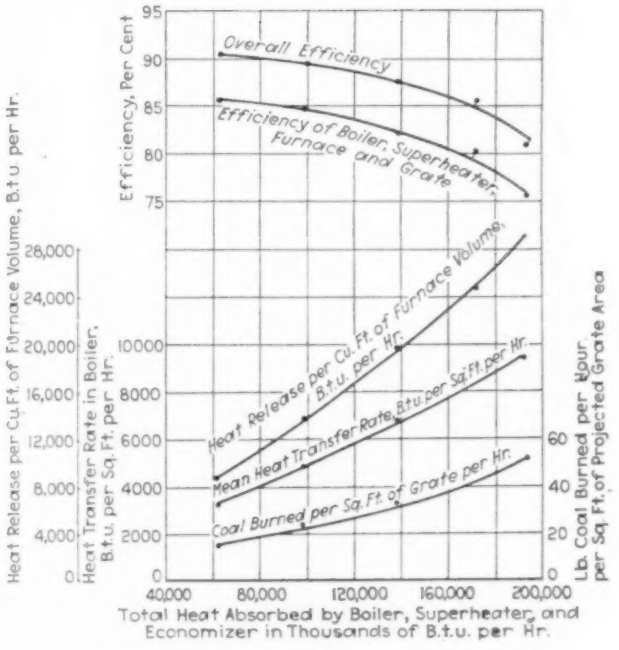


FIG. 8 PERFORMANCE CURVES OF UNIT NO. 49, RICHMOND STATION
(Based on data from Prime Movers Committee Report No. 267-33.)

FSP-53-9

Flow Characteristics of Special Fe-Ni-Cr Alloys and Some Steels at Elevated Temperatures[1]

By H. J. FRENCH,[2] WILLIAM KAHLBAUM,[3] and A. A. PETERSON[4]

In this paper the authors give the results of "creep" tests at different temperatures for three groups of alloys. The eleven metals in the first group included commercial alloys of nickel, chromium, and iron, both with and without tungsten, and low-chromium steels containing also tungsten, vanadium, or molybdenum. The second group comprised two carbon steels, a 3½ per cent nickel steel, and two low-nickel-chromium steels which were tested only at 700 deg. fahr.; the twelve alloys of the third group were melted in a high-frequency induction furnace and their compositions were selected to show the general trends at 1000 deg. fahr. in the load-carrying ability of castings of the nickel-chromium-iron system. A metallographic study of the creep-test specimens revealed intercrystalline weakness in some of the wrought nickel-chromium-iron alloys, especially at temperatures between 1160 and 1390 deg. fahr. A study was also made of the effect of deformation in the creep tests at different temperatures on the hardness and impact resistance of a chromium-vanadium steel at atmospheric temperatures.

I—INTRODUCTION

CONSIDERABLE attention is now being given in both this country and Europe to "creep" tests (or flow tests) in which the time-deformation relationship is determined for metals subjected to constant loads for long periods at approximately constant temperatures. Previous tests at the Bureau of Standards have related jointly to the development of a testing technique and to the procurement of data intended to assist engineers in the selection of safe working stresses for equipment designed to operate at elevated temperatures. Correlations have also been made between the laborious flow tests and short-time tension tests with the object of finding, if possible, rapid methods of determining approximately the load-carrying ability of metals at different temperatures.

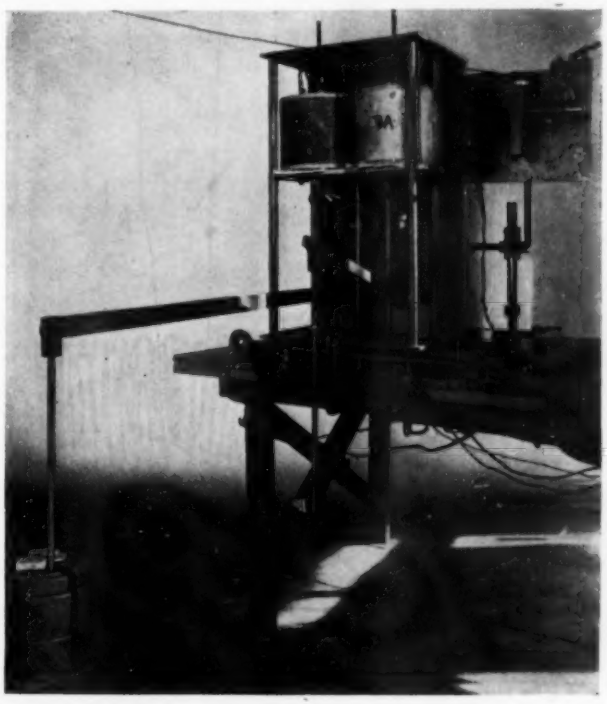


FIG. 1 TWO OF THE VERTICAL TEST UNITS FOR FLOW TESTS AT DIFFERENT TEMPERATURES

(Four vertical and six horizontal test units constituted the testing equipment for the flow tests. The optical micrometer used in making length measurements is shown on the stand at the right.)

So far, data have been published for only five steels, and the purpose of this report is to present the results of additional tests extending over several years so that comparisons can be made of a wider range of metals, based on data secured under comparable conditions in one laboratory. The work simply represents the extension to more materials of the methods of test already described in detail.[5, 6, 7] (See page 2 for footnotes 6 and 7.)

[1] Published with the approval of the Director of the Bureau of Standards of the U. S. Department of Commerce.

[2] Formerly Senior Metallurgist, Bureau of Standards, Washington, D. C. Now Metallurgist, Development and Research Department, The International Nickel Company, Bayonne, N. J. Mr. French was graduated from the School of Mines, Columbia University, in 1915, afterward engaging successively in chemical work at the Murray, Utah, plant of the American Smelting and Refining Co., in metallurgical work for an Eastern automobile and airplane engine plant, and in metallurgical production and inspection in the New York, Philadelphia, and Pittsburgh districts for the Bureau of Aircraft Production, U. S. A. In 1919 he joined the metallurgical division of the U. S. Bureau of Standards, where he was in charge of the section of Thermal Metallurgy and later assistant chief of the division.

[3] Research Associate, The Midvale Company, Philadelphia, Pa. Mr. Kahlbaum was graduated in chemical engineering from the Missouri School of Mines and Metallurgy of the University of Missouri in 1920 and received a degree in Metallurgical Engineering from that institution in 1925. After graduation be held the position of research chemist and metallurgist with the Missouri State Mining Experiment Station cooperating with the Mississippi Valley Station of the U. S. Bureau of Mines.

[4] Scientific Aide, Bureau of Standards, Washington, D. C. Mr. Peterson studied at Carnegie Institute of Technology and at George Washington University. He served on the metallurgical staff of the Bureau of Standards from February, 1922, until June, 1929.

Contributed by the Joint Research Committee on the Effect of Temperature on the Properties of Metals and presented at the Semi-Annual Meeting, Detroit, Mich., June 9 to 12, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[5] H. J. French and W. A. Tucker, "Flow in a Low-Carbon Steel at Various Temperatures," Technologic Papers of the Bureau of Standards, No. 296 (1925).

Tests were made of three groups of metals; the first, comprising commercial nickel-chromium and nickel-chromium-iron alloys and low-chromium steels containing tungsten, vanadium, or molybdenum, were tested, with one exception, under the research associate plan[5] with a research associate from The Midvale Company, Philadelphia, Pa. The second group consisting of two carbon steels, a $3^1/_2$ per cent nickel steel, and two low-nickel-chromium steels were tested at 700 deg. fahr. to assist the Bethlehem Steel Company, Bethlehem, Pa., and the Combustion Engineering Corporation, New York, N. Y., in the selection of a composition suited for particular requirements of power-plant construction. The metals of the third group were used in a study of the flow characteristics at 1000 deg. fahr. of alloys of the nickel-chromium-iron system.

II—TEST METHODS

The test methods employed were substantially the same as those used previously, but subsequent to the last published de-

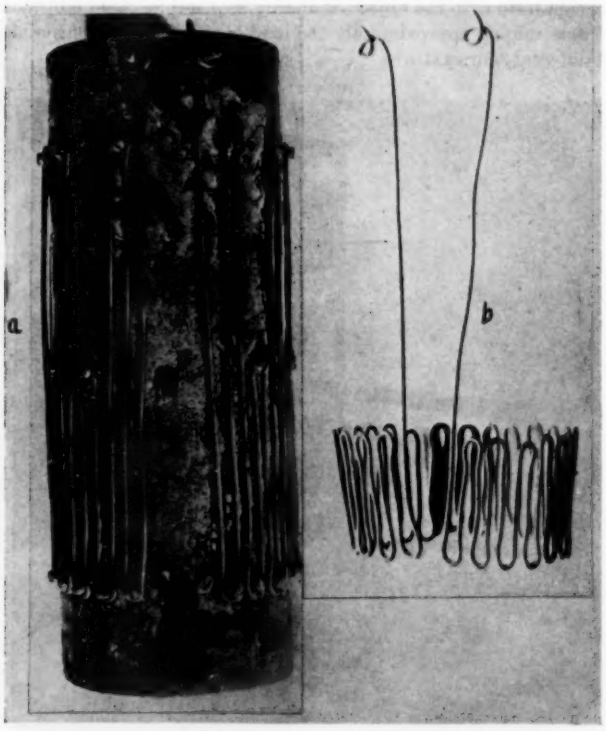


FIG. 2 IMPROVED RESISTORS OF THE HEATING UNITS
a = main resistor on the forming fixture
b = auxiliary resistor placed outside and at the lower end of the main resistor

scriptions a number of modifications were incorporated in the equipment and test procedure. Four of the ten horizontal loading machines were replaced by vertical units (Fig. 1) in which the frictional losses of the mechanical lever system were materially reduced. The new units were therefore better adapted than the old horizontal units for tests at the highest temperatures where the applied loads were the smallest.

[6] H. J. French, "Methods of Test in Relation to Flow in Steels at Various Temperatures," Proc. Am. Soc. Test. Matls., vol. 26, (1926), p. 7.

[7] H. J. French, H. C. Cross, and A. A. Peterson, "Creep in Five Steels at Different Temperatures," Technologic Papers of the Bureau of Standards, No. 362 (1928).

[8] Described in Circular 296 of the Bureau of Standards.

The temperature fluctuations within the test specimens were reduced by placing the control thermocouples at the furnace windings instead of in the test specimens. Resistors of improved form (Fig. 2) were installed in the furnaces of the vertical test units to reduce the temperature gradients from top to bottom in the test specimens. In order to secure greater accuracy in the length measurements with the metals which oxidized most readily, the practice was followed of using platinum or silver wire (32 B. & S. gage) in small grooves as reference points for length measurements with an optical micrometer instead of the edge of the enlarged end section of the specimen.

Only a limited quantity of some of the metals was available. The desire to secure the maximum information under such conditions led to the adoption of a specimen having two different diameters within the customary 2-in. gage length as is shown in Fig. 3. A given applied load produced different unit stresses in the two halves of the gage length of such a specimen, and this was equivalent to two of the customary specimens. However, the use of the modified test specimen is not advocated under ordinary conditions. It is disadvantageous to reduce the measured length from 2 in. to 1 in., and the abrupt change in section at the center of the 2-in. gage length may introduce stress concentrations and also tend to increase the temperature variations in the test sections. As a result the flow produced by a given stress in the

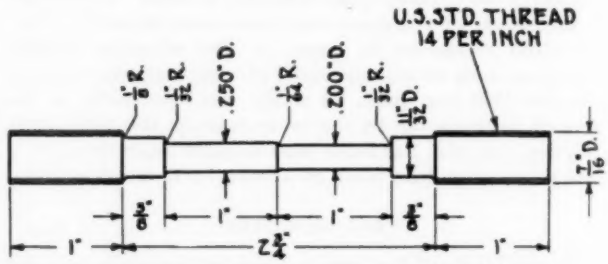


FIG. 3 MODIFIED FLOW-TEST SPECIMEN USED IN SOME OF THE TESTS
(The flow under two different stresses can be followed on the one test specimen. See text for discussion.)

large half was not always as close as could reasonably have been expected to that produced by the same stress in the small half of the gage length. In cases of doubt many of the tests were repeated on test bars having a gage length of uniform diameter.

III—MATERIALS TESTED

The composition and other details of the metals tested are summarized in Table 1. The eleven metals of the first group may be divided into three classes; the first comprised the one alloy, K-4, which was prepared for the Joint Research Committee on Effect of Temperature on the Properties of Metals[9] to represent the alloy 80 nickel, 20 chromium, widely used as resistance wire. Its composition is actually not representative of the bulk of the metal used in such wires, but it should serve to indicate in a general way the flow characteristics of this class of material. The second class in the first group comprised nickel-chromium-iron alloys both with and without appreciable proportions of tungsten, while the third class consisted of low-alloy steels having not more than 4 per cent of alloying elements.

The six nickel-chromium-iron alloys in the first group represent compositions which have been used industrially in the construction of chemical or engineering equipment operating at high temperatures. Alloys E1894 and E1567 were made in this country and the test bars from heat E1894 were taken from the riser of

[9] A committee sponsored by the American Society for Testing Materials and The American Society of Mechanical Engineers.

TABLE 1 CHEMICAL COMPOSITION AND OTHER DETAILS OF THE METALS TESTED

Designation	Chemical Composition—Per Cent[1] C	Mn	P	S	Si	Ni	Cr	W	Mo	V	Fe	Cu	Specimens taken from	Treatment before test	Properties at room temperature: Yield point, 1000 lb. per sq. in.	Tensile strength, 1000 lb. per sq. in.	Elongation in 2 in., per cent	Reduction of area, per cent	Brinell number
GROUP I—TESTED AT DIFFERENT TEMPERATURES																			
K4	0.31	2.79	...	...	0.27	76.0	18.8	..	..	..	0.80	0.66	Rolled ⅞ inch bars	1900 deg. fahr.—3 hr. air cooled	39.5	116.5	37.5	50.3	...
E1894	0.25	1.40	0.028	0.020	0.06	60.8	10.75	2.8	..	..	..	..	Riser from casting	Annealed 1470 deg. fahr.	23.0	75.2	10.0	11.5	...
FS-1 Lot A	0.19	1.62	0.028	0.027	0.10	57.6	10.28	3.19	..	..	..	..	Rolled bars	As rolled	63.0	112.5	30.0	36.9	207
FS-1 Lot B	0.20	1.66	0.026	0.029	0.08	57.4	10.34	3.30	..	..	..	..	Rolled bars	Annealed 1650 deg. fahr.	57.5	110.0	31.0	51.6	200
FS-2 Lot A	0.32	1.52	0.028	0.038	0.03	34.9	12.93	tr.	..	..	..	..	Rolled bars	As rolled	68.0	100.0	33.0	54.10	181
FS-2 Lot B	0.28	1.45	0.032	0.027	0.03	35.9	11.12	0.34	..	..	..	..	Rolled bars	Annealed 1650 deg. fahr.	54.0	99.5	33.0	47.7	172
E1567	0.28	1.50	0.024	0.018	0.07	38.6	10.64	1.77	..	..	..	..	As rolled		69.0	120.5	24.0	44.9	...
EE1114	0.29	0.50	...	...	1.18	8.0	19.86	4.5	..	..	..	..	Experimental melt—forged						
E1912	0.15	0.35	0.020	0.021	0.22	8.46	17.32	..	..	..	..	..	Rolled bars	Lot A	40.0	95.0	45.0	60.0	...
														Lot B—1850 deg. fahr.—water	64.8	112.2	54.5[3]	63.5[3]	...
10-518 Lot A	0.40	0.57	...	...	0.25	...	2.26	..	..	0.20	..	..	Rolled bars	1575 deg. fahr.—1 hr. slow cool; 1375 deg. fahr.—1 hr. furnace	54.2	87.2	28.3	69.2	...
10-518 Lot B	0.40	0.57	...	...	0.25	...	2.26	..	..	0.20	..	..	Rolled bars	1575 deg. fahr.—1 hr. air; 1300 deg. fahr.—1 hr. furnace	86.0	105.6	23.7	68.2	...
EE1139	0.46	0.37	...	...	0.19	...	2.2	1.65	..	0.25	..	..	Experimental melt (rolled)	1400 deg. fahr.—2 hr. slow cool; 1740 deg. fahr.—1 hr. oil; 1000 deg. fahr.—1 hr. air cool	207.0	231.5	7.8	22.8	444
E1549	0.53	0.95	0.037	0.015	0.23	...	1.07	..	0.44	..	..	..	Forged bars	1550 deg. fahr.—1 hr. oil; 1200 deg. fahr.—4 hr. slow cool	124.0	143.0	18.7[4]	56.0[4]	...
E1490	0.75	0.52	0.028	0.016	0.31	0.15	1.45	..	0.54	0.27	..	..	Forged bars	1550 deg. fahr.—1 hr. oil; 1100 deg. fahr.—4 hr. slow cool	..	174.0	7.5[4]	16.7[4]	...
GROUP II—TESTED AT 700 DEG. FAHR. (370 DEG. CENT.)																			
4H451	0.23	0.48	0.015	0.024	0.18	0.05	...	..	..	..	..	..	6" x 6" rolled blooms[2]	1650 deg. fahr.—4 hr. furnace; 1600 deg. fahr.—4 hr. air; 1330 deg. fahr.—4 hr. furnace	39.0	60.0	17.5	29.2	...
2H384	0.45	0.55	0.016	0.027	0.15	0.24	...	..	..	..	..	..	5" x 5" rolled blooms[2]	1650 deg. fahr.—4 hr. furnace; 1500 deg. fahr.—4 hr. air; 1330 deg. fahr.—4 hr. furnace	42.5	80.0	17.0	27.5	...
4H247	0.37	0.68	0.018	0.023	0.21	3.46	...	..	..	..	..	..	6" x 5" rolled blooms[2]	1650 deg. fahr.—4 hr. furnace; 1500 deg. fahr.—4 hr. air; 1360 deg. fahr.—4 hr. furnace	57.0	97.0	16.0	25.1	...
1H391	0.36	0.63	0.014	0.022	0.19	1.25	0.62	..	..	..	..	..	6" x 6" rolled blooms[2]	1650 deg. fahr.—4 hr. furnace; 1500 deg. fahr.—4 hr. air; 1280 deg. fahr.—4 hr. furnace	55.0	94.0	15.0	17.0	...
1H396	0.35	0.49	0.017	0.017	0.19	1.66	0.99	..	..	..	..	..	5" x 4" rolled blooms[2]	1650 deg. fahr.—4 hr. furnace; 1500 deg. fahr.—4 hr. air; 1320 deg. fahr.—4 hr. furnace	52.0	93.0	24.0	47.7	...
GROUP III—TESTED AT 1000 DEG. FAHR. (540 DEG. CENT.)																			
1000A	0.19	0.75	...	...	0.69	...	...	..	..	..	bal.[5]	..	1½-in. octagonal ingots weighing about 3 lb.	1650 deg. fahr.—½ hr. air cool	..	...	..	..	150
1001D	0.25	0.80	...	...	0.89	bal.	...	..	..	..	..	..		1500 deg. fahr.—½ hr. air cool	..	...	..	..	114
1002A	0.24	0.88	...	...	0.52	68.5	...	..	..	..	bal.[5]	..		1650 deg. fahr.—½ hr. air cool	..	..	..	..	131
1003A	0.18	0.75	...	...	0.57	30.6	...	..	..	..	bal.[5]	..		1450 deg. fahr.—1 hr. furnace cool	..	...	..	..	147
1004A	0.37	0.67	...	...	0.53	...	14.5	..	..	..	bal.[5]	..		1650 deg. fahr.—2 hr. furnace cool	..	...	..	..	298–319
1004B	0.24	0.59	...	...	0.39	...	14.6	..	..	..	bal.	..		1650 deg. fahr.—2 hr. furnace cool	..	...	..	..	219–222
1005B	0.38	0.73	...	...	0.82	...	30.2	..	..	..	bal.[5]	..		1800 deg. fahr.—2 hr. air cool	..	...	..	..	217
														1900 deg. fahr.—3 hr. air cool	..	...	..	..	363–388
1006B	0.49	0.68	...	...	1.01	...	55.3	..	..	..	bal.[5]	..							
1007B	0.51	0.74	..	...	0.75	78.2	18.3	..	..	..	..	..		1900 deg. fahr.—3 hr. air cool	..	...	..	..	183
1008B	0.24	0.88	...	...	1.14	49.7	46.3	..	..	..	..	..		1900 deg. fahr.—3 hr. air cool	..	...	..	..	285
1009A	0.41	0.53	...	...	0.78	58.7	14.8	..	..	..	..	..		1900 deg. fahr.—3 hr. air cool	..	...	..	..	138
1010A	0.11	0.54	...	...	0.77	25.5	16.4	..	..	..	..	..		2000 deg. fahr.—1 hr. air cool	..	...	..	..	113
1011A	0.11	0.59	...	...	0.87	35.4	27.2	..	..	..	bal.[5]	..		1900 deg. fahr.—3 hr. air cool	..	...	..	..	144
GROUP IV—STEELS PREVIOUSLY TESTED (INCLUDED FOR CONVENIENCE OF GENERAL COMPARISON)																			
.....	0.24	0.37	0.021	0.028	0.01	...	...	..	..	..	..	..	1-in. hot-rolled plate		33.0	66.7	30.0	59.2	...
K3	0.39	0.51	0.015	0.029	0.19	...	0.87	..	0.21	..	..	..	1¼-in. diam. rolled bars	1625 deg. fahr.—2 hr. air; 1550 deg. fahr.—1½ hr. oil; 1300 deg. fahr.—2 hr. oil	..	98.5	22.0	70.9	...
A336	0.77	0.24	0.031	0.035	0.42	0.14	3.9	13.6	..	1.9	..	..	1-in. diam. rolled bars	Tested "as received"—probably annealed	95.7	110.1	17.0[3]	36.5[3]	...
A340	0.28	0.38	0.026	0.013	0.17	...	20.5	..	..	..	..	0.98	½-in. diam. rolled bars	1830 deg. fahr.—water; 1200 deg. fahr.—air	67.3	89.8	21.5[3]	55.6[3]	...
A347	0.24	0.53	0.009	0.005	2.96	23.3	18.05	..	..	..	..	..	1¼-in. diam. rolled bars	1450 deg. fahr.—2 hr. furnace cool	49.0	110.0	33.0	47.5	...

[1] As reported by the maker for Groups I and II; by the Bureau of Standards, H. A. Bright, for Groups III and IV.
[2] Specimens cut at 90 deg. to the direction of rolling, half-way between center and surface of blooms at some distance from the ends.
[3] Determined on tensile-test specimens with a diameter of about 0.25 in. and a gage length of 2 in.
[4] Determined on tensile-test specimens with a section ⅝ in. × 0.1565 in.
[5] Containing impurities such as small amounts of phosphorus and sulphur.

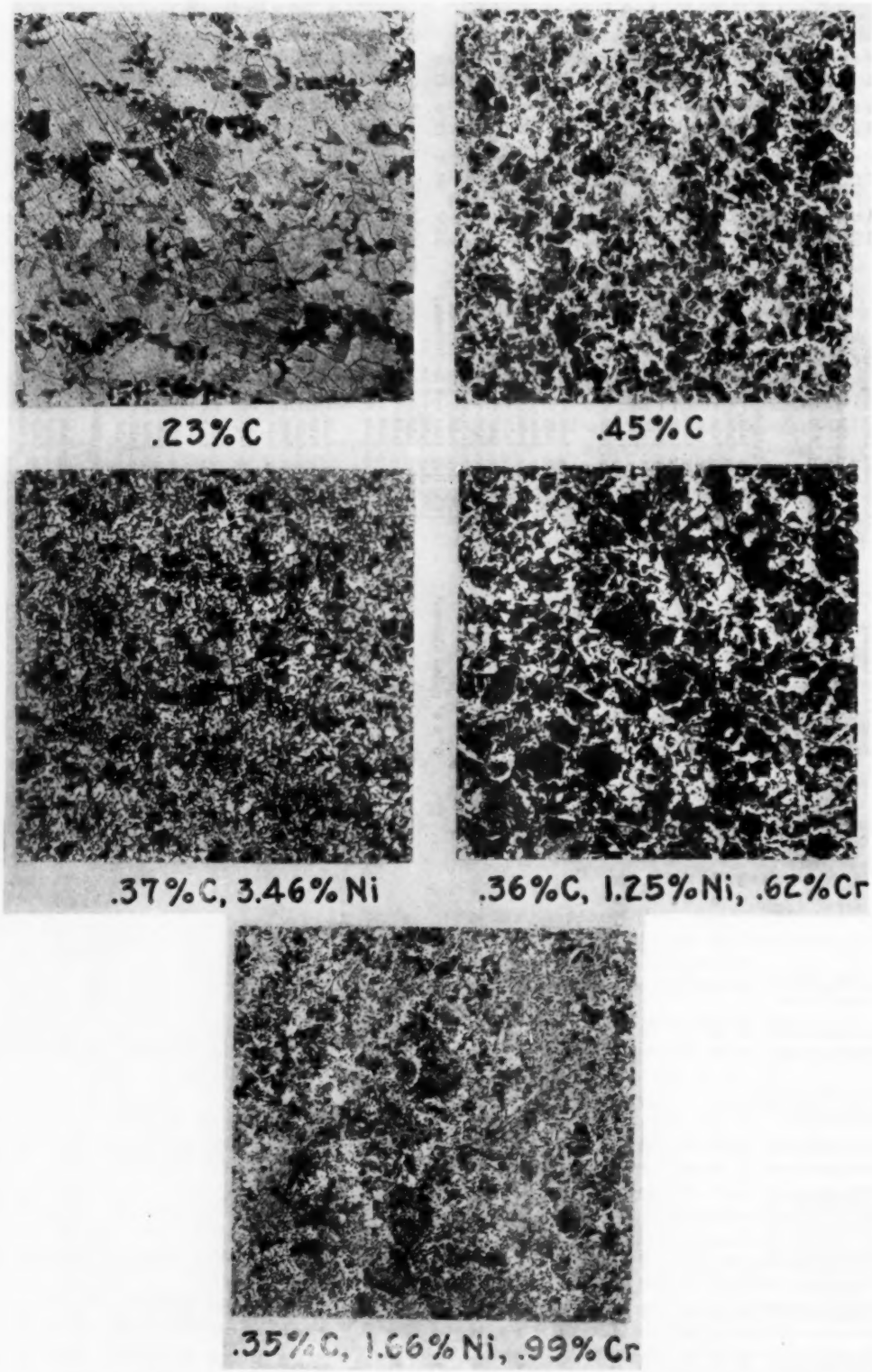


FIG. 4 MICROGRAPHS OF THE STEELS OF GROUP II, TABLE 1, IN THE CONDITIONS TESTED. ×100
(These photographs supplied by the maker. Etched with 3 per cent HNO_3 in alcohol.)

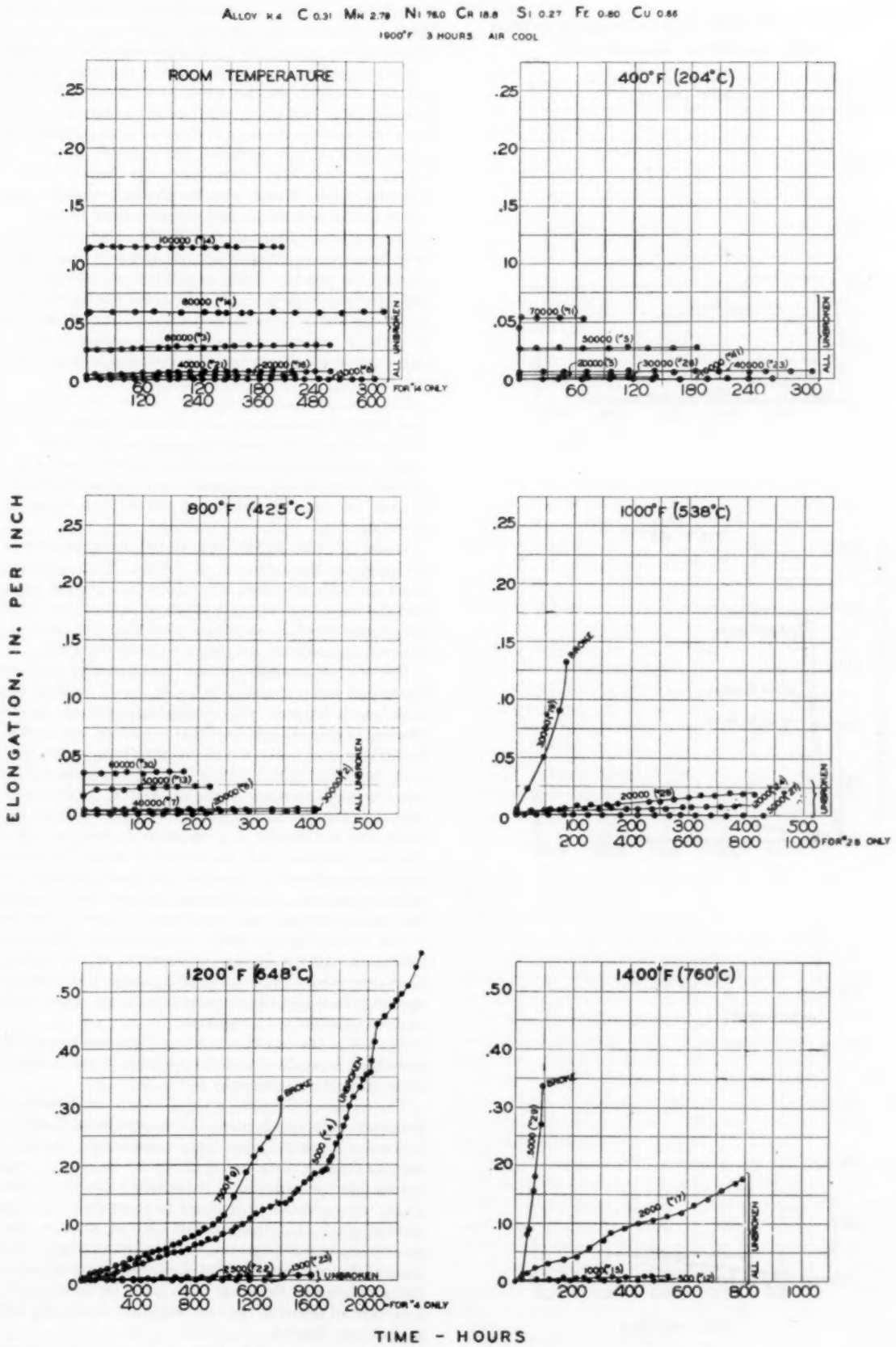


FIG. 5 TIME-ELONGATION CURVES FOR THE ALLOY K-4

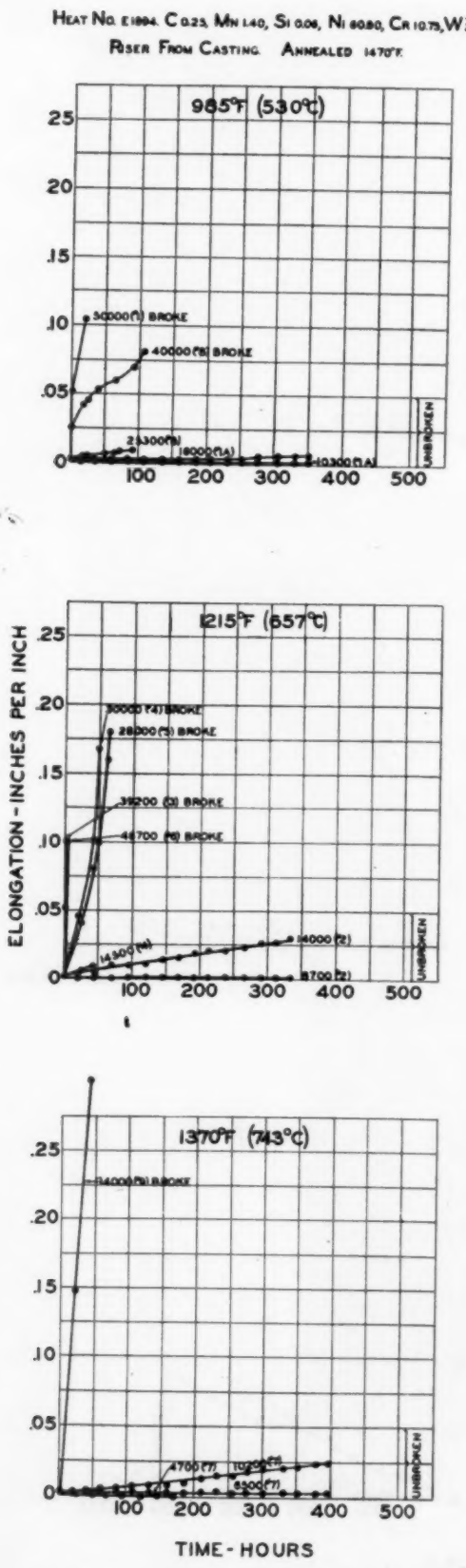


FIG. 6 TIME-ELONGATION CURVES FOR THE NI-CR-FE ALLOY E1894

a casting and did not show the best properties obtainable in this alloy at either room or elevated temperatures (Table 1). The similar alloys marked FS-1 and FS-2 were made abroad.

Alloy E1912 is an austenitic steel of the type of Krupp's V2A and the English corrosion-resistant steel called Stabrite. Since austenitic steels are often said to retain a high proportion of their room-temperature strength at elevated temperatures, this nickel-chromium steel was included for comparison with the high-chromium "stainless" steels previously tested. Alloy EE1114 is similar to alloy E1912, with the exception that it has somewhat higher carbon and silicon and 4.5 per cent of tungsten.

The materials designated as E1894, FS-1, FS-2, and E1567 have been used extensively abroad, and to a lesser degree in this country, the first two chiefly in the form of castings and the latter pair largely as forgings. Alloy EE1114, which has been used both in castings and in forgings, is claimed to be resistant to softening at elevated temperatures and to resist scaling, even in oxidizing and sulphurous atmospheres at temperatures as high as 2000 deg. fahr.

The chromium-vanadium steel, No. 10-518, Table 1, has been used widely for autoclaves, and, especially with a somewhat lower carbon content, for vessels for ammonia synthesis. The treatments to which this steel was subjected before test were intended to produce conditions comparable to those existing in the walls of forgings too large to quench and too thick to harden effectively in the air.

Steels EE1139, E1549, and E1490 are compositions used in cylinders for the extrusion of metals. Such cylinders, or the liners for such cylinders, are subjected to severe service. The material must flow only very slowly under fairly high pressures at temperatures as high as 1200 or 1300 deg. fahr., and at the same time exhibit sufficient toughness to resist cracking.

The two carbon steels and the three low-alloy steels comprising the second group of metals in Table 1 were made in 75-ton basic open-hearth furnaces and cast into large-end-up, hot-top ingots. The ingots were charged hot into the soaking pits, and after being brought to the proper rolling temperatures, rolled to the bloom sizes shown in Table 1. After heat treatment, as indicated, flow-test specimens were cut from the blooms, the long axis being at 90 deg. to the direction of rolling, at points half-way between the center and surface but at an appreciable distance from the ends. The heat treatment and the method of taking the flow-test specimens were selected to represent the metal in large forgings for high-temperature, high-pressure power-plant equipment and the tests at 700 deg. fahr. were made to assist in the selection of a steel to meet special design requirements. The specified heat treatments did not reduce the strengths of the steels at atmospheric temperatures to the desired values in all cases, and it was necessary to repeat the treatments for some of the steels in order to obtain the desired properties.

In view of the fact that some of these steels were heat treated more than once, the structural condition of the material as tested is shown in the micrographs of Fig. 4.

The metals of the third group, Table 1, were melted in a high-frequency induction furnace, the charges being based on a carbon content of about 0.25 per cent, a manganese content of 0.5 per cent, and a silicon content of about 0.5 per cent. As far as was practicable the nickel and chromium contents of the different alloys were selected to conform to those found in typical commercial alloys, but the primary basis of selection was a widespread scattering over the nickel-chromium-iron ternary diagram so that the general trend of effects of composition on the flow characteristics at 1000 deg. fahr. could be determined. The alloys were all tested in the cast condition after being heat treated as shown in Table 1.

The results of tensile tests at atmospheric temperatures or

Brinell hardness tests are given in Table 1 for many of the materials in order to define better the conditions in which the different metals of the several groups were tested. Since the properties of the metals in groups I and II are compared with those of five steels previously reported upon, the chemical compositions and preliminary treatments of the latter have been included for convenience in Table 1.

IV—RESULTS AND DISCUSSION

1—General Relations Between Stress, Temperature, Deformation, and Time

The flow of the metals of Groups I and II, Table 1, when loaded at different temperatures, is shown graphically in Figs. 5 to 17, inclusive. The relations between stress, temperature, deformation, and time are similar to those observed in the steels previously tested, and the time-elongation curves fall into three classes which depend upon the metal and the temperature.

At atmospheric and slightly elevated temperatures, initial changes in dimensions (initial flow) are not necessarily followed by continuous flow, but, due to strain hardening, the rates of flow may decrease with time. In many cases the flow rates subsequent to the initial period (second stage of flow) may be considered to be zero for all practical purposes, and determination of allowable working stresses is therefore a function of the permissible initial deformation.

At temperatures above those at which strain hardening is observed, any appreciable initial flow is followed by further and continuous deformation which may take place at approximately the same rate, a higher or a lower rate, depending upon the conditions of test. At such high temperatures the flow ordinarily increases with time and the allowable working stresses usually become a function of the rate of deformation subsequent to any initial changes in dimensions (secondary flow rates).

At intermediate temperatures, the time-elongation curves may show some of the characteristics of each of the two classes of curves mentioned. In other words, the transition from the relations typical of temperatures at which marked strain hardening is observed to the relations typical of much higher temperatures is most often gradual. The time-elongation curves may show appreciable initial deformation followed by decreases in flow rates, but the flow may be continuous so that fracture occurs in relatively short time.

In ductile materials appreciable elongation eventually results in appreciable contraction in area. This in turn results in stress concentration, and the rates of flow are greatly increased just before fracture (final flow).

Different methods have been used in summarizing the results of flow tests, but probably the most acceptable to engineers are those which show the relations between deformation, stress, temperature, and time, since the selection of allowable working stresses is usually not solely a function of the time for fracture but demands consideration of the allowable deformation during any selected life. The desired life and the maximum allowable deformation will vary with different classes of structures, and the viewpoints of individual designers and are likewise affected by the details of construction and operation. For this reason, the results are summarized in terms of per cent deformation per thousand hours at the different temperatures as is shown in Figs. 30 to 41, inclusive, and in Table 2.

The stresses producing the deformation rates shown in Figs. 30 to 41 were determined from charts giving the relations between the applied load and the initial flow or secondary flow rates, shown in Figs. 18 to 29, inclusive. The methods by which Figs. 30 to 41 were obtained from data in Figs. 18 to 29 and Table 2, and the assumptions involved, have been discussed in detail in reports already referred to.

TABLE 2 FLOW-TEST DATA FOR FIVE STEELS TESTED AT 700 DEG. FAHR.

Steel no.	Type	Composition[1]	Load, lb. per sq. in.	Initial flow, in. per in.	Secondary flow, in. per in. per hr.	Stress producing flow per 1000 hr. of 0.1%	Stress producing flow per 1000 hr. of 1%
4H451	0.23 C		10,000	0.0012	0	9,500	19,500
			15,000	0.0054	0		
			25,000	0.017	0		
			35,000	0.03	0.0002		
2H384	0.45 C		15,000	0	0	19,000	(38,000)[2]
			20,000	0.0012	0		
			35,000	0.0046	0		
			50,000	continuous	..		
4H247	0.37 C	3.46 Ni	10,000	0	0	17,300	31,400
			15,000	0.0006	0		
			20,000	0.0016	0		
			35,000	0.016	0		
			50,000	0.021	0.00011		
1H391	0.36 C 0.62 Cr	1.25 Ni	10,000	0	0	26,000	32,000
			15,000	0.0002	0		
			25,000	0.0006	0		
			40,000	0.011	0.00002		
			50,000	0.026	0.00008 or less		
1H396	0.35 C 0.99 Cr	1.66 Ni	15,000	0	0	20,500	38,000
			20,000	0.0009	0		
			35,000	0.0051	0		
			50,000	0.022	0.000015		
			60,000	0.043	0.00017		

[1] For details of chemical composition and heat treatment refer to Table 1.
[2] Bracketed values estimated.

Where comparisons are made with the metals of group IV, Table 1, the required flow data for these steels were taken from previously reported tests which were carried out under comparable conditions.[10]

One important point, which has not been given much consideration in the interpretation of flow tests, is that the numerical values obtained on one bar, lot, or melt of a given alloy may not coincide closely with the values obtained on other bars, lots, or melts. Similar variations in the properties of metals at atmospheric temperatures are well recognized and should likewise be given consideration in the application of flow-test data. The extent of these variations is not now accurately known but it has been shown[11] that the differences in load-carrying ability, from heat to heat of steel, may be large. Without additional data, it is not safe to assume that the reported values represent the average, maximum, or minimum of the range characteristic of a given metal, and safety factors must still be employed.

2—The Commercial Nickel-Chromium-Iron Alloys

In Fig. 42, comparisons are given at 700, 1000, 1125, and 1350 deg. fahr. of the metals in Groups I and II, Table 1, and the five steels previously tested, Group IV. The stresses producing 0.1 per cent and 1 per cent initial elongation and also 0.1 and 1 per cent secondary elongation (calculated per 1000 hours) are represented by the lower and upper limits, respectively, of the blocks in the chart and serve as a basis of comparison of the flow characteristics of the different metals. Comparisons can be made at lower temperatures in some cases, and likewise at higher rates of flow, from data given in Figs. 30 to 41, inclusive.

In the relation of stress to total deformation for alloys at certain temperatures, it is necessary to consider both the initial and the secondary flow. These differences can be studied in detail from data given in Figs. 18 to 29 and Table 2.

The data given in Fig. 42 confirm some of the facts developed from practical experience, but in addition show certain features which have not been given much, if any, attention. At temperatures between 1125 and 1350 deg. fahr. the high-nickel-chromium steels were generally superior to the carbon and low-alloy steels, but if the deformation per 1000 hours at the higher temperatures is limited to 1 per cent or less, the maximum allowable stresses drop to low values even for the high-alloy steels.

[10] See footnotes 5, 6, and 7.
[11] See footnote 7.

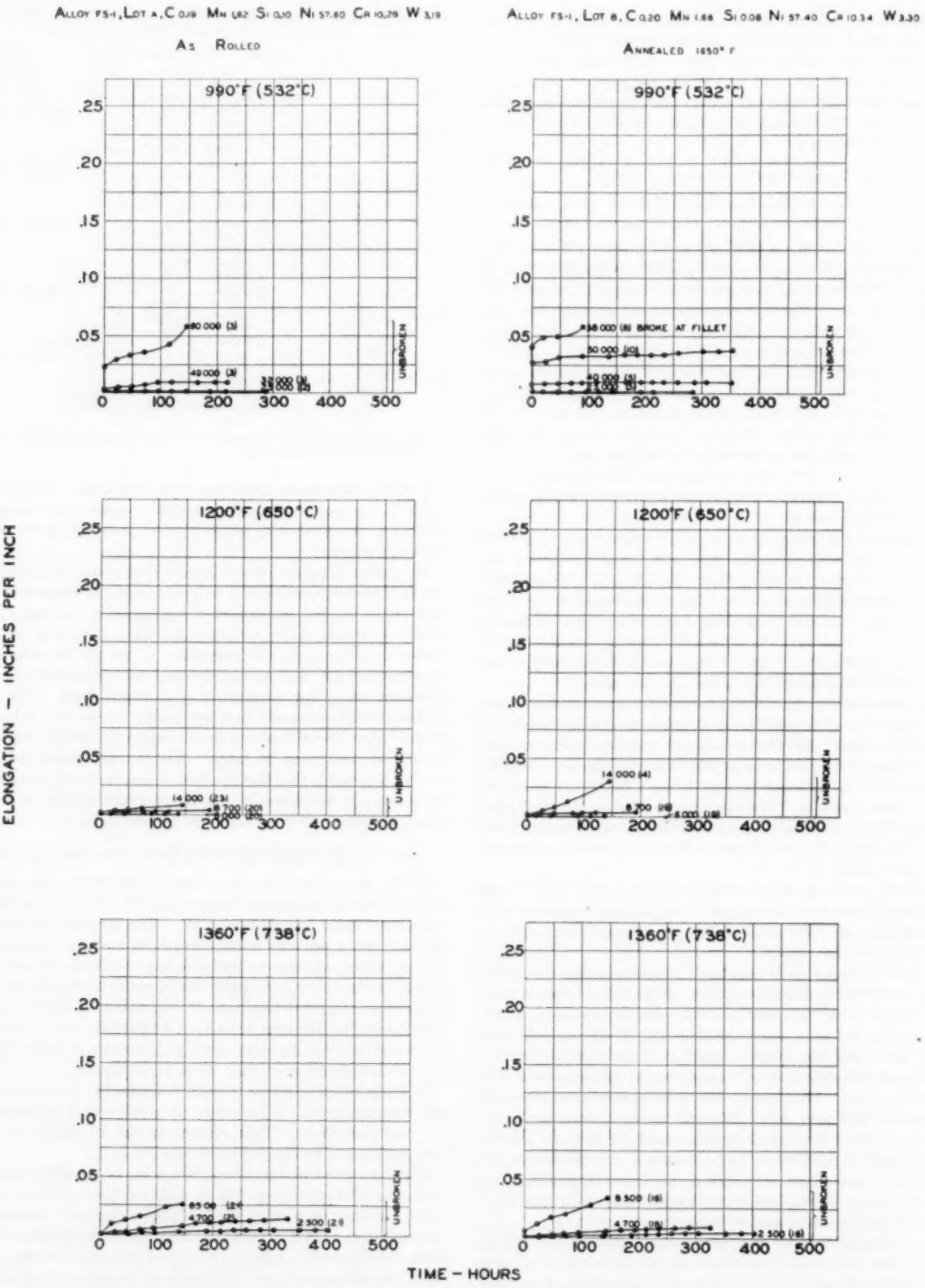


FIG. 7 TIME-ELONGATION CURVES FOR THE NI-CR-FE-W ALLOY FS-1, LOTS A AND B

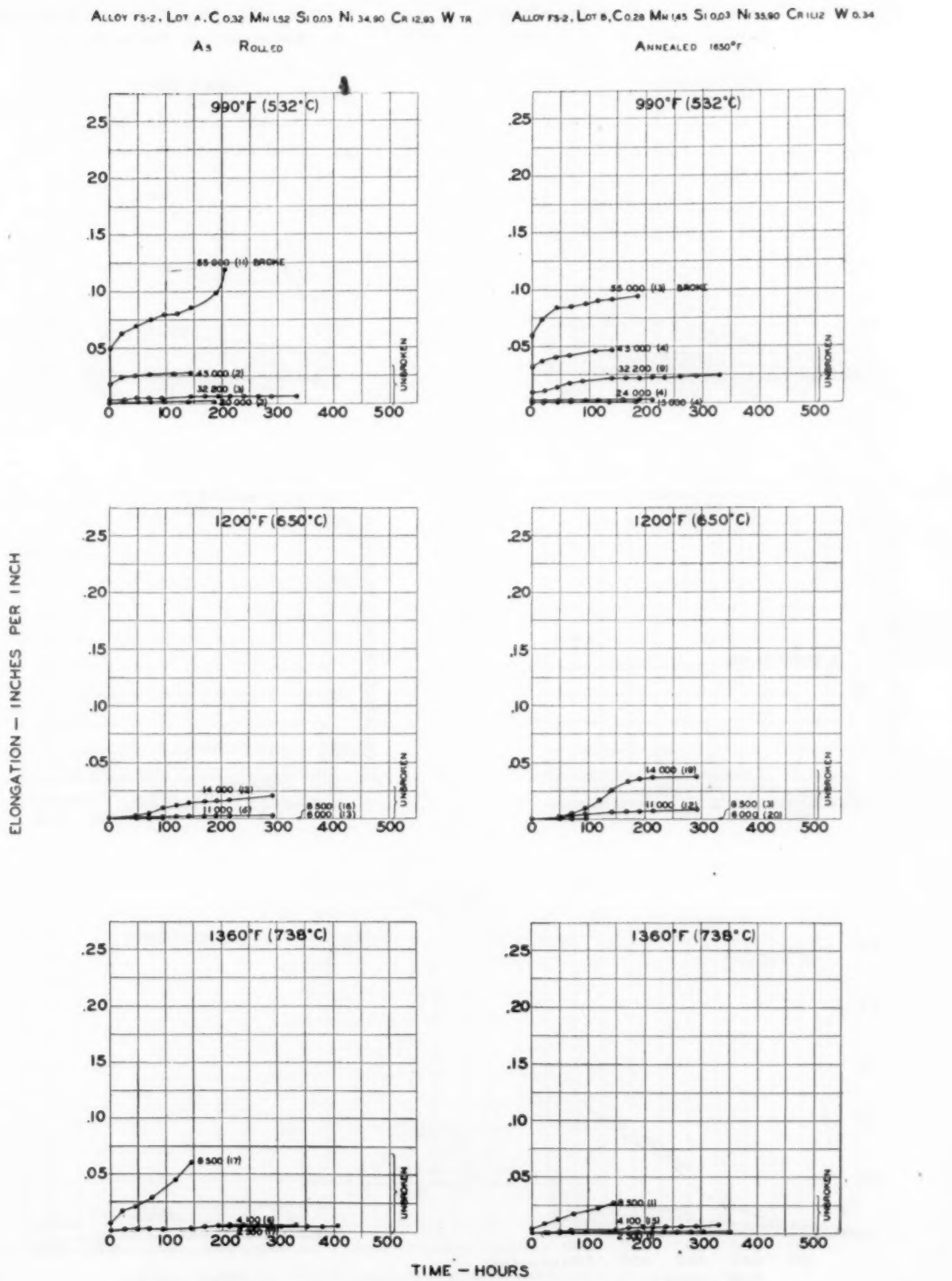


FIG. 8 TIME-ELONGATION CURVES FOR THE NI-CR-FE ALLOY FS-2, LOTS A AND B

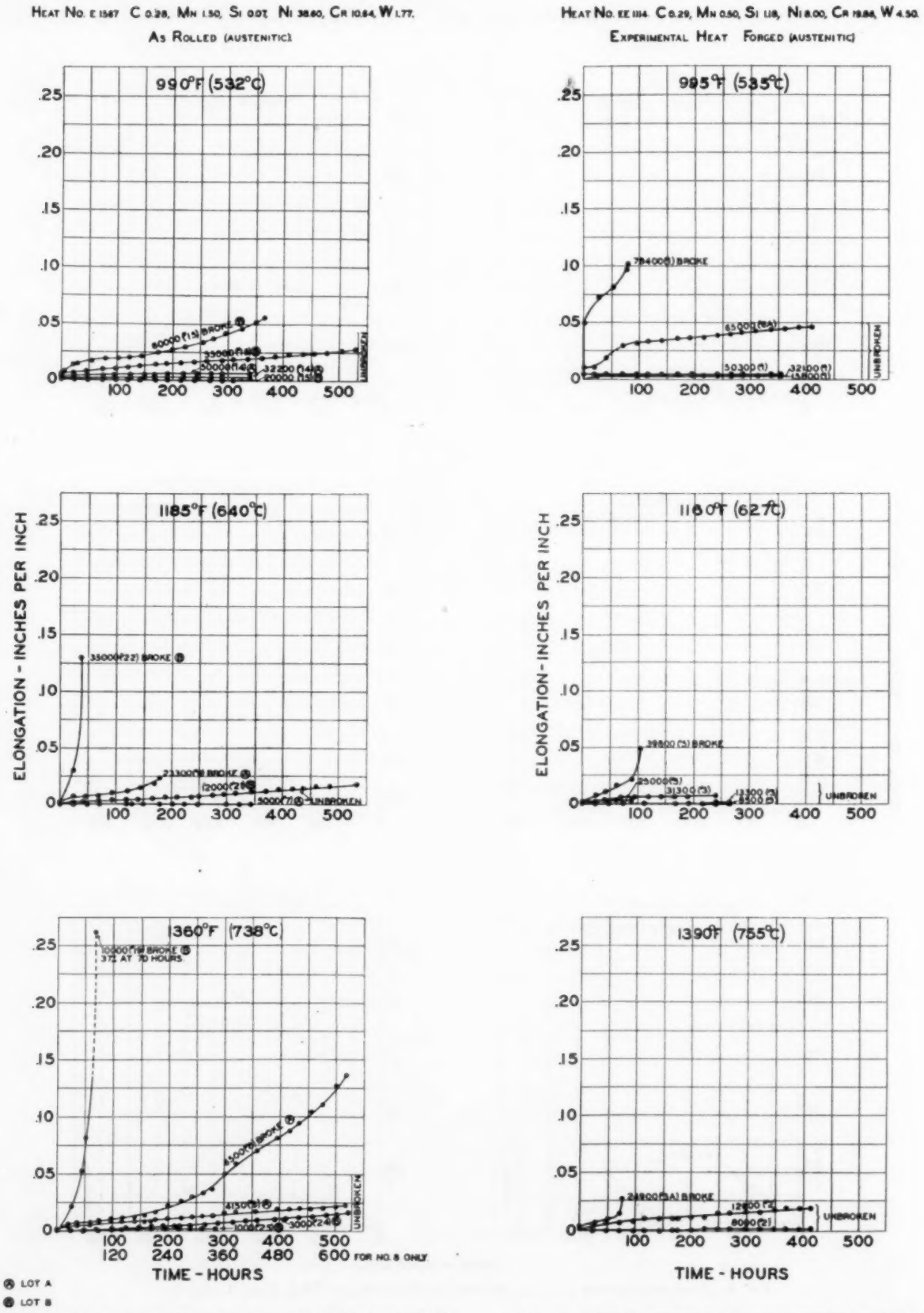


FIG. 9 TIME-ELONGATION CURVES FOR THE NI-CR-FE ALLOY E1567

FIG. 10 TIME-ELONGATION CURVES OF THE STEEL EE1114

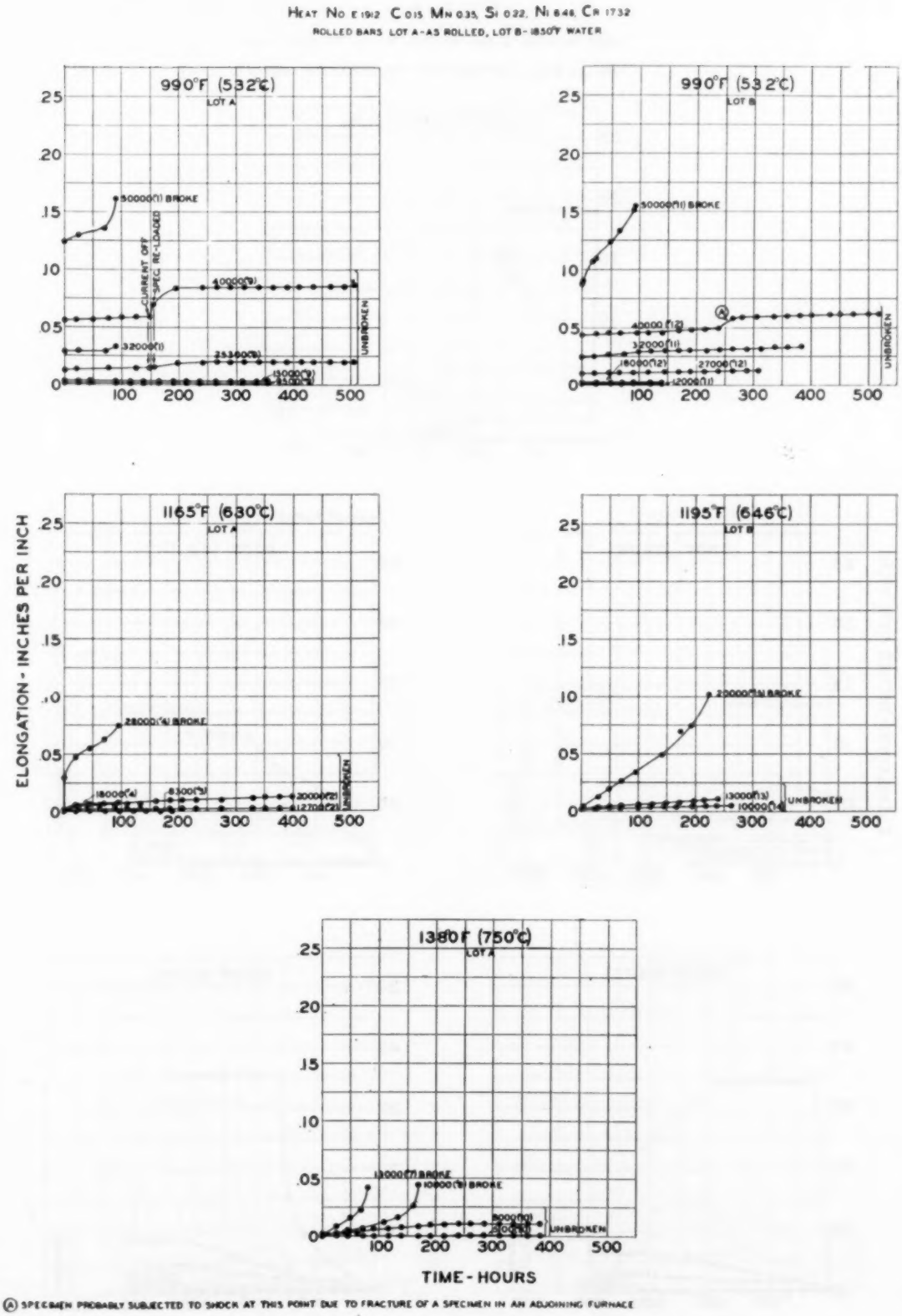


FIG. 11 TIME-ELONGATION CURVES OF THE STEEL E1912

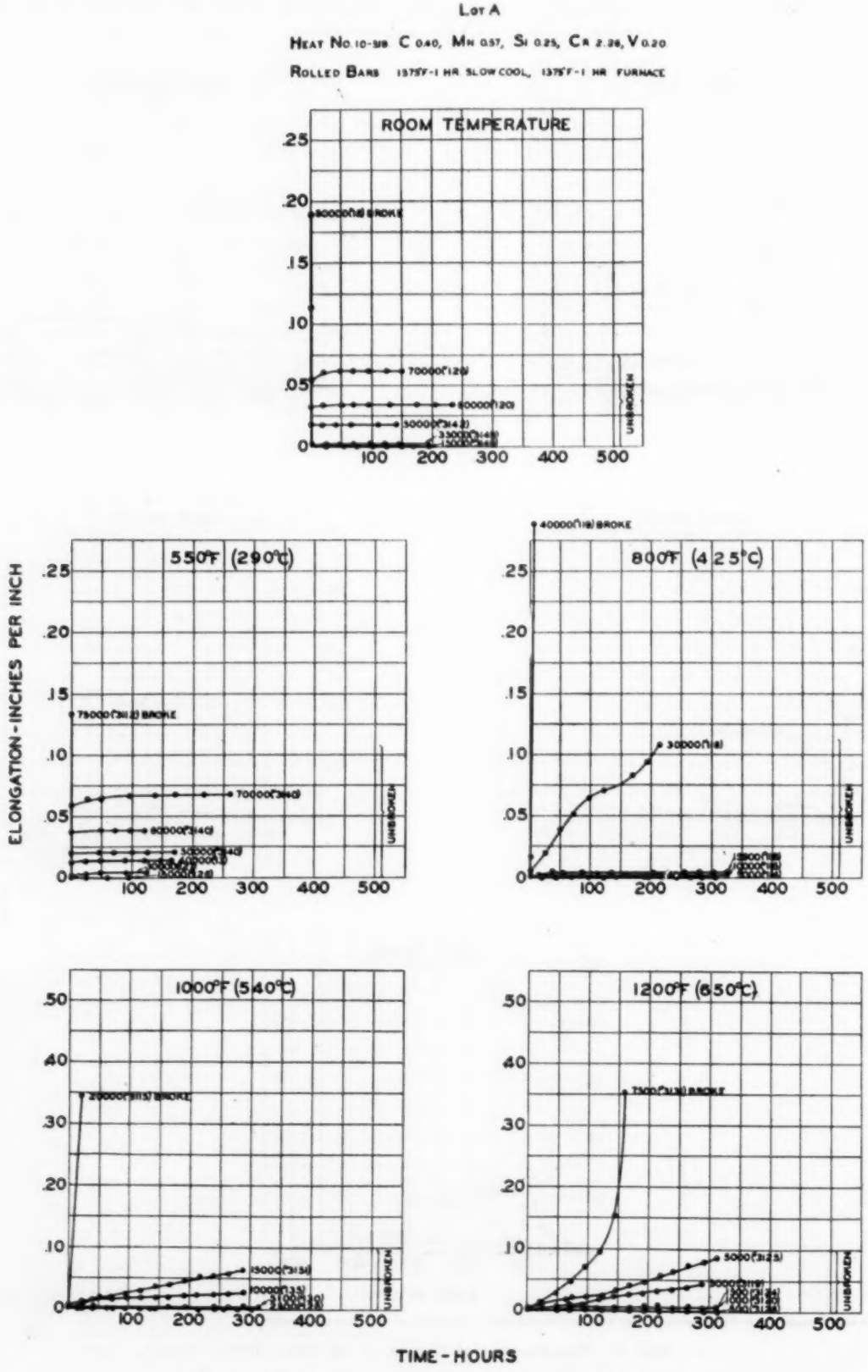


FIG. 12 TIME-ELONGATION CURVES OF STEEL 10-518, LOT A

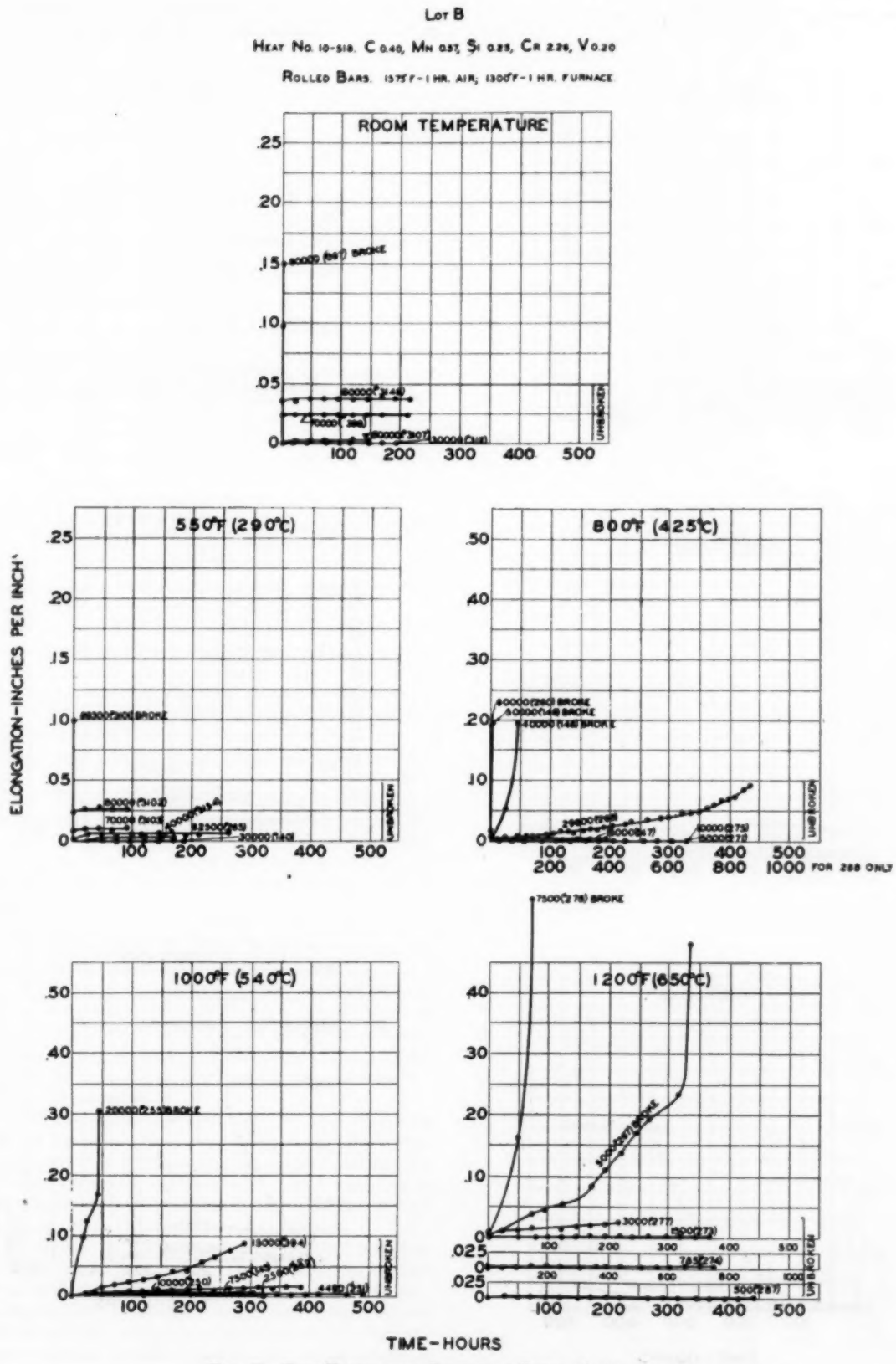


FIG. 13 TIME-ELONGATION CURVES OF STEEL 10-518, LOT B

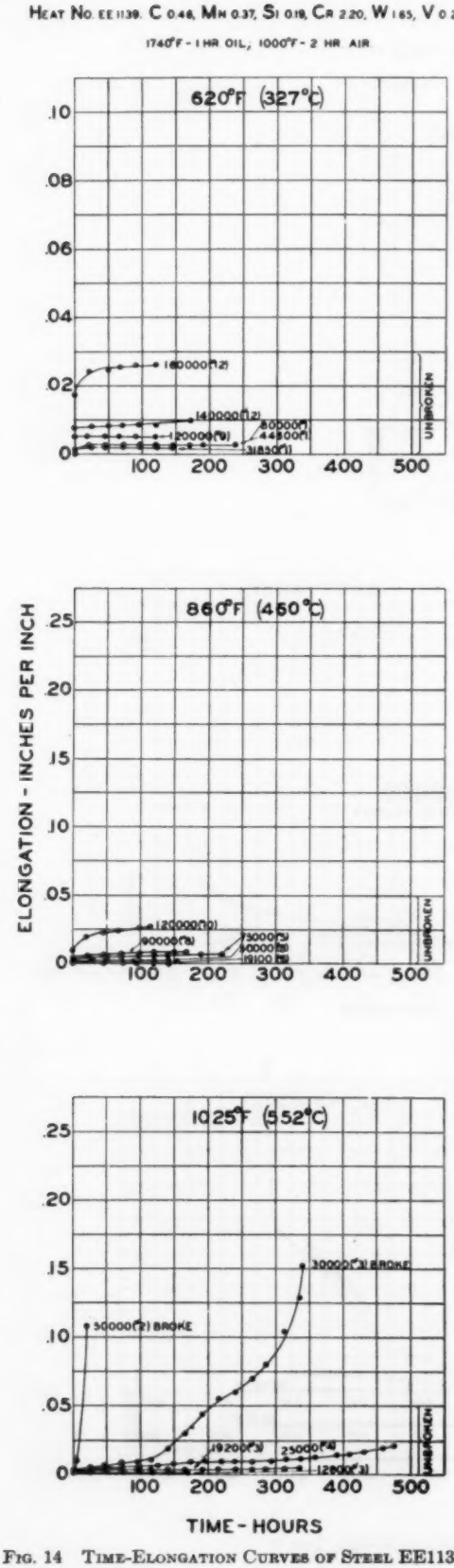


Fig. 14 Time-Elongation Curves of Steel EE1139

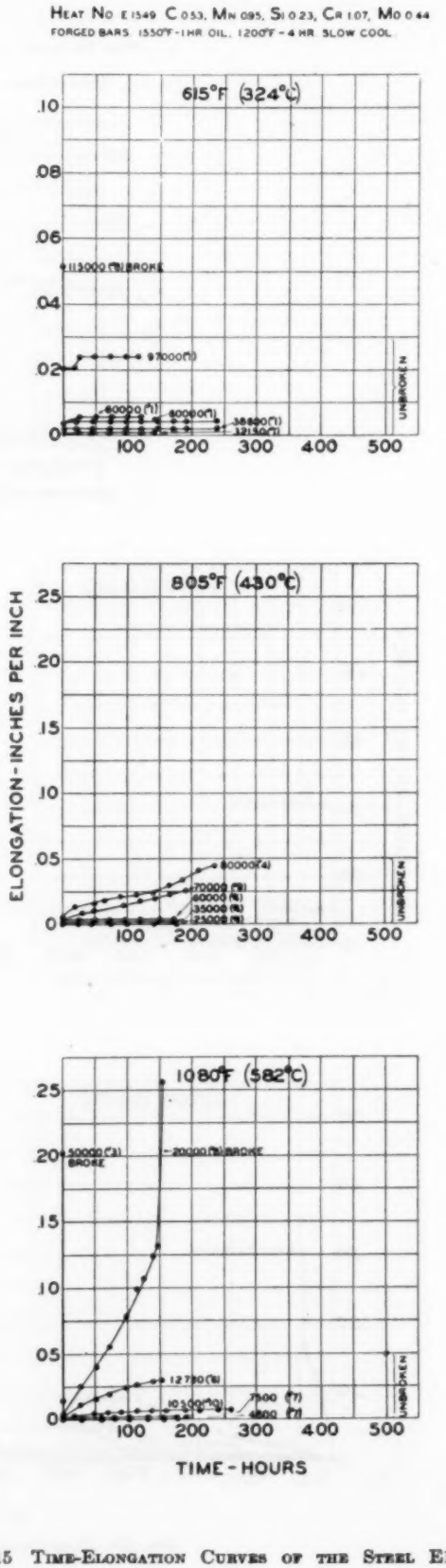


Fig. 15 Time-Elongation Curves of the Steel E1549

The outstanding alloy of the group at temperatures between 1000 and 1350 deg. fahr. was the one containing about 0.3 per cent carbon, 1 per cent silicon, 20 per cent chromium, 8 per cent nickel, and 4.5 per cent tungsten; it showed as high or higher load-carrying ability than any of the other alloys. Its superiority was most marked at 1350 deg. fahr., but tended to decrease with decrease in the permissible rates of flow and with decrease in temperature.

With this one exception, alloys showing superior properties at 1350 deg. fahr. did not necessarily show superiority at lower temperatures. At 1350 deg. fahr. superior load-carrying ability was shown by the alloys, 20 Cr–8 Ni–1 Si–4.5 W; 18 Cr–8 Ni; and 61 Ni–11 Cr–2.8 W; at 1000 deg. superiority was shown by the alloys 20 Cr–8 Ni–1 Si–4.5 W; 38 Ni–11 Cr–1.8 W, and 57 Ni–10 Cr–3 W (FS-1 lot A; as rolled).

These different alloys all represent compositions used industrially for service at different temperatures and it was not practicable to attempt to determine the effects of variations in important elements such as nickel, chromium, and silicon upon their high-temperature-flow characteristics, since in only a few cases were the differences in composition of any two alloys restricted to a single element. For example, the effect of tungsten on the alloy nominally containing 18 per cent chromium and 8 per cent nickel cannot be established from data in Fig. 42 since the alloy with tungsten contains about twice the carbon and five times the silicon in the alloy without tungsten. Such circumstances do not affect the usefulness of the flow-test data, but made it necessary to depend upon the experimental melts, group III, Table 1, to show the general trends in the system, nickel-chromium-iron.

The two lots of alloy E1912, containing about 18 per cent chromium and 8 per cent nickel with 0.15 per cent carbon, offer some interesting comparisons. It was originally intended to test this alloy only in the hot-worked condition as represented by lot A, Table 1. However, more samples were required before the tests were completed, and the bars of the second lot (B) were found to have a higher tensile strength at atmospheric temperatures than bars of lot A, due quite probably to a somewhat lower finishing temperature in rolling. Therefore lot B was softened by water quenching from 1850 deg. fahr. and subsequently tested, although its room-temperature tensile strength was still somewhat higher than that of lot A. Also, the grain size of the quenched samples was smaller than that of the hot-finished samples, as is shown in Figs. 43 and 44.

At 990 deg. fahr. these two lots of alloy E1912 exhibited about the same load-carrying ability, but at 1180 deg. the fine-grained (water-quenched) steel was weaker than the coarse-grained material (Fig. 36). Comparisons of the two lots were not made at higher temperatures, but the evidence available gives interesting confirmation of the view that fine-grained metals, having more intercrystalline material than corresponding coarse-grained metals, should be stronger at atmospheric temperatures but weaker at high temperatures.

There is some evidence, however, that coarsely crystalline metals may be dangerous from other viewpoints. While they flow at lower rates than corresponding fine-grained metals, fracture *may* occur with less total elongation. There is also the possible added disadvantage of weakness in resisting shock at high temperatures as well as at atmospheric temperatures.

In the case of the steel nominally containing 18 per cent chromium and 8 per cent nickel, a marked intercrystalline weakness was observed at temperatures between 1165 and 1380 deg. fahr., and to some extent also at 990 deg. fahr., in both fine-grained and coarse-grained samples, though the latter seemed to show the more marked tendency in this direction. When the specimens were allowed to remain under load at 1165–1195 deg. or 1380 deg. fahr. until fracture occurred, the fracture was intercrystalline and

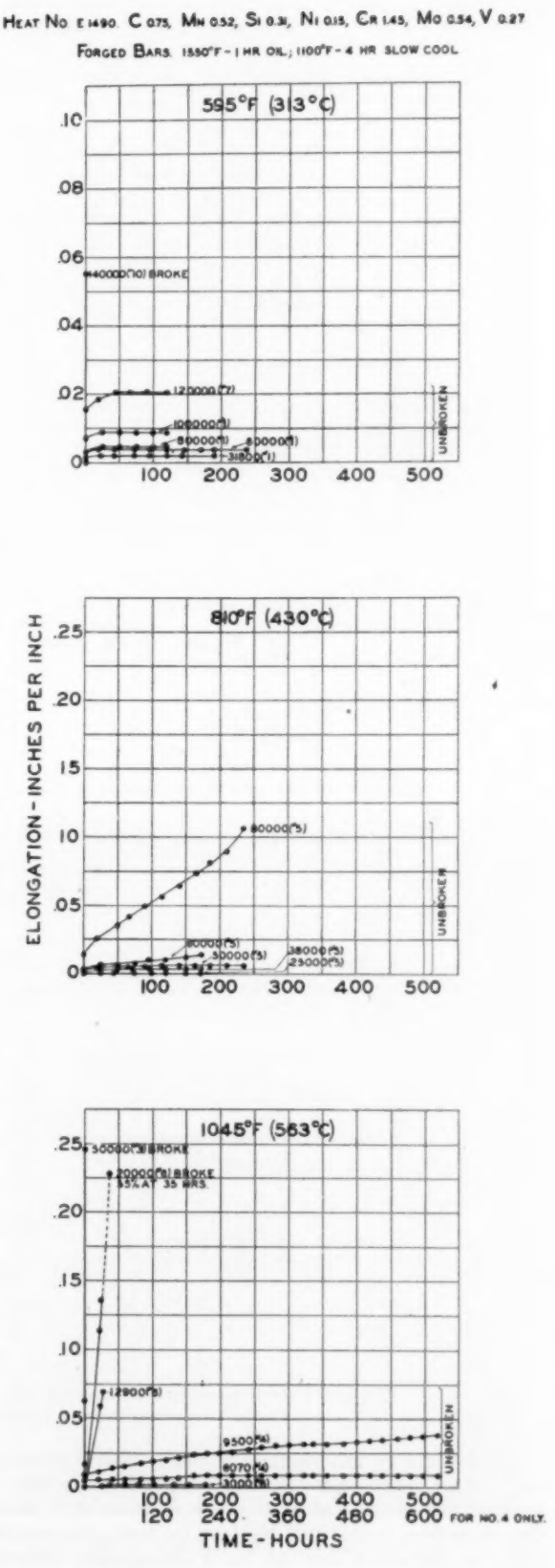


FIG. 13 TIME-ELONGATION CURVES OF THE STEEL E1490

grain separations were to be observed at some distance back of the fractures, especially at the surface. These features are shown in Figs. 43 and 44. There is no clear evidence that this intercrystalline weakness is associated with carbide precipitation at the grain boundaries.

above 990 deg., embrittlement of steels with over 0.07 per cent carbon has been reported to occur and has been attributed to the reprecipitation of carbides at the grain boundaries.

The steel containing about 0.3 per cent carbon, 1 per cent silicon, and 4.5 per cent tungsten, in addition to 20 per cent chro-

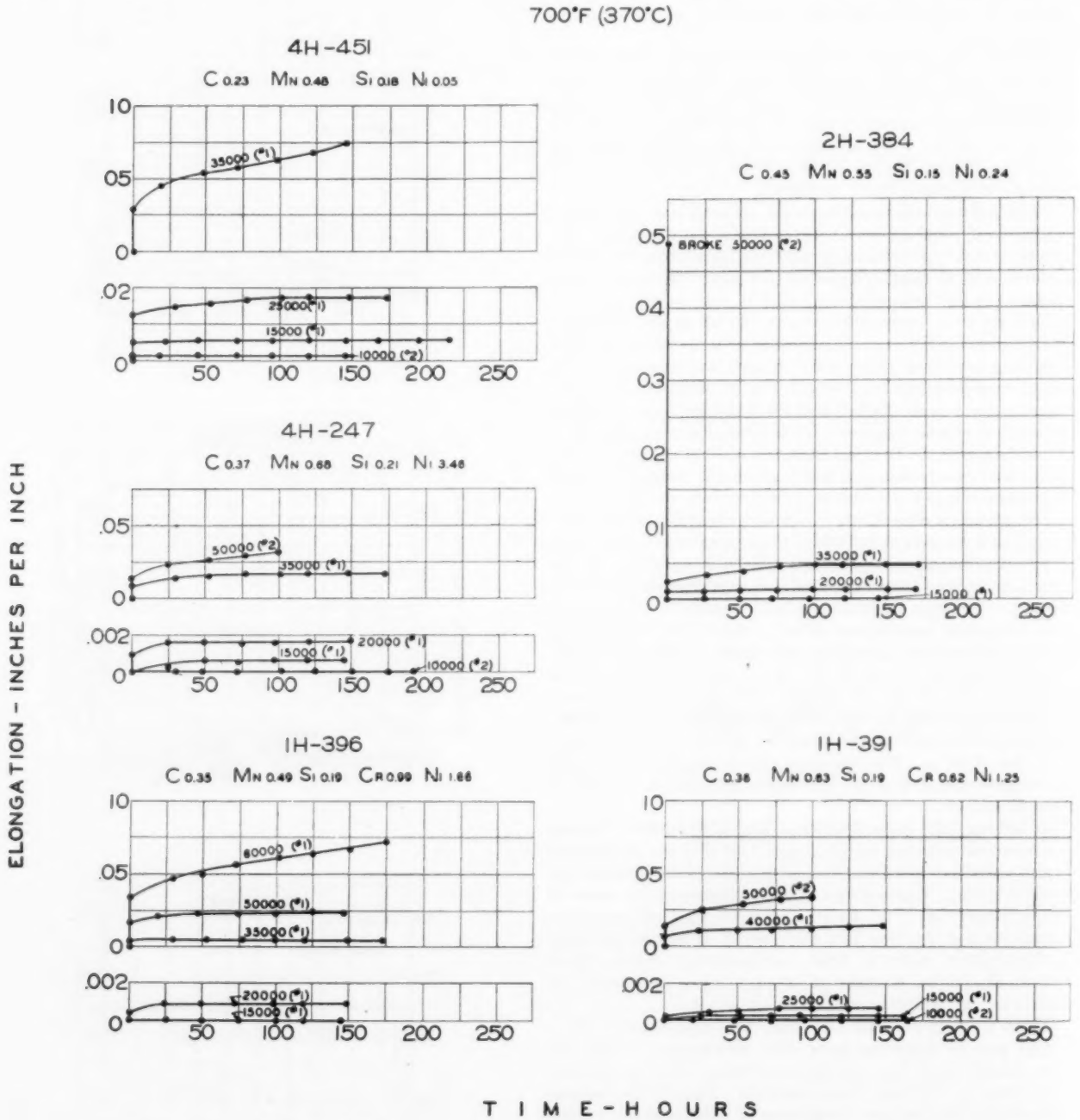


FIG. 17 TIME-ELONGATION CURVES AT 700 DEG. FAHR. OF TWO CARBON AND THREE ALLOY STEELS
(Table 1 contains details of chemical composition and preliminary heat treatments.)

According to one manufacturer,[12] there is decided difference in behavior of this nickel-chromium steel which depends upon whether the carbon content is below or above 0.07 per cent. Quenching from around 2100 deg. fahr. has been recommended to bring the carbides in solution, but if subsequently reheated to

[12] Private communication.

mium and 8 per cent nickel, showed intercrystalline weakness at 1160 and 1390 deg. fahr., but, so far as could be determined from the usual metallographic examination, this was less marked than in the 0.15 per cent carbon steel with 18 per cent chromium, 8 per cent nickel, and no tungsten. (Figs. 43, 44, and 45.)

Intercrystalline weakness, possibly associated in some cases

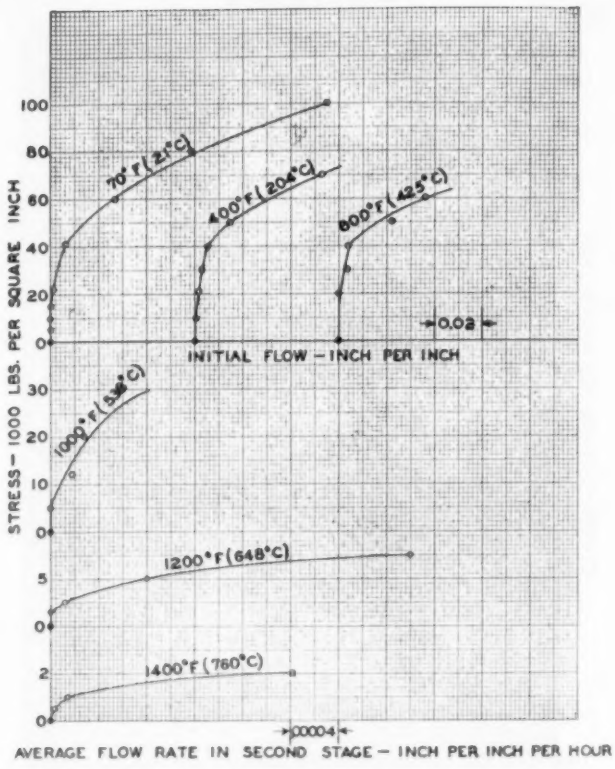


FIG. 18 FLOW DATA FOR NI-CR ALLOY K-4 AT DIFFERENT TEMPERATURES
(Obtained from Fig. 5.)

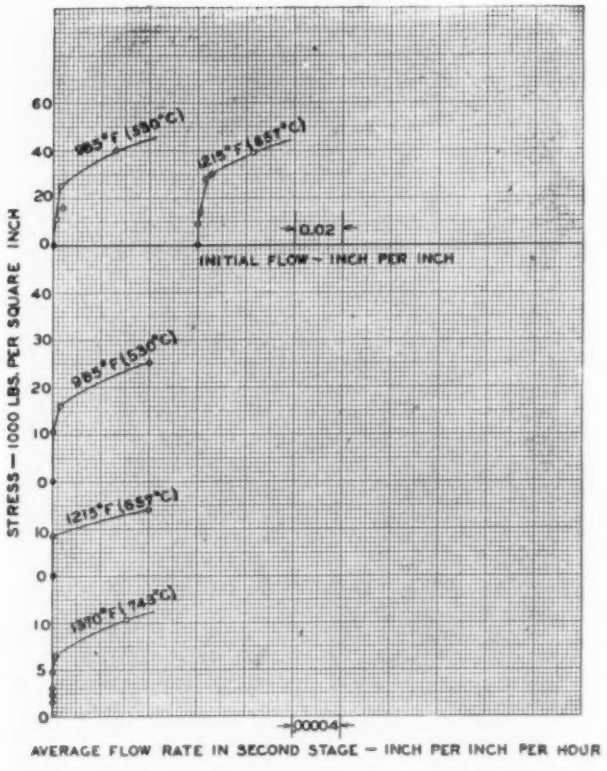


FIG. 19 FLOW DATA FOR NI-CR-FE-W ALLOY E1894 AT DIFFERENT TEMPERATURES
(Obtained from Fig. 6.)

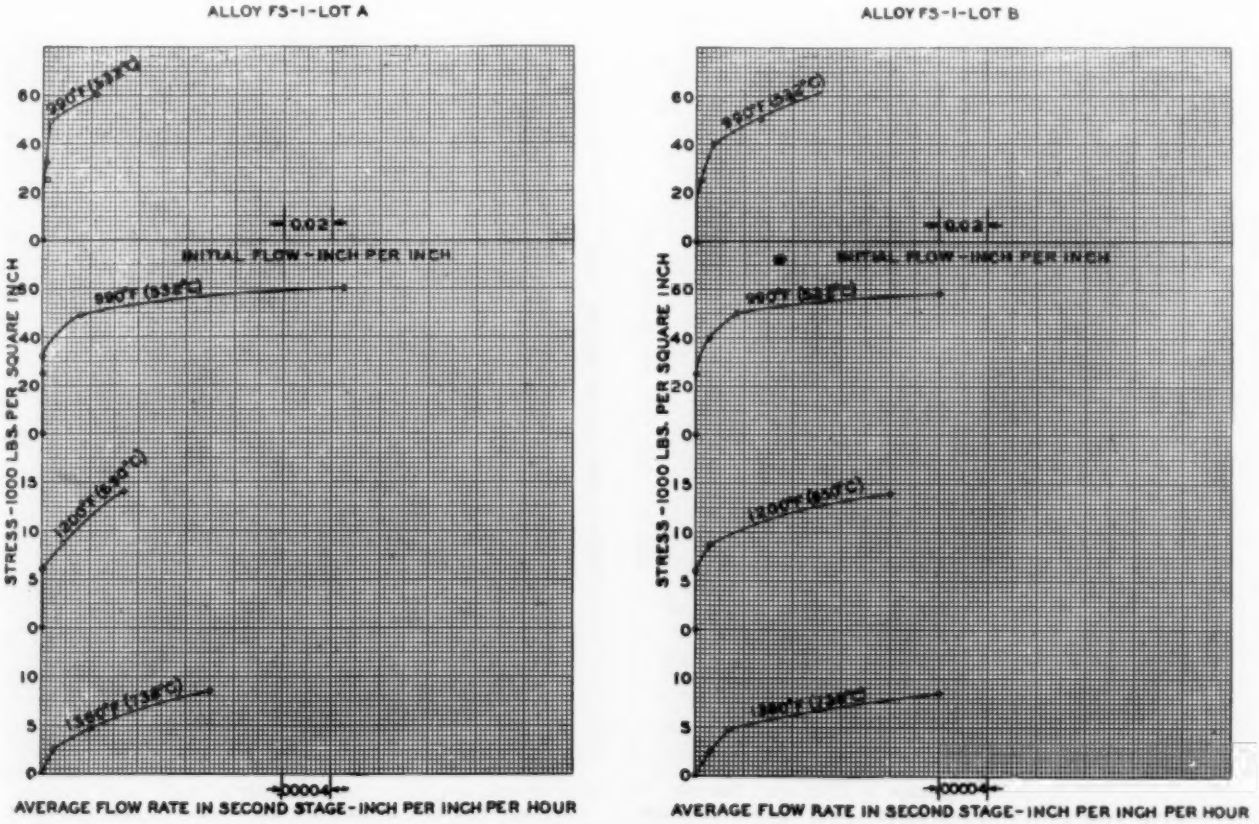


FIG. 20 FLOW DATA FOR THE HIGH-FE-CR-NI-W ALLOY FS-1, LOTS A AND B, AT DIFFERENT TEMPERATURES
(Obtained from Fig. 7.)

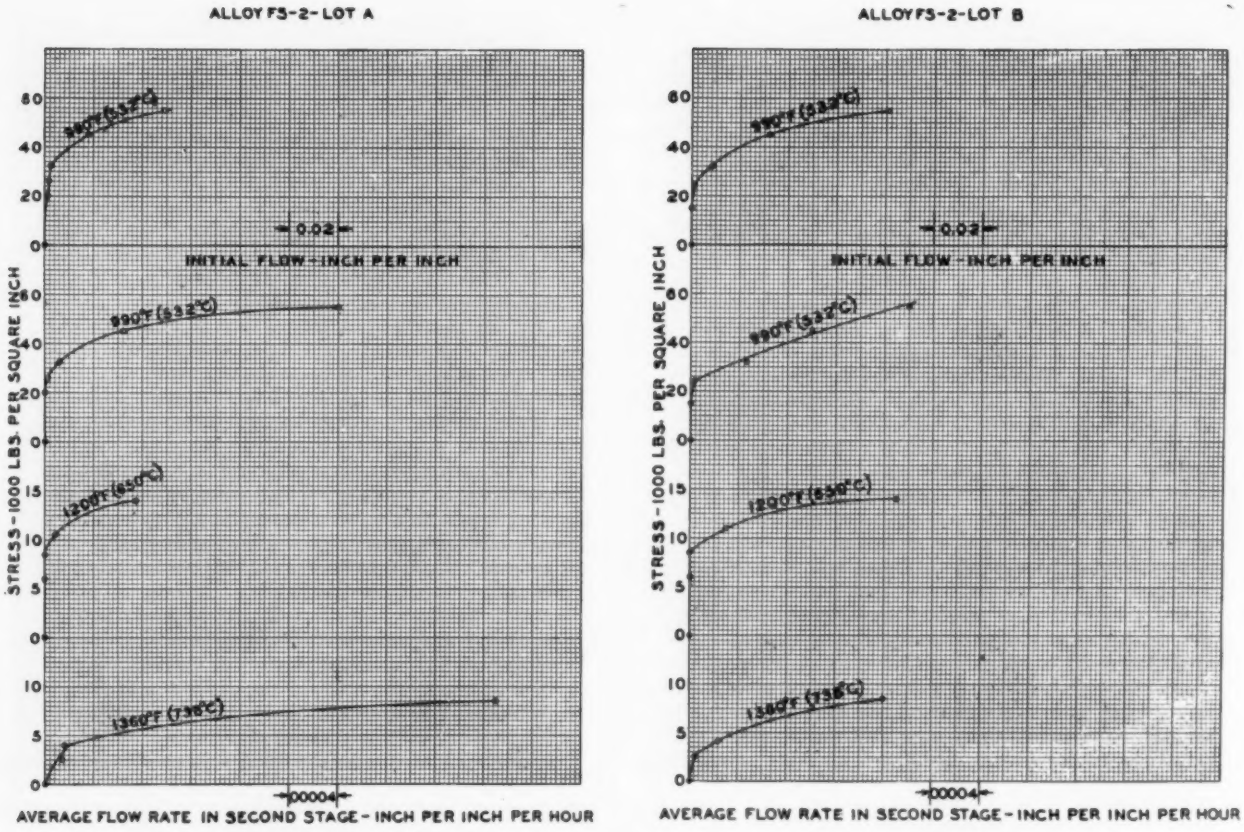


FIG. 21 FLOW DATA FOR THE HIGH-FE-CR-NI ALLOY FS-2, LOTS A AND B, AT DIFFERENT TEMPERATURES
(Obtained from Fig. 8.)

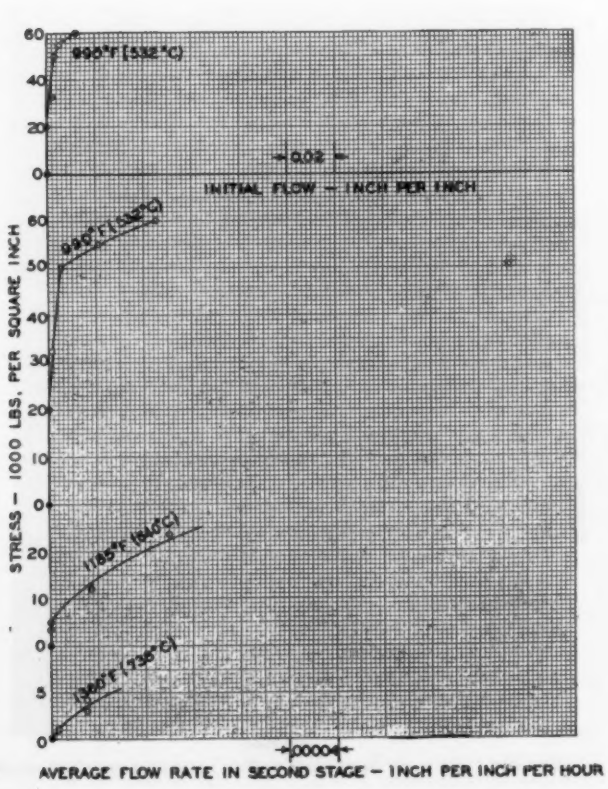


FIG. 22 FLOW DATA FOR NI-CR-FE-W ALLOY E1567 AT DIFFERENT TEMPERATURES
(Obtained from Fig. 9.)

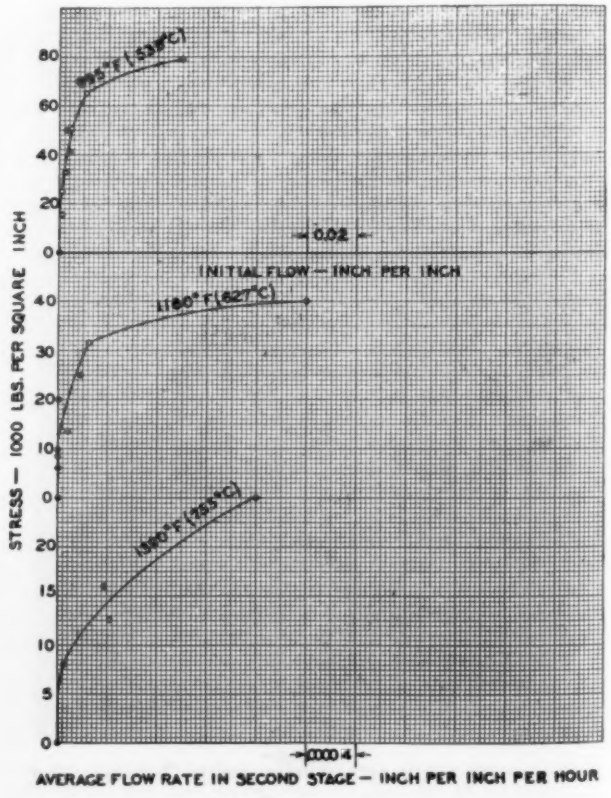


FIG. 23 FLOW DATA FOR THE HIGH-CHROMIUM-NICKEL-TUNGSTEN STEEL EE1114 AT DIFFERENT TEMPERATURES
(Obtained from Fig. 10.)

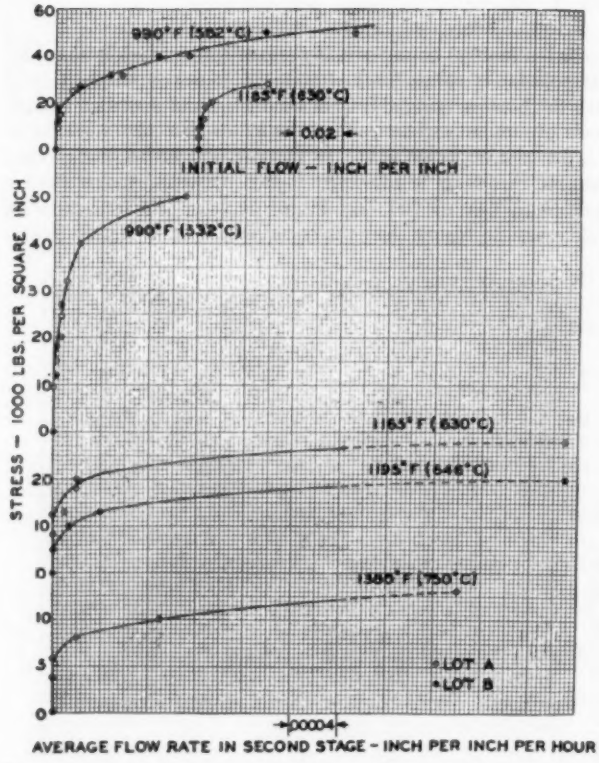


FIG. 24 FLOW DATA FOR THE HIGH-CHROMIUM-NICKEL STEEL E1912 AT DIFFERENT TEMPERATURES
(Obtained from Fig. 11.)

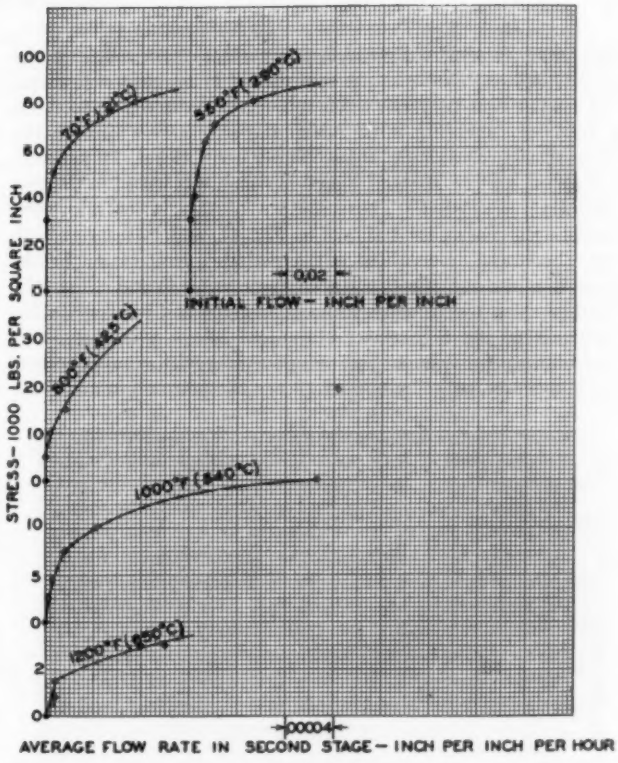


FIG. 26 FLOW DATA FOR THE CR-V STEEL 10-518, LOT B, AT DIFFERENT TEMPERATURES
(Obtained from Fig. 13.)

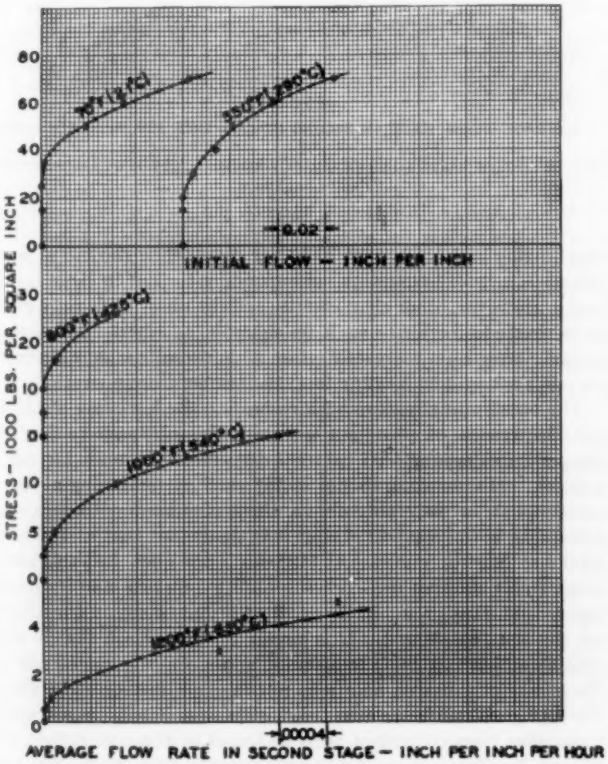


FIG. 25 FLOW DATA FOR THE CR-V STEEL 10-518, LOT A, AT DIFFERENT TEMPERATURES
(Obtained from Fig. 12.)

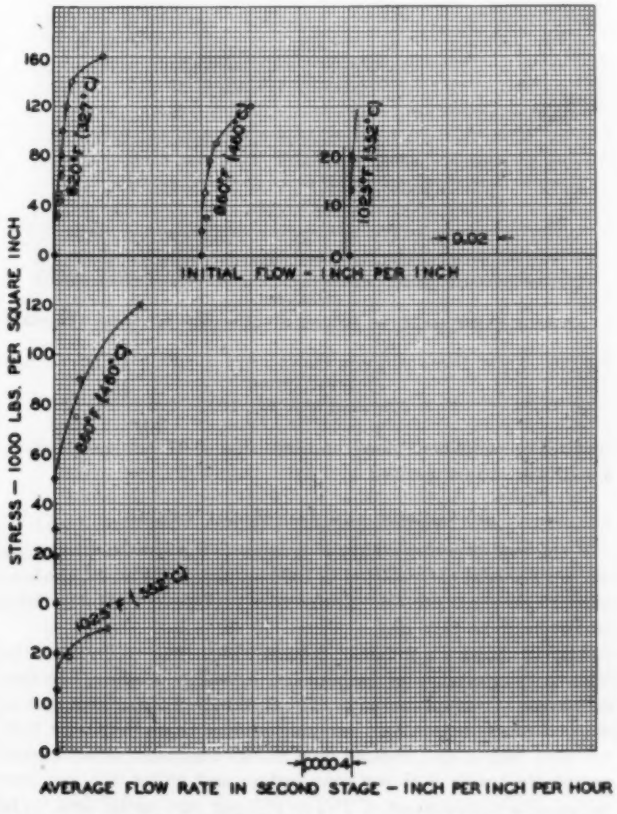


FIG. 27 FLOW DATA FOR THE CR-V-W STEEL EE1139 AT DIFFERENT TEMPERATURES
(Obtained from Fig. 14.)

with grain growth, was observed in two of the samples of the alloy E1567 containing 38 per cent nickel, 11 per cent chromium, and 1.8 per cent tungsten when subjected continuously to stress at 1185 deg. fahr. (Fig. 46). No similar effects were found in the alloy, E1894, containing about 60 per cent nickel, 11 per cent chromium, and 2.8 per cent tungsten which was tested in the cast (annealed) condition. The structure of this latter alloy was exceedingly coarse, as is shown in Fig. 47, and in tests at 1370 deg. there was evidence of either carbide precipitation or the formation of a eutectic at the grain boundaries (*b* and *c*, Fig. 47). Further attention to constitutional changes and intercrystalline weakness of the high-nickel chromium alloys when subjected continuously to stress at high temperatures with comparisons of the alloys nominally containing 18 per cent chromium and 8 per

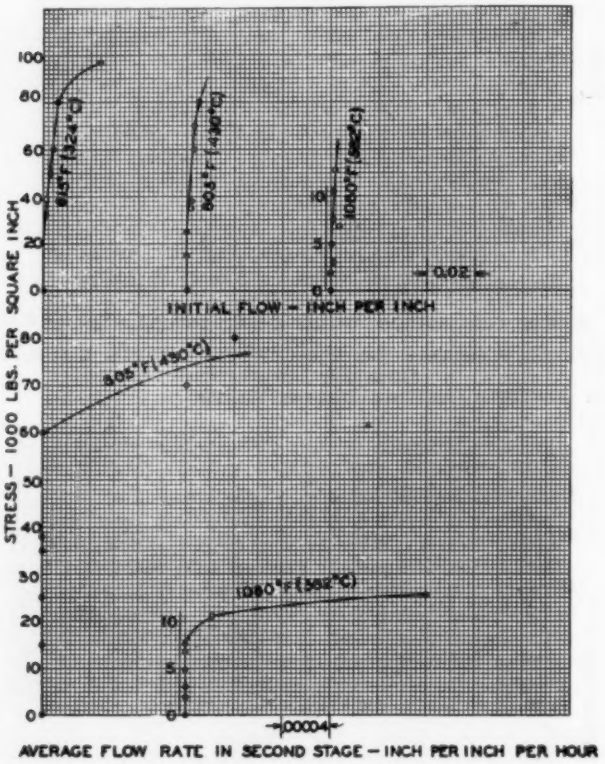


FIG. 28 FLOW DATA FOR THE CR-MO STEEL E1549 AT DIFFERENT TEMPERATURES
(Obtained from Fig. 15.)

cent nickel with both more and less than 0.07 per cent carbon, would seem to be merited.

3—CARBON AND LOW-ALLOY STEELS

If the load-carrying ability is judged by the stresses producing 0.1 to 1 per cent deformation in 1000 hours, as in Fig. 42, the low-alloy steels show promising properties at 700 to 1000 deg. fahr. and even up to 1125 deg.

Fig. 42 shows a number of factors which tend to improve the load-carrying ability of carbon and low-alloy steels. The first is increase in carbon content. The steel with 0.45 per cent carbon showed much better load-carrying ability than that with 0.23 per cent carbon and resisted flow at 700 deg. fahr. about as well as the steel with 0.35 per cent carbon and either $3^1/_2$ per cent nickel or a combination of $1^1/_4$ to $1^3/_4$ per cent nickel and $^1/_2$ to 1 per cent chromium.

A comparison of the stresses producing 0.1 to 1 per cent deformation in 1000 hours for 0.35 per cent carbon steel obtained by interpolation of values for the 0.23 and 0.45 per cent carbon steels with the load-carrying ability of the nickel and nickel-chromium steels of similar carbon content indicates that the addition of $3^1/_2$ per cent of nickel or $1^1/_4$ to $1^3/_4$ per cent nickel and $^1/_2$ to 1 per cent chromium is relatively ineffective in increasing the resistance to flow at 700 deg. fahr. At least the load-carrying ability was no higher than that of a plain carbon steel of some-

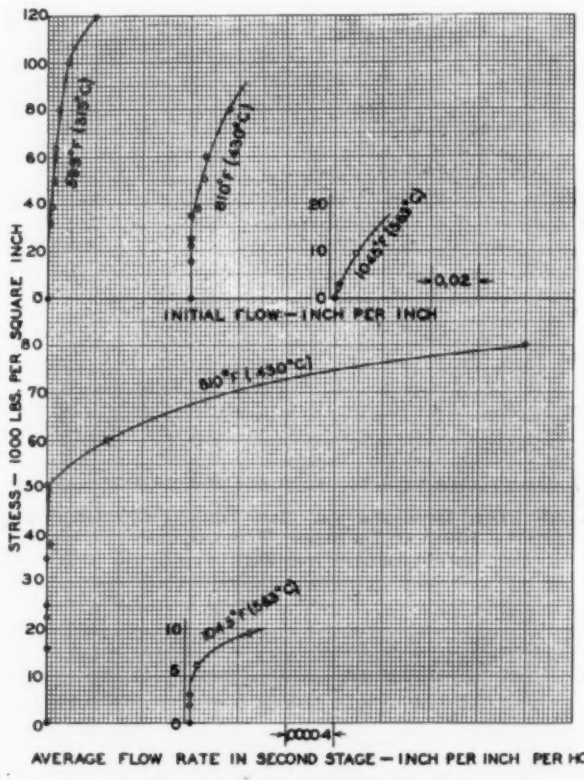


FIG. 29 FLOW DATA FOR THE CR-MO-V STEEL E1490 AT DIFFERENT TEMPERATURES
(Obtained from Fig. 16.)

what higher carbon content (0.45 per cent) when the metal was placed in a "soft" condition corresponding to that in large forgings. However, other characteristics of the alloy steels may give them marked superiority, such, for example, as superior resistance to grain growth on recrystallization which results from deformation and subsequent heating in cyclic temperature operation.

The outstanding feature with respect to the low-alloy steels is the effectiveness of quenching in raising the resistance to flow at temperatures around 700 to 1000 deg. fahr. It may be said, a priori, that quenching should be followed by tempering at temperatures at, or preferably above, those of subsequent operation if stability is to be expected, but it is significant that at 700 and 1000 deg. fahr. (in one case at 1125 deg.), the quenched and tempered steels showed load-carrying ability proportionately much higher than that produced by small additions of any of the alloying elements considered. It is largely for this reason that it was considered impracticable to attempt to develop the magnitude of the effects of tungsten additions to the chromium or other alloy steels (Fig. 42) since the heat treatments were not all comparable.

Another item of interest is to be found in the comparison of the results obtained on the chromium-vanadium steel, No. 10-518, Table I, subjected to two different preliminary heat treatments. At atmospheric and slightly elevated temperatures the

samples with the higher tensile strengths, as determined at room temperature, showed the higher load-carrying ability, but at 1200 deg. fahr., the order of superiority was reversed although the differences between the two sets of samples were not marked. It is probable that this reversal is analogous to that found in the high-chromium-nickel steel, E1912, Table 1, already discussed. Once again evidence has been obtained to indicate that high tensile strength at ordinary temperatures does not necessarily signify high load-carrying ability at high temperatures.

4—Effect of Prolonged Stress at Different Temperatures on Hardness and Impact Resistance at Atmospheric Temperatures

One question which may arise in the practical application of metals at high temperatures is whether prolonged loading will produce appreciable changes in their properties at either atmospheric or elevated temperatures. Aside from precipitation of carbides, claimed to embrittle high-chromium or high-nickel-chromium steels similar to E1912, Table 1, deformation at high temperatures followed by cooling and reheating, which may be encountered in cyclic operation of high-temperature equipment, may result in grain growth and changes in impact resistance or other properties.[13]

Recently, Rosenhain and Hanson[14] reported an appreciable increase in the Brinell hardness of mild-steel strips containing 0.11 per cent carbon when subjected for about 5 years at 570 deg. fahr. to loads equivalent to one-third to two-thirds of their tensile strength determined at ordinary temperatures. This hardening was observed despite the fact that the samples were reported to have undergone only a very small amount of deformation in the specified interval.

No satisfactory explanation was offered for these effects, but if appreciable hardening can take place generally in steels without deformation, when they are subjected to prolonged stresses at high temperatures, this matter must be considered in the interpretation of flow-test data. On this account hardness and notched-bar impact tests were made on samples of the chromium-vanadium steel, No. 10-518, Table 1, after they had been stressed for long periods in the flow tests.

The impact specimens were prepared from the reduced sections of the flow-test specimens, the notch being placed in the center of the gage length. On account of the small size of the specimens used in the flow tests (0.25 in. diameter), a standard impact-test specimen could not be obtained. Instead, specimens were prepared for a low-range Charpy impact-testing machine and had a length of 2.1 in., a thickness of 0.1 in., and widths varying from 0.14 to 0.2 in., depending upon the diameters of the tested flow-test specimens. Whenever possible, specimens having a width of 0.2 in. were used. However, the results were all converted[15] to equivalent values for specimens having a width of 0.2 in.

Rockwell hardness tests were also made on the broken impact specimens, a $^1/_{16}$-in. ball and 100-kg. load (B scale) being used. The results are summarized together with pertinent flow-test data in Tables 3 and 4.

No measurable changes were found in the hardness or impact resistance at atmospheric temperatures so long as no appreciable deformation was produced in the previous flow tests. Deformation at temperatures at which strain hardening occurs in the flow tests was accompanied quite generally by an increase in the hardness and a decrease in the resistance to impact. In all cases, however, the changes were small. At higher temperatures, 1000 to 1200 deg. fahr., no measurable changes were produced in the impact resistance or hardness by deformation in the flow tests. The flow tests were not continued for such long periods as in the tests reported by Rosenhain and Hanson, and a steel of very different composition was studied. The results indicated, however, that no changes occurred other than those which might normally be expected from strain hardening.

TABLE 3 EFFECT OF PROLONGED STRESS AT DIFFERENT TEMPERATURES ON SUBSEQUENT HARDNESS AND IMPACT RESISTANCE AT ATMOSPHERIC TEMPERATURES OF THE CR-V STEEL, NO. 10-518, LOT A, TABLE 1

(1575 deg. fahr.—slow cool; 1375 deg. fahr.—furnace cool)

Specimen no.	Flow test: Temp., deg. fahr.	Flow test: Load, lb. per sq. in.	Duration of flow test, hours	Final elongation in flow test, Per cent	Reduction of area in flow test, Per cent	Tests at room temperatures subsequent to the flow tests: Impact resistance, ft-lb.	Tests at room temperatures subsequent to the flow tests: Rockwell (B) scale hardness
3149	70	25,000	96	..	..	5.10	85.0
3149	70	35,000	195	0.17	..	5.00	95.9
3142	70	50,000	140	1.87	..	4.68	87.4
120	70	60,000	94	3.47	..	4.55	90.5
120	70	70,000	148	6.24	..	4.48	98.4
126	550	20,000	96	..	..	4.97	83.3
127	550	30,000	121	0.43	..	4.93	86.1
13	550	40,000	164	1.40	..	5.11	86.0
3140	550	50,000	168	2.07	..	4.22	90.8
3140	550	60,000	121	3.84	..	3.08	94.0
3140	550	70,000	168	6.40	..	3.53	98.8
C 114	800	5,000	308	..	..	4.85	85.1
C 115	800	10,000	327	..	..	5.07	93.3
C 118	800	15,900	327	0.06	..	3.23	86.1
116	800	30,000	218	10.13	14.6	3.58	88.5
C 128	1000	2,500	305	..	..	5.26	84.5
C 130	1000	5,100	313	..	..	5.30	84.5
C 135	1000	10,000	291	1.89	2.04	4.45	85.2
C 138	1000	15,000	194	4.10	4.50	4.23	85.7
3151	1000	15,000	288	6.24	5.30	4.36	84.9
C3126	1200	500	315	..	..	5.21	83.4
C3120	1200	1,000	313	..	..	5.11	84.5
3124	1200	1,500	312	..	..	5.21	82.7
3119	1200	3,000	288	..	..	4.99	81.9
C3125	1200	5,000	315	7.8	..	5.18	83.9

TABLE 4 EFFECT OF PROLONGED STRESS AT DIFFERENT TEMPERATURES ON SUBSEQUENT HARDNESS AND IMPACT RESISTANCE AT ATMOSPHERIC TEMPERATURES OF THE CR-V STEEL, NO. 10-518, LOT B, TABLE 1

(1575 deg. fahr.—air cool; 1300 deg. fahr.—furnace cool)

Specimen no.	Flow test: Temp., deg. fahr.	Flow test: Load, lb. per sq. in.	Duration of flow test, hours	Final elongation in flow test, Per cent	Reduction of area in flow test, Per cent	Tests at room temperatures subsequent to the flow tests: Impact resistance, ft-lb.	Tests at room temperatures subsequent to the flow tests: Rockwell (B scale) hardness
C3111	70	30,000	194	0.08	..	5.12	95.0
3107	70	50,000	143	0.38	..	4.47	98.0
C3105	70	70,000	143	0.39	..	4.84	97.3
398	70	70,280	214	1.77	1.20	4.21	97.8
C3148	70	80,000	216	3.13	3.15	3.88	98.5
140	550	30,000	187	..	..	4.18	98.3
142	550	37,750	187	0.02	..	4.13	97.9
C3154	550	40,000	168	..	..	4.22	93.0
C 265	550	62,500	260	0.33	..	4.25	96.4
C3103	550	70,000	91	0.74	..	4.10	97.7
C3102	550	80,000	91	2.19	..	3.20	100.6
266	800	5,000	308	..	..	2.74	97.2
271	800	5,000	327	..	..	3.42	95.7
C 262	800	9,440	197	..	..	2.20	97.1
C 275	800	10,000	327	0.10	..	2.46	97.3
C 147	800	15,000	191	0.18	..	2.61	97.2
C 259	800	22,000	282	0.55	..	2.50	96.6
C 268	800	29,600	866	8.99	1.43	3.78	95.4
252	1000	2,500	322	0.33	..	4.19	98.2
C 251	1000	4,490	489	0.10	..	4.47	96.7
C 143	1000	7,500	305	0.52	..	4.65	95.3
C 250	1000	10,000	383	1.49	..	4.45	97.1
C 394	1000	15,000	288	8.52	13.06	4.46	95.7
287	1200	500	441	0.18	..	4.95	92.3
C 274	1200	785	750	1.13	..	5.25	88.7
C 273	1200	1,500	367	0.57	..	4.92	90.1
C 277	1200	3,000	218	2.77	..	5.06	93.4

5—Trends at 1000 Deg. Fahr. in the System Ni-Cr-Fe

In the tests of the commercial alloys of nickel, chromium, and iron it was impracticable to determine the effects of variations

[13] F. P. Fischer, "Rekristallisationsversuche allgemeiner Art und zahlenmässige Feststellungen über Festigkeitseigenschaften rekristallisierten Flusseisens (Weicheisens)," *Kruppsche Monatshefte*, vol. 4 (1923), p. 77.

[14] W. Rosenhain and D. Hanson, "The Behavior of Mild Steel Under Prolonged Stress at 300° C.," *Jl. Iron and Steel Inst.*, vol. 116 (1927), p. 117.

[15] Since the thickness of the specimens was the same in all cases, the impact values vary directly with the width.

in nickel or chromium on the load-carrying ability of the alloys at high temperatures since the variations in composition were not restricted to these individual metals. It was largely on this account that the twelve metals of group III, Table 1, were prepared. Their approximate locations in the nickel-chromium-iron ternary diagram are shown in Fig. 48.

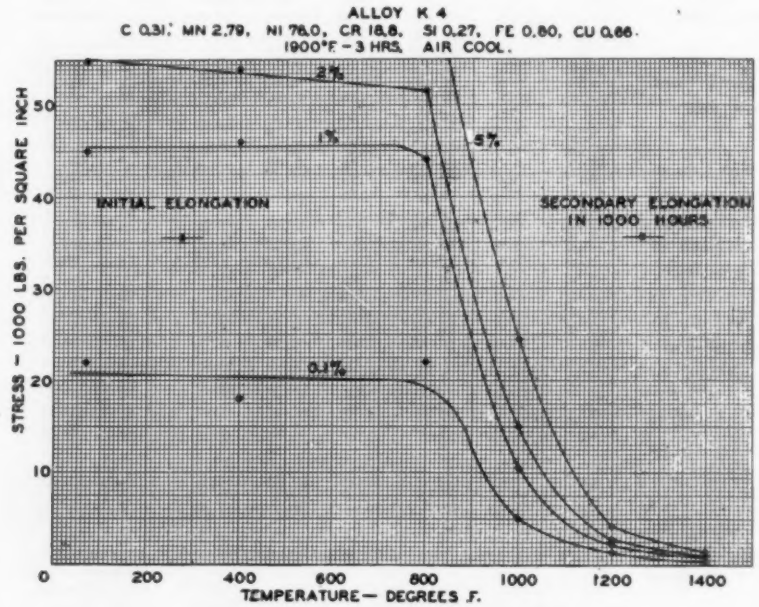


FIG. 30 FLOW CHART FOR ALLOY K-4

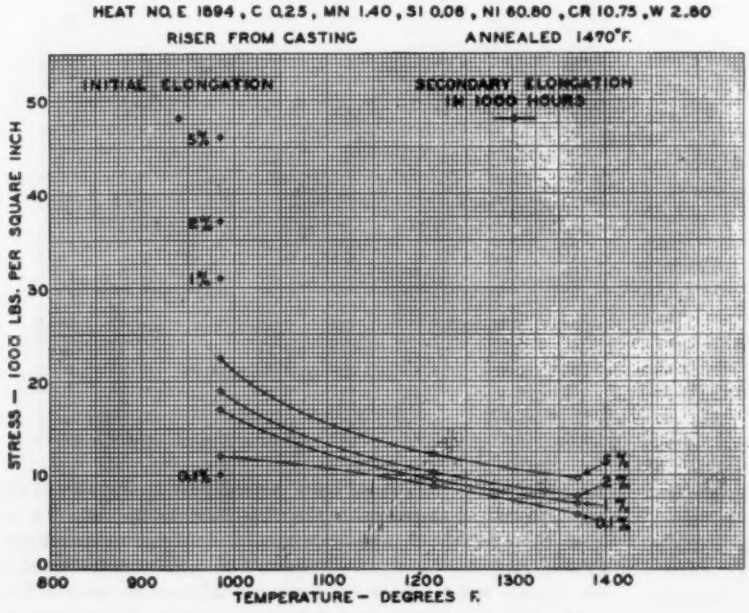


FIG. 31 FLOW CHART FOR ALLOY E1894

The flow tests were not continued for such long periods as those made on the commercial alloys since the principal object was to determine general trends in the system, nickel-chromium-iron, rather than to secure numerical values of the load-carrying ability of the different alloys.

The metals were melted in zirconia crucibles in a high-frequency induction furnace, cast in iron molds (preheated to approximately 390 deg. fahr.), and subsequently heat treated as indicated in Table 1. The heat treatments were selected to produce at least a partial grain refinement, promote diffusion, and leave the alloys in a "softened" condition.

The alloys tested were not pure in the sense that they were free from all elements except nickel, chromium, and iron, since intentional additions were made of carbon, silicon, and manganese. The presence of appreciable proportions of the last three elements probably exerted a pronounced effect upon numerical values of the flow at 1000 deg. fahr., and by acting in different ways in different alloys may have modified somewhat the magnitude of the changes existing in the pure nickel-chromium-iron system. However, it is hardly probable that the general trends were upset by the presence of the carbon, silicon, or manganese, and since most industrial-alloy castings of this general type contain appreciable amounts of these three metals, the results are probably closer to those obtainable in practice than would be the case with pure metals.

Table 1 shows that not all of the alloys contained the desired amounts of carbon, silicon, and manganese. The carbon contents were 0.20 to 0.35 per cent in most cases, but some of the alloys contained as much as 0.50 per cent or as little as 0.11 per cent. The manganese was between 0.6 and 0.9 per cent, but the silicon varied from about 0.5 to 1.1 per cent. These variations probably affected the numerical values of the flow appreciably, but in general may be disregarded since only the major trends in the nickel-chromium-iron system are to be considered.

The specimens for the flow tests were cut from the castings at points midway between the outside and the center in the direction of the long axis. The Brinell hardness values subsequent to the recorded heat treatments are given in Table 1, and have been used as the basis of the three-dimension model in Fig. 49 to give a better picture of the conditions in which the metals were tested. The time-elongation curves from the flow tests at 1000 deg. fahr. are given in Fig. 50, while the relations between the flow and the applied loads are summarized in Fig. 51.

As in the case of the alloys of groups I and II, Table 1, the load-carrying ability may be represented by the stresses producing 1 per cent elongation in the first 1000 hours, and a comparison of the metals on this basis is given in the three-dimension ternary model in Fig. 52. Fig. 52 shows that the addition of nickel to the iron, containing about 0.2 per cent carbon, etc. (Table 1) resulted in no marked changes in the stresses producing 1 per cent elongation in the first 1000 hours.

The addition of chromium to iron containing about 0.2 per cent carbon, etc. (Table 1), did not improve the load-carrying ability at 1000 deg. fahr. appreciably until the chromium content was in the neighborhood of 50 per cent. In fact, the iron alloy with 30 per cent chromium showed the lowest resistance to flow of all of the alloys tested and that with 14 per cent chromium, representing the composition of "stainless steel," was about on a par with steel containing no chromium.

On the other hand, the addition of a little chromium to the

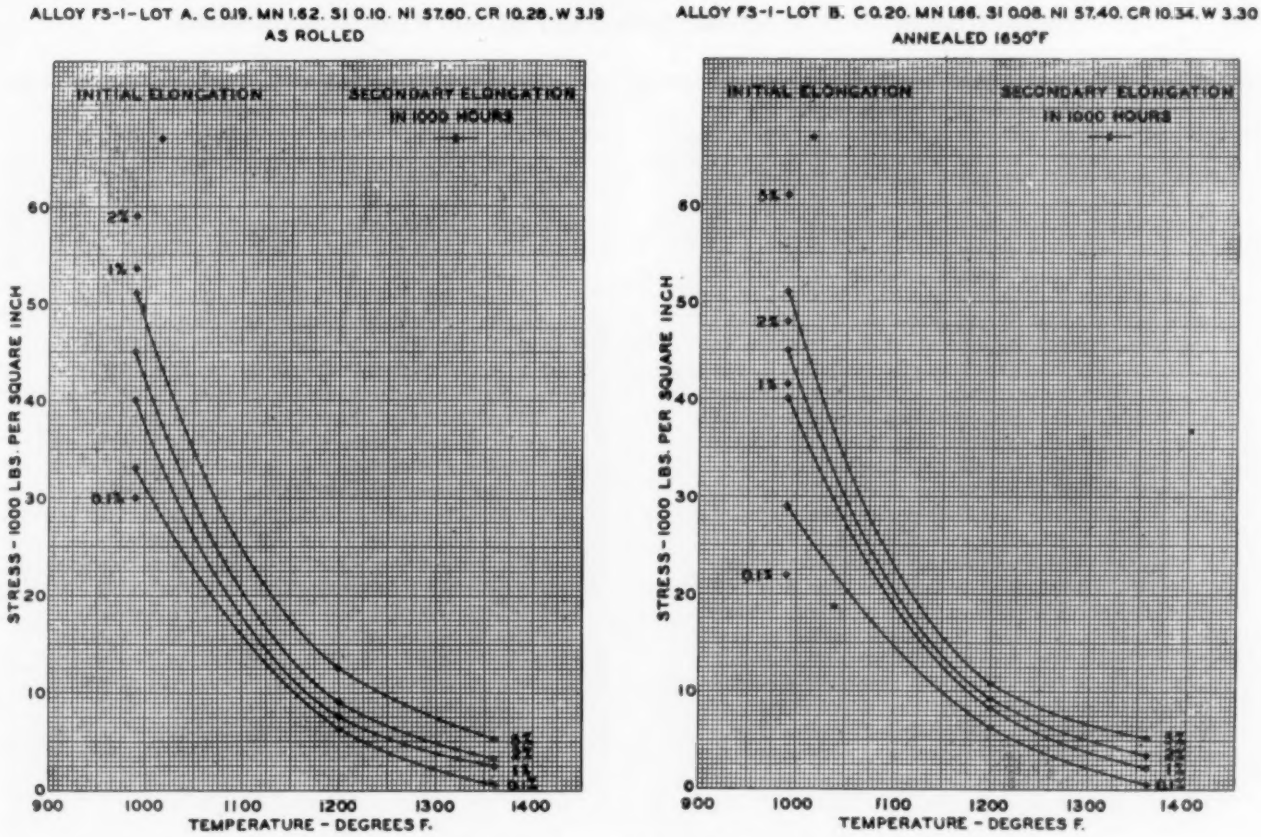


FIG. 32 FLOW DATA FOR ALLOY FS-1, LOTS A AND B

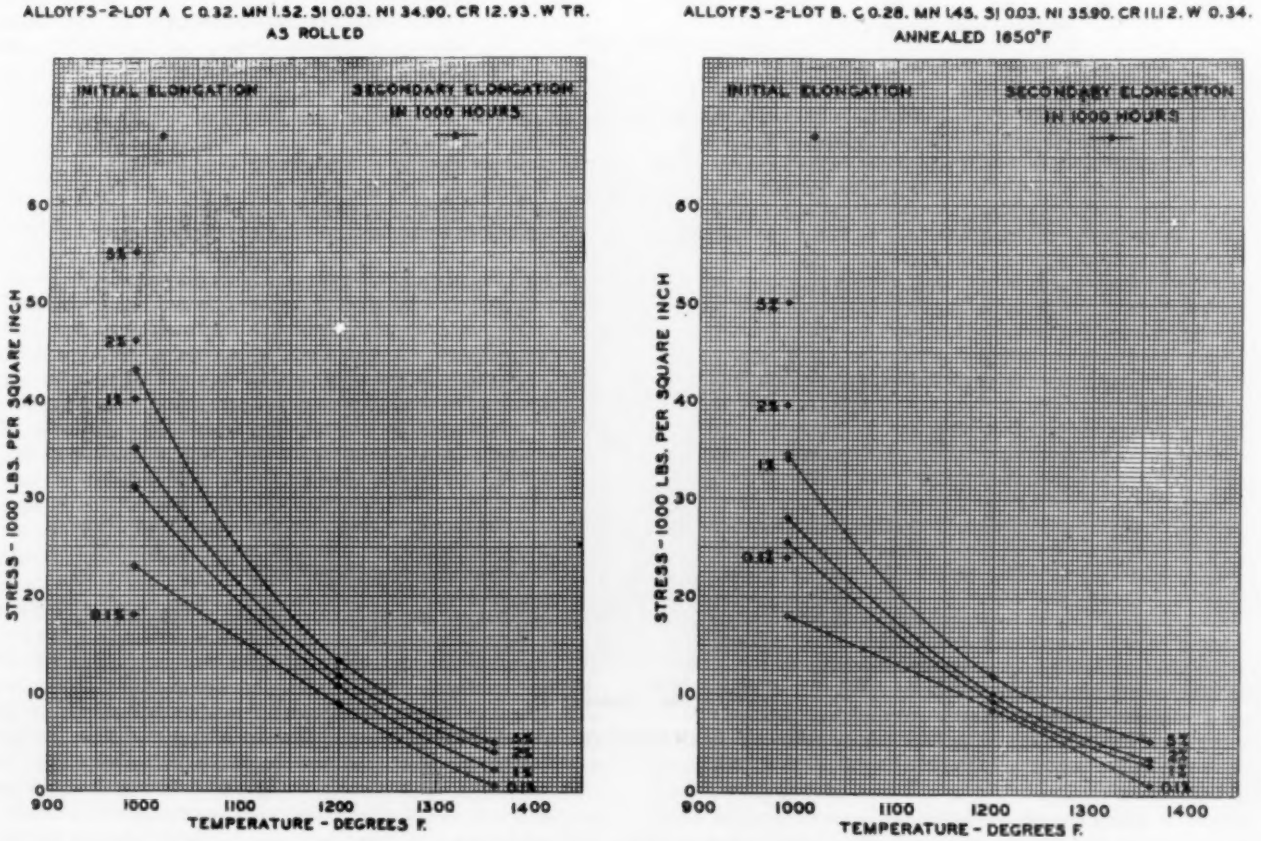


FIG. 33 FLOW DATA FOR ALLOY FS-2, LOTS A AND B

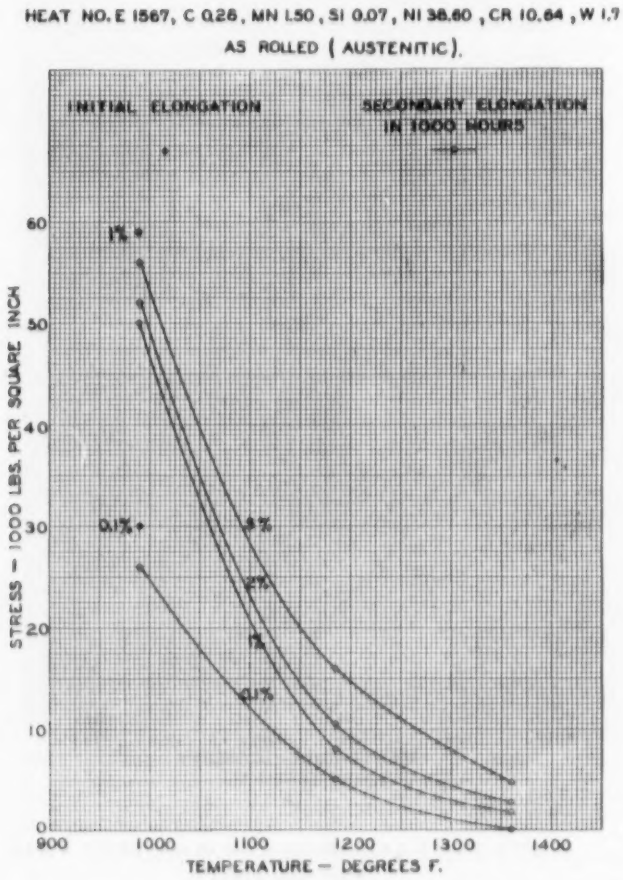


Fig. 34 Flow Chart for Alloy E1567

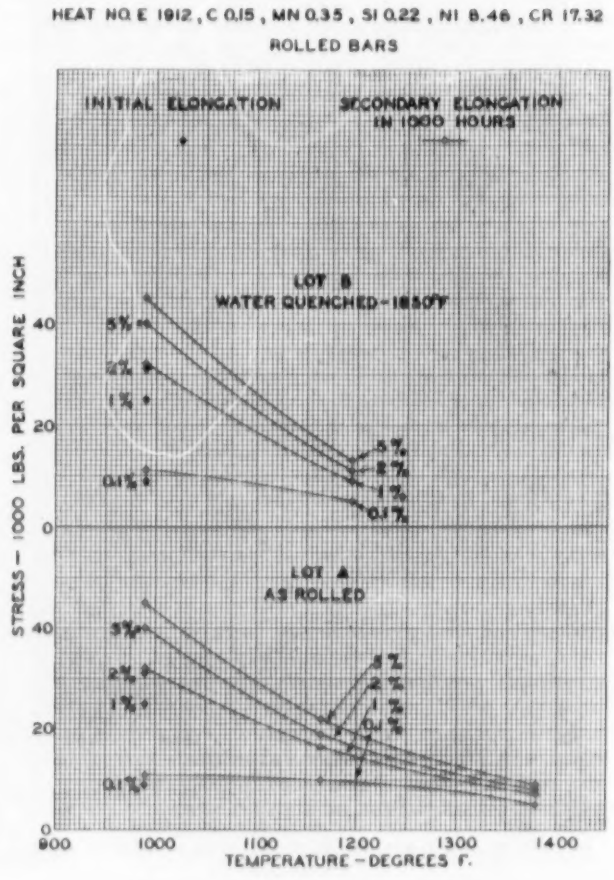


Fig. 36 Flow Chart for Steel E1912

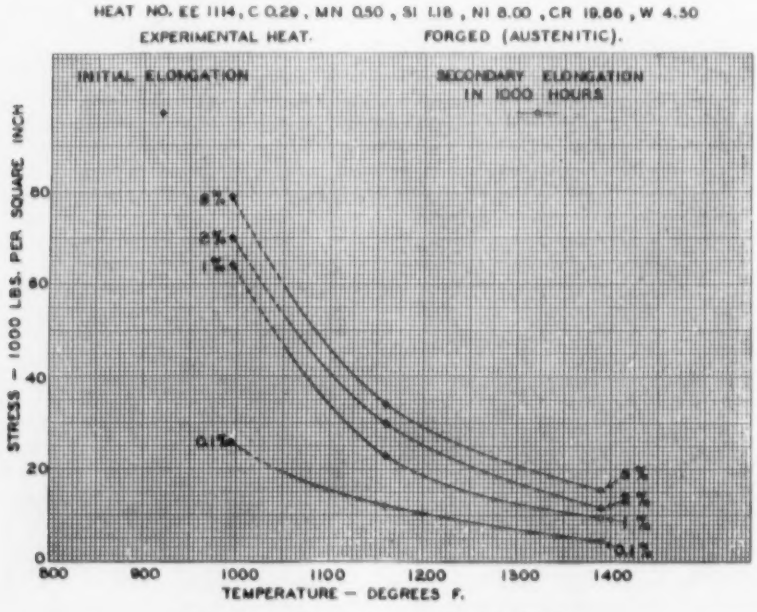


Fig. 35 Flow Chart for Steel EE1114

iron alloys with 40 per cent or more of nickel, produced a marked increase in the load-carrying ability. The most effective proportions of chromium were between 5 and 20 per cent; higher proportions produced further increases but not in the same degree.

These results explain in part the popularity of the commercial iron-nickel-chromium casting alloys containing about 60 to 65 per cent of nickel and 10 to 20 per cent of chromium. This composition is in the range of high load-carrying ability, although it does not have the best resistance to flow.

The strongest of the alloys at 1000 deg. fahr. were those with little or no iron, 50 to 80 per cent nickel, and 50 to 20 per cent chromium. The highest load-carrying ability was obtained in the castings containing about 50 per cent each of nickel and chromium (alloy 1008, Table 1) but this alloy would be expected to offer difficulties in the foundry and had the serious disadvantage of fracturing with low elongation (Fig. 50). The remedy for this latter condition may be in decrease in the impurities, but foundry difficulties would be expected to persist under such conditions. A more practical solution of the problem of securing higher resistance to flow at high temperatures would probably be in additions of other elements to the alloys containing 60 to 65 per cent of nickel and 10 to 20 per cent of chromium. Tungsten is one of the additions used industrially for this purpose, as is shown by the composition of the commercial alloys of group I, Table 1.

In the alloys containing 50 per cent or more of nickel and 10 to 20 per cent of chromium, it would seem to be advisable to keep the chromium close to 20 per cent, since a decrease to 15 per cent or less brings the metal into a range of rapidly decreasing load-carrying ability, as is shown in Fig. 52. This means that the composition variations ordinarily encountered in commercial production would result in wider variations in properties in the nickel-iron alloys with 5 to 15 per cent chromium than in those with 15 to 20 per cent. Furthermore, lowering the chromium should also tend to lower the oxidation resistance of the metal.

It is of interest to note that the iron-nickel-chromium alloy having the nominal composition of 18 per cent chromium and 8 per cent nickel, which is now so widely used in the wrought condition, is in the field of low load-carrying ability at 1000 deg. fahr. It cannot compete with some of the other alloys in castings for service requiring high resistance to flow, but is attractive on account of excellent resistance to oxidation and because it can be formed either hot or cold.

While this and some of the other alloys in the nickel-chromium-iron system were found to have relatively low resistance to flow at 1000 deg. fahr. it should be recognized that improvement may be obtained through modifications in treatment or by varying the proportions of elements such as carbon, silicon, etc. An example is found in the iron alloy containing 14 per cent chromium. Tests on two different ingots containing, respectively, 0.24 and 0.37 per cent carbon showed (Fig. 51) that the alloy with 0.37 per cent carbon had a much better resistance to flow at 1000 deg. fahr. than alloy with 0.24 per cent.

It should be recognized that variations in the general structure (cast or wrought metal), in heat treatment, or in the proportions of elements such as carbon and silicon, may affect the load-carrying ability to a greater degree than appreciable variations in the proportions of nickel, chromium, or iron at least in some parts of the system (Fig. 52).

V—SUMMARY

Flow tests were made at different temperatures on three groups of alloys. The eleven metals of the first group comprised one nickel-chromium alloy, six nickel-chromium-iron alloys, with or without tungsten, and four low-chromium steels containing also vanadium, tungsten, or molybdenum. The flow characteristics of these metals are summarized in charts given in the report.

Two carbon steels, two low-nickel-chromium steels, and a 3½ per cent nickel steel comprised the second group of metals which were tested at 700 deg. fahr. When annealed so as to

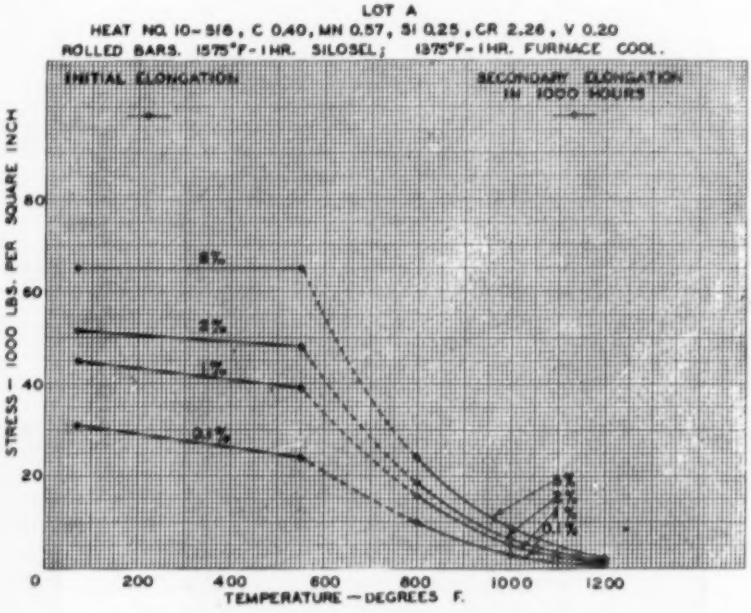


FIG. 37 FLOW CHART FOR STEEL 10-518, LOT A

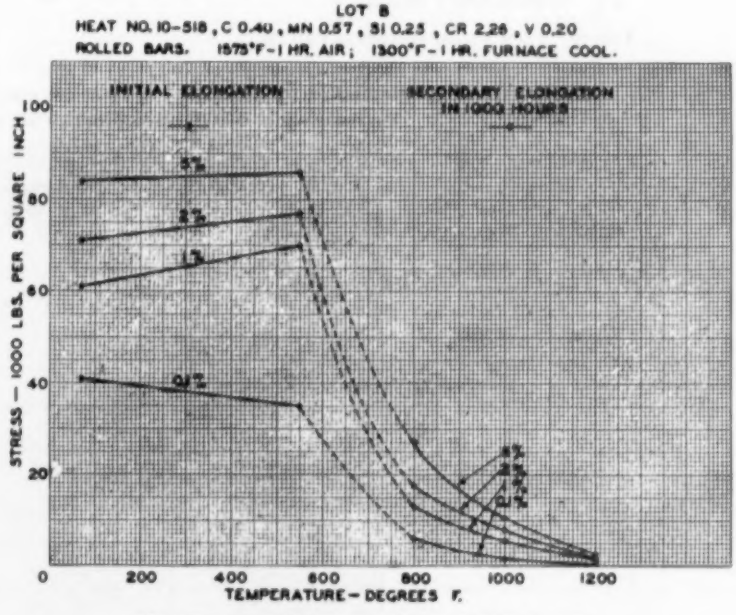


FIG. 38 FLOW CHART FOR STEEL 10-518, LOT B

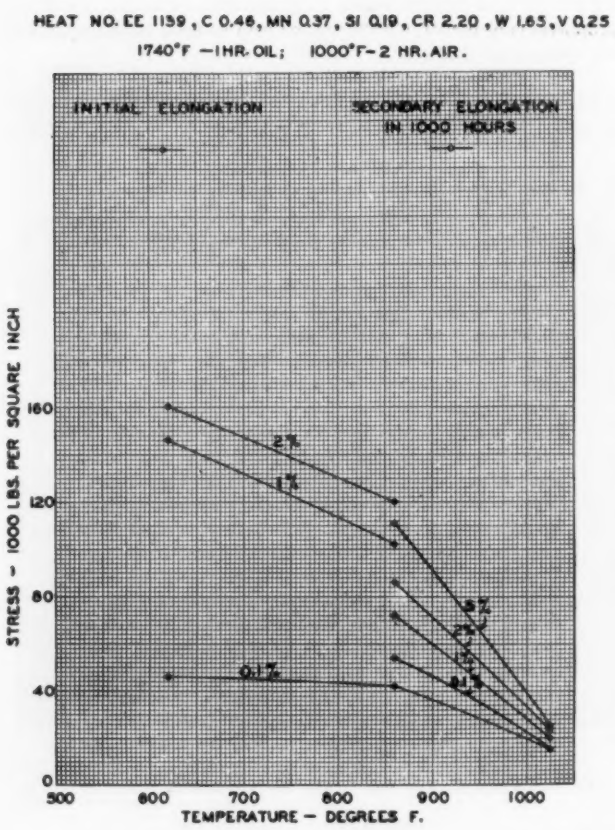


FIG. 39 FLOW CHART FOR STEEL EE1139

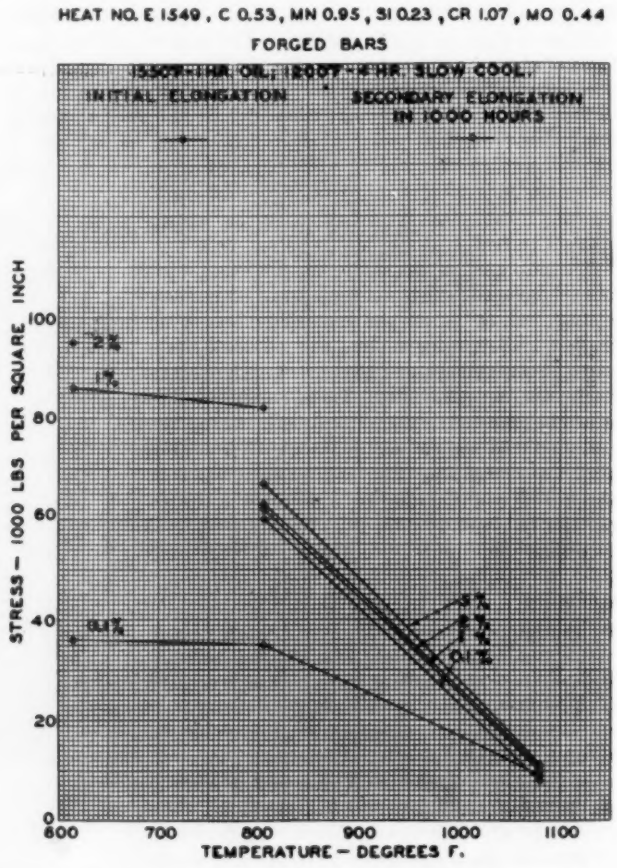


FIG. 40 FLOW CHART FOR STEEL E1549

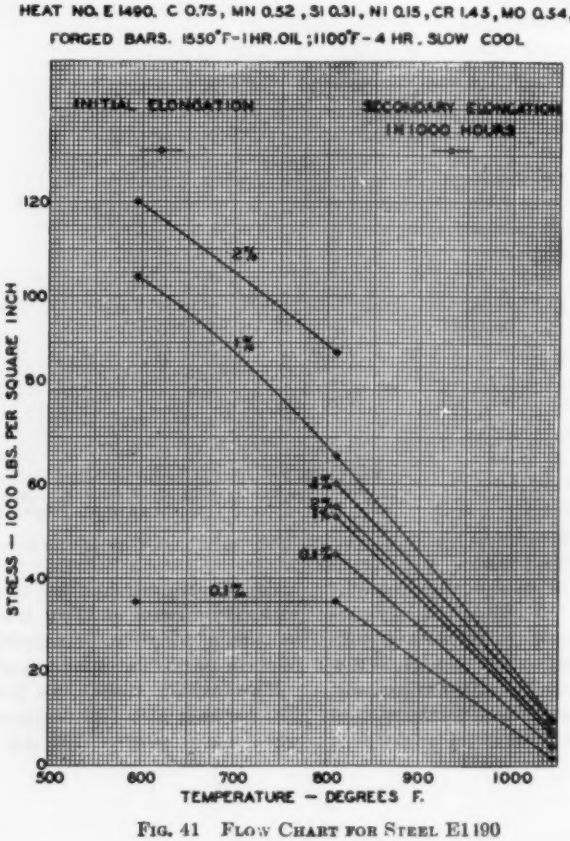


FIG. 41 FLOW CHART FOR STEEL E1190

reproduce the conditions existing in large forgings the plain carbon steel containing 0.45 per cent carbon showed better load-carrying ability than that with 0.23 per cent carbon, and resisted flow at 700 deg. fahr. about as well as the steels with 0.35 per cent carbon and either $3^1/_2$ per cent nickel or a combination of $1^1/_4$ to $1^3/_4$ per cent nickel and $^1/_2$ to 1 per cent chromium. However, other characteristics of the alloy steels may give them marked superiority, The addition of chromium to the iron did not greatly improve its load-carrying ability until the chromium content reached about 50 per cent. In fact, the iron alloy with 30 per cent chromium showed the lowest resistance to flow of all of the alloys tested at 1000 deg. fahr., and the one with 14 per cent chromium, representing the composition of "stainless steel," was about on a par with steel containing no chromium.

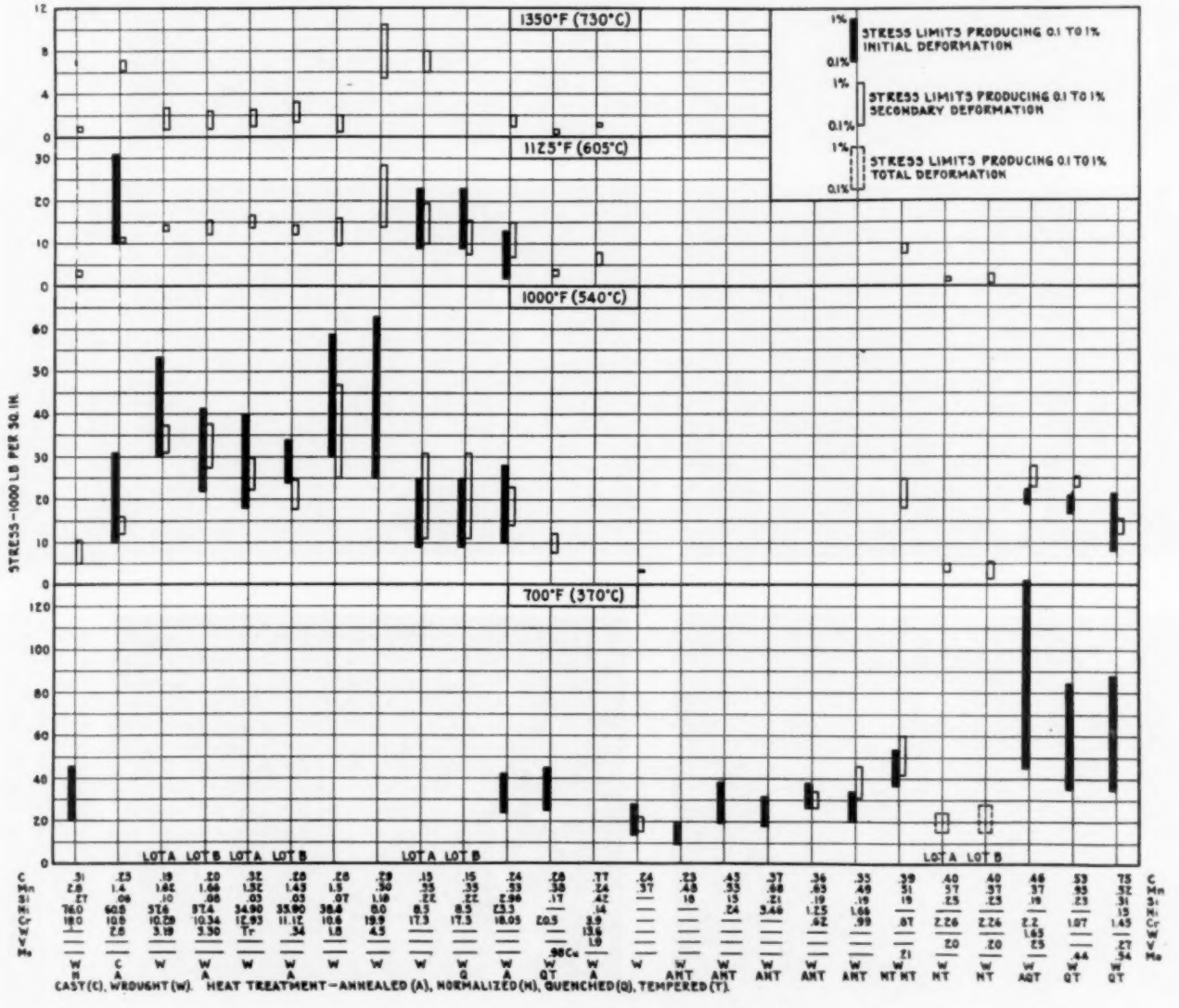


FIG. 42 COMPARISONS OF ALLOYS ON THE BASIS OF THE STRESSES PRODUCING BOTH INITIAL AND SECONDARY DEFORMATION
(Note: Secondary deformation is calculated for 1000 hours.)

such, for example, as superior resistance to grain growth on recrystallization which results from deformation and subsequent heating in cyclic temperature operation.

The third group of metals consisted of nickel-chromium-iron-alloy castings melted in a high-frequency induction furnace. Their chemical compositions were selected to cover a broad area in the nickel-chromium-iron system, and the tests of the metals were made at 1000 deg. fahr. after annealing or normalizing to produce softening, grain refinement, and diffusion.

It was found that the addition of nickel to iron containing about 0.20 per cent carbon, 0.75 per cent manganese, and 0.7 per cent silicon resulted in no marked changes in the stresses required to produce 1 per cent elongation in the first 1000 hours.

On the other hand, the addition of a little chromium to the iron alloys with about 40 per cent or more of nickel, produced a marked increase in the load-carrying ability. The most effective amounts of chromium were between 5 and 20 per cent; higher proportions produced further increases, but these were not commensurate with increases produced by additions below 20 per cent.

The highest load-carrying ability was obtained in the castings containing about 50 per cent each of nickel and chromium, but this alloy would be expected to offer difficulties in the foundry, and, with the proportions of carbon, silicon, and manganese present, had the serious disadvantage of low ductility.

In the case of the wrought commercial alloy containing 0.15 per cent carbon, 18 per cent chromium and 8 per cent nickel,

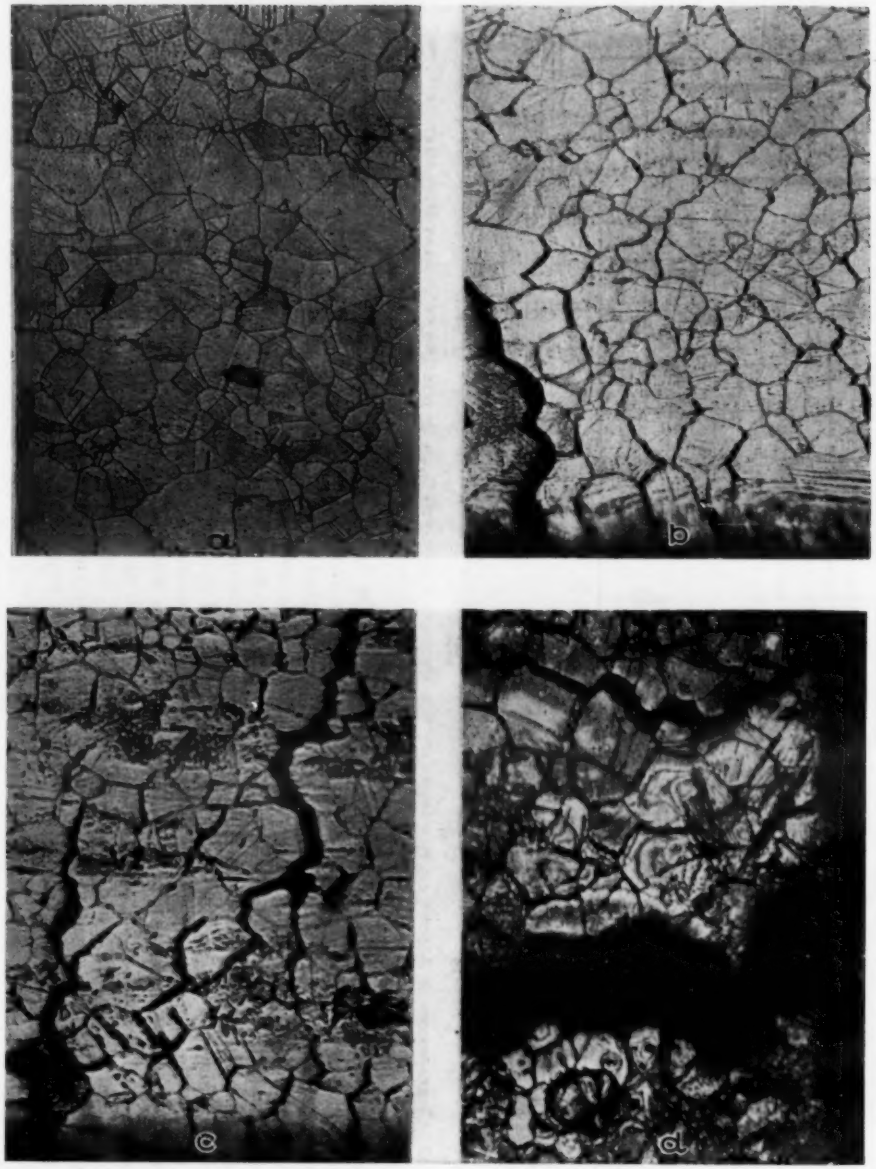


FIG. 43 STRUCTURES OF THE ROLLED BARS (LOT A) OF ALLOY E1912 AFTER TEST AT DIFFERENT TEMPERATURES. ×250

a 40,000 lb. per sq. in., at 990 deg. fahr. for 560 hr. *b* 50,000 lb. per sq. in. at 990 deg. fahr. for 90 hr.; metal 1/8 in. back of fracture at surface of specimen. Total elongation, 15.5 per cent. *c* 28,000 lb. per sq. in. at 1165 deg. fahr. for 93 hr.; metal 1/8 in. back of fracture at surface of specimen. Total elongation, 7.7 per cent. *d* 10,000 lb. per sq. in. at 1380 deg. fahr. for 168 hr.; metal 1/8 in. back of fracture at surface of specimen. Total elongation, 4.4 per cent. Etched with 2 parts conc. HCl, 1 part conc. HNO_3, 1 part H_2O_2, and 2 parts glycerin.

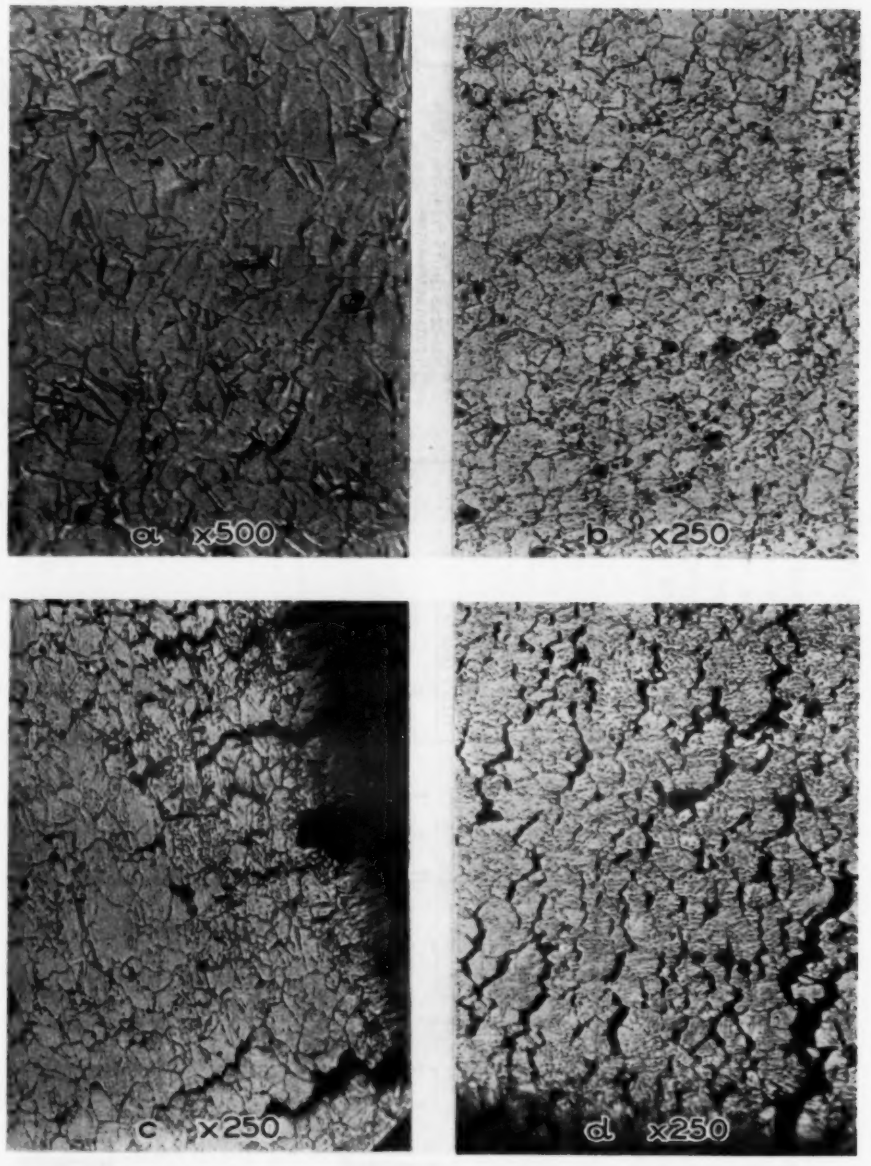


FIG. 44 STRUCTURES OF THE WATER-QUENCHED BARS (LOT B) OF ALLOY E1912 BEFORE AND AFTER TEST AT DIFFERENT TEMPERATURES

a Original quenched bar. *b* 40,000 lb. per sq. in. at 990 deg. fahr. for 520 hr.; unbroken with 6.1 per cent elongation. *c* 50,000 lb. per sq. in. at 990 deg. fahr. for 93 hr.; metal 1/8 in. back of fracture at surface of specimen. Total elongation, 15.5 per cent. *d* 20,000 lb. per sq. in. at 1195 deg. fahr. for 223 hr.; metal 1/8 in. back of fracture at surface of specimen. Total elongation, 10.1 per cent. Etched as indicated in Fig. 43.

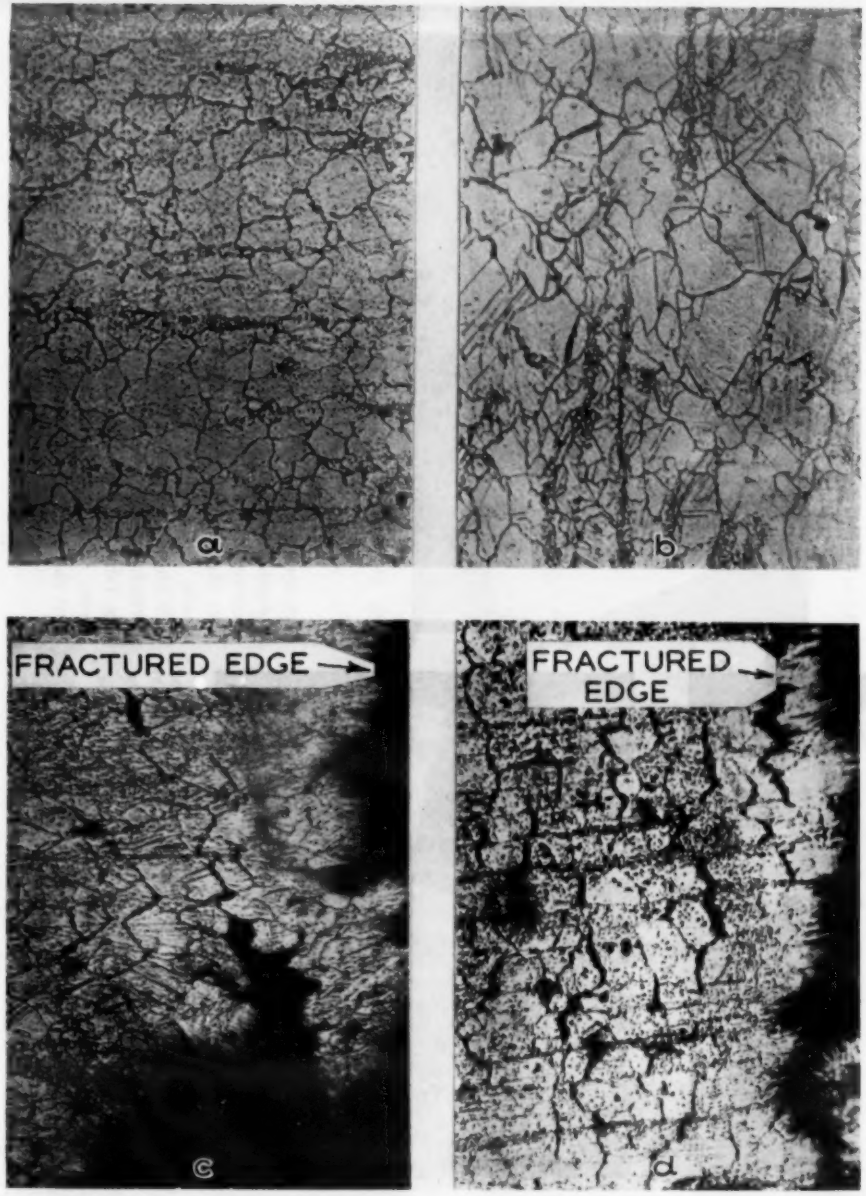


FIG. 45 STRUCTURE OF THE FORGED BARS OF ALLOY EE1114 AFTER TEST AT DIFFERENT TEMPERATURES. ×250

a 78,400 lb. per sq. in. at 995 deg. fahr. for 79 hr.; metal 1/8 in. back of fracture near center of bar. Total elongation, 10.2 per cent. *b* 25,000 lb. per sq. in. at 1160 deg. fahr. for 106 hr. Unbroken with total elongation of 0.32 per cent. *c* 39,800 lb. per sq. in. at 1160 deg. fahr. for 106 hr.; section at fracture. Total elongation, 4.9 per cent. *d* 24,900 lb. per sq. in. at 1390 deg. fahr. for 77 hr.; section at fracture. Total elongation, 2.7 per cent. Etched as indicated in Fig. 43.

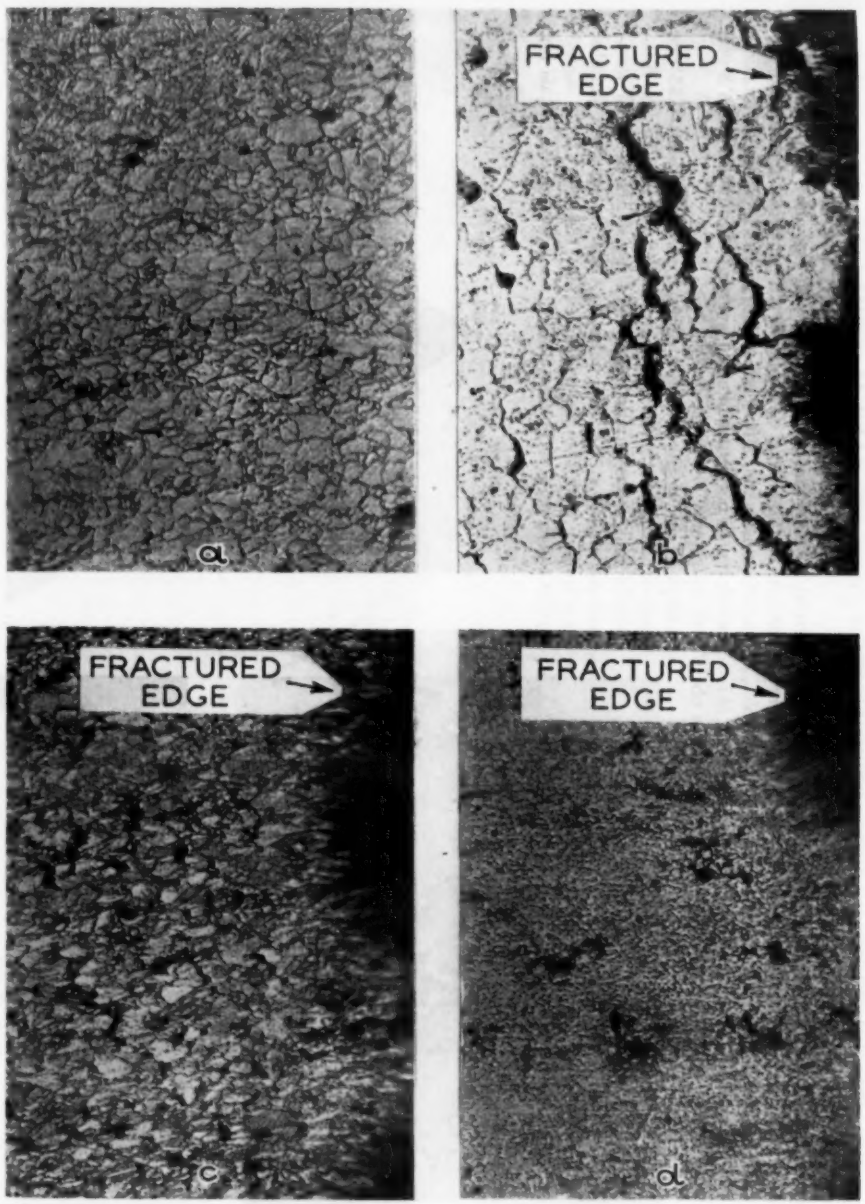


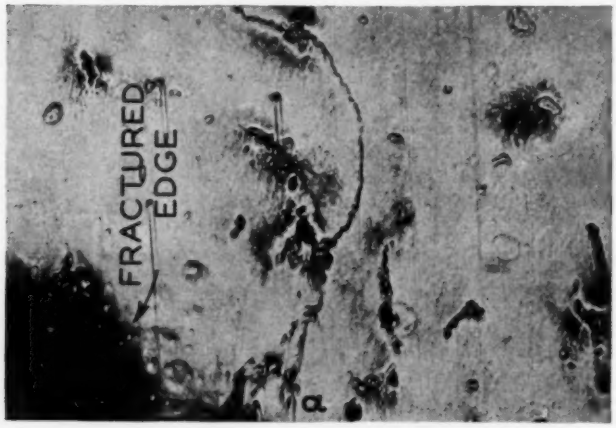
FIG. 46 STRUCTURES OF THE ROLLED BARS OF ALLOY E1567 BEFORE AND AFTER TEST AT DIFFERENT TEMPERATURES. ×250

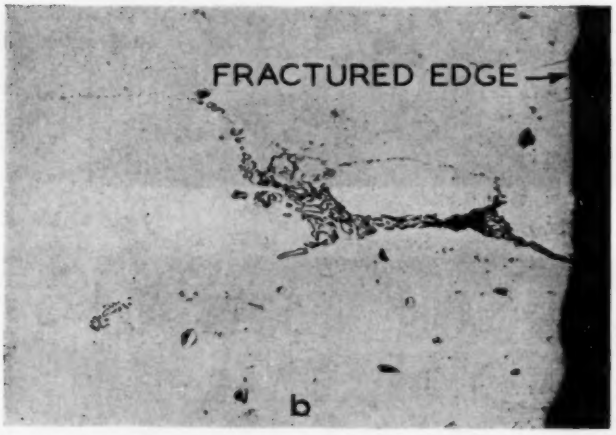
a Original bar of lot B. *b* 23,300 lb. per sq. in. at 1185 deg. fahr. for 180 hr.; metal at fracture. Total elongation, 2.4 per cent (Lot A). *c* 35,000 lb. per sq. in. at 1185 deg. fahr. for 138 hr.; metal at fracture. Total elongation, 13 per cent. *d* 10,000 lb. per sq. in. at 1360 deg. fahr. for 70 hr.; metal at fracture. Total elongation, 37 per cent. Etched as indicated in Fig. 43.

intercrystalline weakness was observed at temperatures between 1165 and 1380 deg. fahr., and to some extent also at 990 deg. fahr. This was noted in both fine-grained and coarse-grained samples, although the latter seemed to show the greater tendency in this direction. A similar tendency but in a smaller degree was observed at 1160 and 1390 deg. fahr. in the alloy containing 0.3 per cent carbon, 1 per cent silicon, 4.5 per cent tungsten, 20 per cent chromium, and 8 per cent nickel, and at 1185 deg. in an alloy containing 38 per cent nickel, 11 per cent chromium, and 1.8 per cent tungsten.

In the case of the wrought alloy containing about 0.15 per cent carbon, 18 per cent chromium, and 8 per cent nickel, fine-grained and coarse-grained samples showed about equal load-carrying ability at 990 deg. fahr., but the fine-grained samples were weaker than the coarse-grained at 1180 deg. This should not be taken as a recommendation of coarse-grained metals for high-temperature service since fracture may occur with less total elongation than in the corresponding fine-grained metals and so counteract any advantages from lower rates of flow. There is also the possible added disadvantage of weakness in resisting shock.

With one exception the wrought commercial alloys of nickel, chromium, and iron which showed superiority at about 1380 deg. fahr. did not necessarily show superiority at lower temperatures. This exception was the alloy containing about 20 per cent chromium, 8 per cent nickel, 4.5 per cent tungsten, 0.3 per





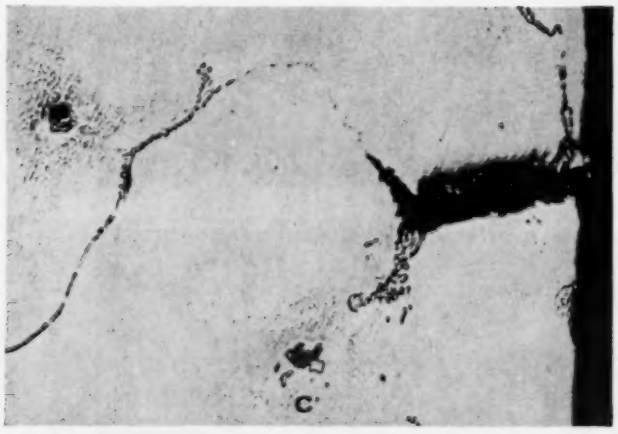


FIG. 47 STRUCTURES OF THE CAST AND ANNEALED SPECIMENS OF ALLOY E1894 AFTER TEST AT DIFFERENT TEMPERATURES. ×100

a 28,000 lb. per sq. in. at 1215 deg. fahr. for 64 hr.; metal at fracture. Total elongation 18 per cent.
b 14,000 lb. per sq. in. at 1370 deg. fahr. for 41 hr.; metal at fracture. Total elongation, 35.4 per cent.
c Same as *b* but at surface 1/8 in. back of fracture. Etched as indicated in Fig. 43.

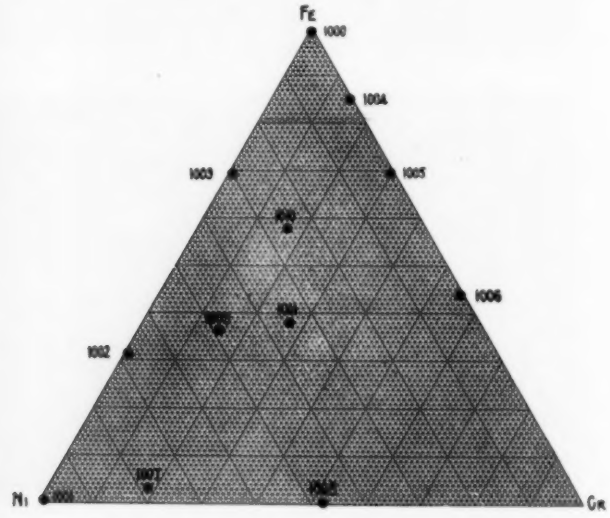


FIG. 48 LOCATIONS OF THE ALLOYS OF GROUP III, TABLE 1, ON THE NICKEL-CHROMIUM-IRON TERNARY DIAGRAM
(The melt numbers given in the chart correspond to those given in Table 1.)

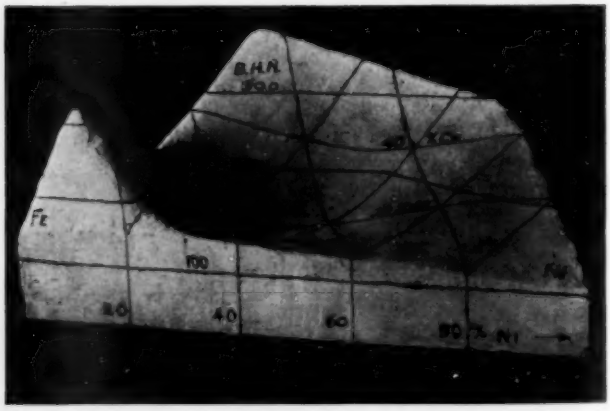


FIG. 49 SUMMARY OF THE BRINELL HARDNESS OF THE ALLOYS IN GROUP III, TABLE 1, IN THE CONDITIONS TESTED
(This model is based on data given in Table 1.)

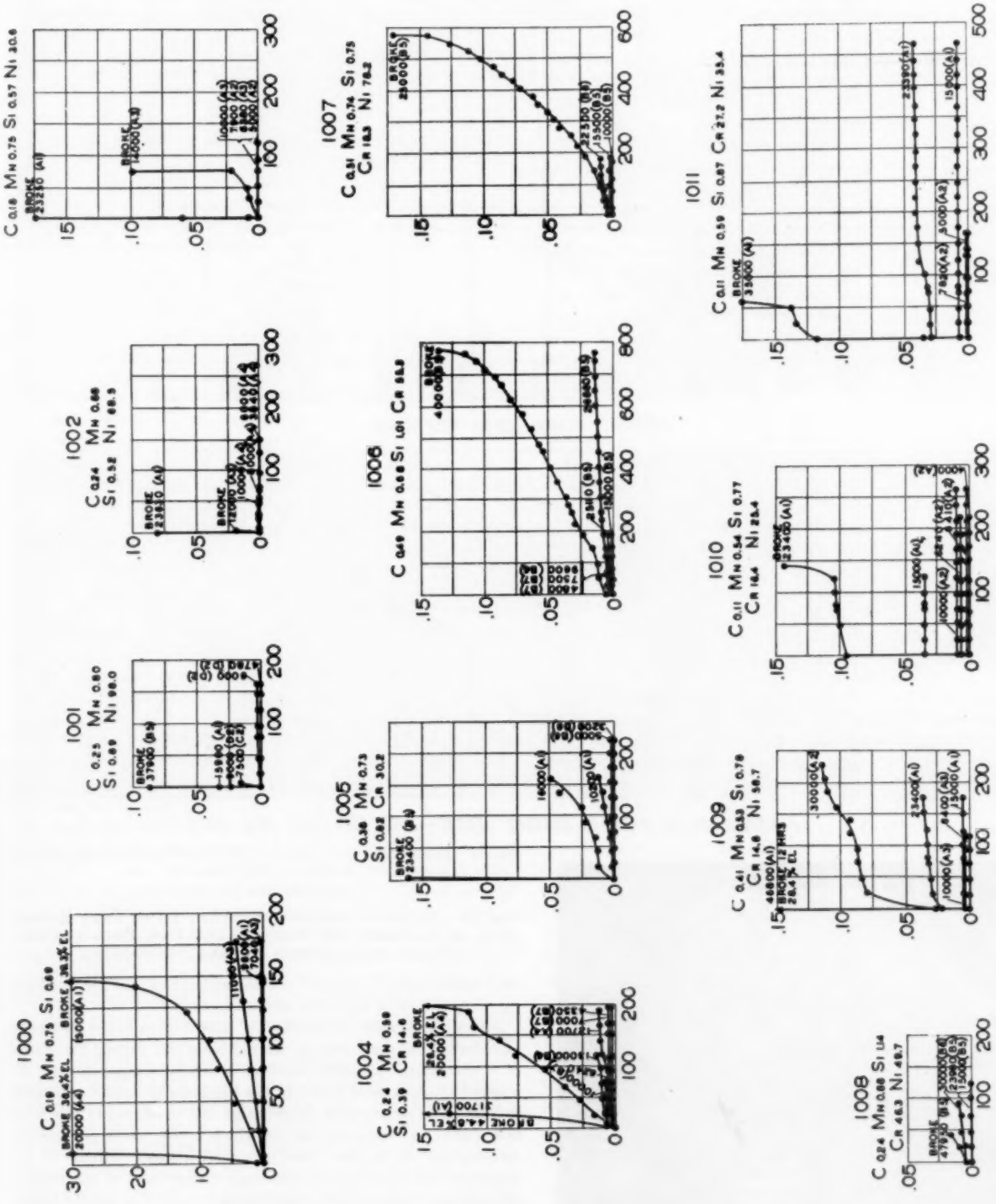


FIG. 50 TIME-ELONGATION CURVES FROM TESTS AT 1000 DEG. FAHR. OF THE ALLOYS OF GROUP III, TABLE 1

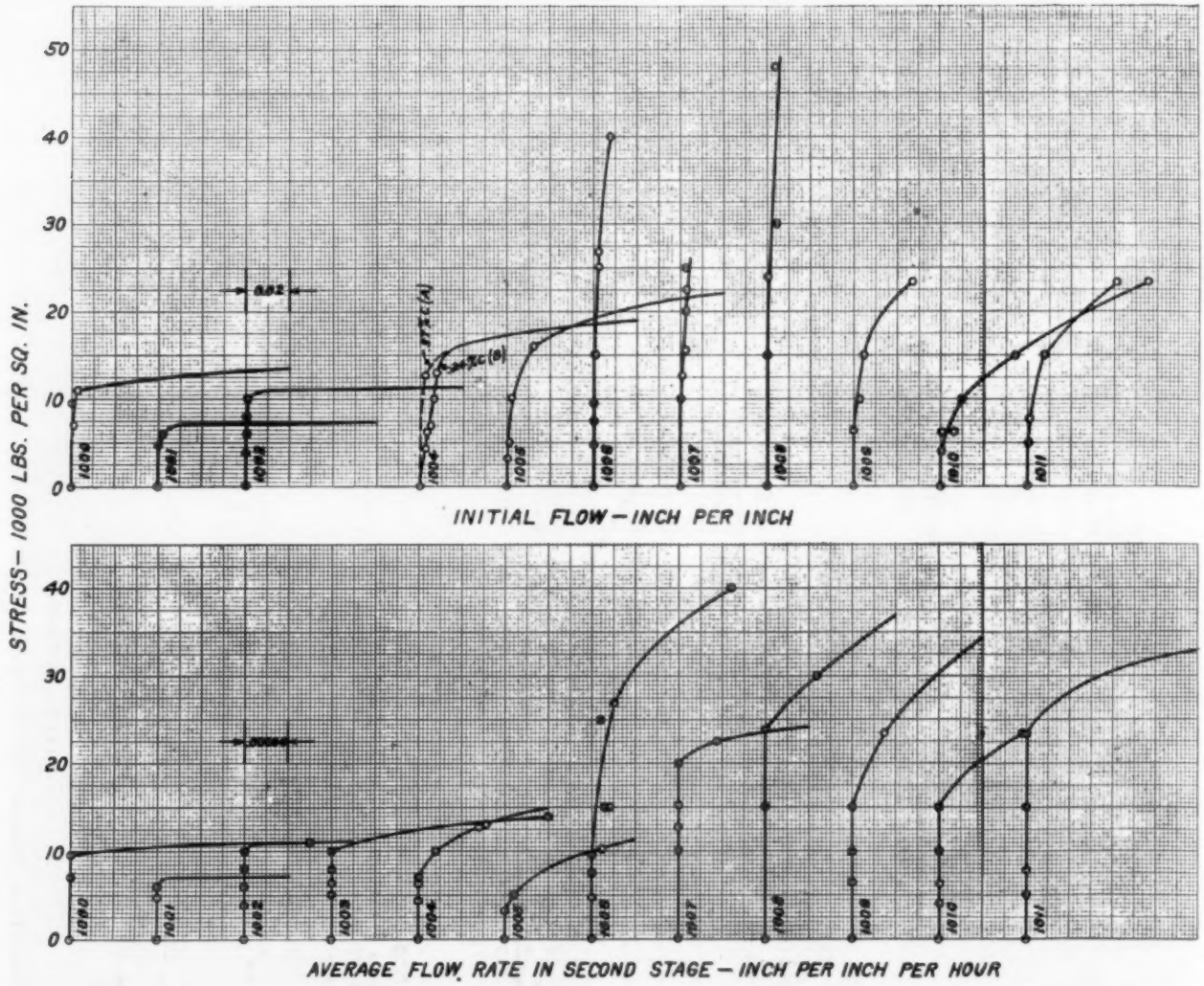


FIG. 51 FLOW DATA FOR TESTS AT 1000 DEG. FAHR. ON THE ALLOYS OF GROUP III, TABLE 1
(Obtained from Fig. 50.)

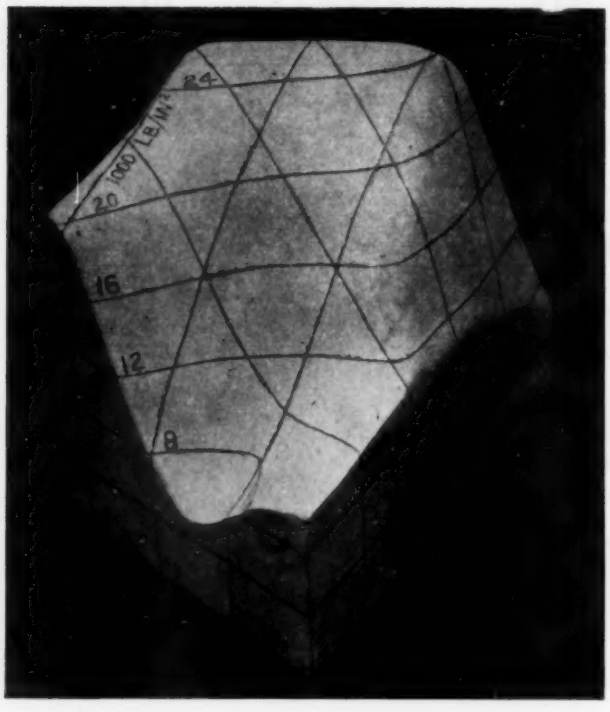


FIG. 52 (AT LEFT) STRESSES PRODUCING 1 PER CENT ELONGATION IN THE FIRST 1000 HOURS AT 1000 DEG. FAHR. IN CAST ALLOYS OF THE SYSTEM NICKEL-CHROMIUM-IRON

cent carbon, and 1 per cent silicon, which had the highest load-carrying ability of all of the alloys of group I.

A study was made of the effect of deformation in the flow tests at different temperatures on the hardness and impact resistance of a chromium-vanadium steel at ordinary temperatures. No marked changes were found in the hardness or impact resistance so long as no measurable deformation was produced in the prior flow tests. At temperatures at which strain hardening occurs, deformation in the flow tests produced a small increase in the hardness and a decrease in the impact resistance of the steel at atmospheric temperatures; deformation in the flow tests at higher temperatures did not affect these at atmospheric temperatures.

ACKNOWLEDGMENTS

The authors are indebted to H. K. Herschman, Associate Metallurgist, for the micrographs included in this report, to H. A. Bright, Chemist, for the chemical analysis of the alloys, and to H. C. Cross, Junior Metallurgist, for carrying out the hardness and impact tests on the flow-test specimens and for assistance, from time to time, in the flow tests. The very efficient aid of Dr. R. L. Dowdell, Senior Metallurgist, and W. A. Tucker, Senior Laboratory Assistant, has contributed greatly to the progress of the work in its later phases and is gratefully acknowledged.

Discussion

F. K. Ziegler.[16] While the results obtained in the tests upon various metals and alloys at elevated temperatures will doubtless be of considerable value to designers of structures operating at temperatures up to 1400 deg. fahr., there is still a distinct need for data on heat-resisting alloys which operate at temperatures up to 2000 deg. fahr. The manufacturers of heat-resisting alloys are at present using figures which are based upon a very high factor of safety, since no comprehensive experiments have been carried on at temperatures in the range where heat-resisting alloys are most economical to use.

Every manufacturer of heat-resisting alloys hopes that in the very near future some work will be done at the high temperatures, and it is suggested that such work be undertaken as soon as possible and that the work be carried on with the same degree of care and accuracy shown by the authors of this paper.

John L. Cox.[17] Few who have not had close association with the determination of creep stresses have any proper appreciation of the tedious drudgery involved in their evaluation and graphic presentation in useful form. The paper is the epitome of over two years' steady work and contains much information nowhere else obtainable with anything like its degree of accuracy. The thanks of the engineering fraternity are due its authors for their painstaking work.

The data given largely concern alloys of a type coming steadily into more general use, as to the behavior of which under prolonged loading at elevated temperatures we have had little real knowledge. In the usual tables of properties of such metals are found the tensile strength and yield point under short-time testing. What the behavior of the materials was under long exposure to stress at high temperatures has been largely a matter of conjecture or of experience to those who have used them. We are now in the way of getting much nearer to the facts, for in spite of many attempts to find a correlation factor between the short- and long-time testing, nothing satisfactory has yet been developed nor is likely to be.

So many possible variables enter into long-time testing that in spite of the utmost care the results of different laboratories, even of the first class, on duplicate specimens, still vary widely. Differences in the sensitivity of the provision for measuring elongation, differences in the distribution of heat in the furnace, differences in the accuracy at which the temperature is maintained constant, differences in the methods of applying the load, and differences in the place and manner of installing even identical equipment have their influences on the results.

In making use of such figures as those produced by the authors, several points must be considered. The curves resulting from all such researches are at best the average of only a few specimens, generally from but one bar or casting of their respective compositions. One must not assume that such compositions always give equal results. The specimens broken may be the best or the worst of their kind, or intermediate in quality. Variations in composition are bound to occur in attempts to run identical compositions, and allowances must be made for all such variations. In spite of these considerations, the more accurate information now available permits the use of a lower factor of safety or a lower coefficient of ignorance.

In many cases the strength shown at higher temperatures will seem astonishingly small in view of known performances of the materials. When such is the case, it probably means that the conditions of service are less severe than supposed, that there may be cycles of alternate heating and cooling to reduce the time at maximum temperature, or that the working heat of the parts is not as high as supposed, or that actual changes of shape were greater than supposed.

Attention may be called to the differences between curves showing 1 per cent elongation in 10,000 hours, the minimum of the paper, and in 100,000 hours, available in one or two instances. This may be as much as 2 to 1, the longer loading giving the lower reading. As 10,000 hours is practically a year's use and 100,000 hours ten years', it is only in cases where practically no distortion is permissible, as in steam turbines, that creep in a longer time or less creep in a shorter time is of real interest.

It might be pointed out that alloy E-1912 is higher in carbon than is advisable for V2A when used at the temperatures of the test. The carbon for such service should not exceed 0.07 and is preferably lower if embrittlement is to be avoided, it being usually produced by carbide precipitation at grain boundaries.

J. J. Kanter.[18] It is evident from the paper that the interpretation of flow tests of metals is associated with certain practical difficulties. A relationship involving time, temperature, stress, and deformation is obviously one presenting numerous aspects when applied to engineering problems. In order to obtain convenient summaries of flow test data, it is apparent that certain well-chosen units and constants pertaining to long-time tests eventually will be generally accepted. The nature of these constants will depend to a great extent upon the assumptions about flow characteristics that can be widely agreed upon. Up to the present time nearly every investigator has presented his data from a slightly different viewpoint, and as a result there are no broadly understood flow-test constants.

The authors have adopted several units for expressing the stress-deformation relationships that they have recorded. At temperatures where strain-hardening effects preponderate they report percentages of total deformation. At higher temperatures, where creep is more easily observed to be continuous, they use stress values yielding various percentages of flow per 1000 hours. The use of units involving only 1000 hours are best justified, perhaps, by the fact that little extrapolation is implied; i.e., the time periods covered in the actual testing in a number of cases are 1000 or more hours.

Reports of long-time tensile tests by other investigators give flow-stress values in terms of time periods up to 100,000 hours. Such values obviously are based upon extrapolations. Their physical significance may be somewhat open to question.

A number of tests, however, conducted for periods in the neighborhood of 10,000 hours have come from various sources. Such tests show that, at least for 10,000 hours, certain of the materials do not exceed the rates of creep established in the first several hundred hours of testing. For the materials for which this fact has been thoroughly substantiated, we perhaps are justified in computing flow-test constants in terms of total deformations for a 10,000-hour period.

Considering the temperature range for steels as consisting of two distinct parts—a lower division for which initial elongation is expressed and an upper characterized by secondary elongation—is quite descriptive of the relative plastic differences in metals that exist between low and high temperatures, but a certain difficulty that requires some emphasis presents itself. As metals are heated through the range of extreme strain-hardening to temperatures at which they are comparatively plastic, the transition takes place gradually. It is difficult to say what the limiting temperature is at which flow rates subsequent to the initial period cease to be of negligible consequence. This is a

[16] Manager, Alloy Division, Ohio Steel Foundry Company, Springfield, Ohio.

[17] Assistant to President, Midvale Company, Philadelphia, Pa. Mem. A.S.M.E.

[18] Crane Company, Chicago, Ill.

problem of considerable importance because so many applications of steels are being made at temperatures which are situated in the middle of this doubtful range.

To illustrate that *initial flow* and *secondary flow* sometimes must be considered as relative terms only, a variable temperature flow test for low-carbon open-hearth steel is shown. In this test, while the stress was held constant at 1000 lb. per sq. in., the temperature was raised periodically, in steps of 50 deg. fahr. (28 deg. cent.) from 550 deg. fahr. (290 deg. cent.) to 950 deg. fahr. (510 deg. cent.), the entire test requiring 58 days (1392 hours). Studying the time-deformation curve obtained, it is found that at each of the temperatures considered more or less deceleration of flow takes place as the testing proceeds. (Fig. 53.)

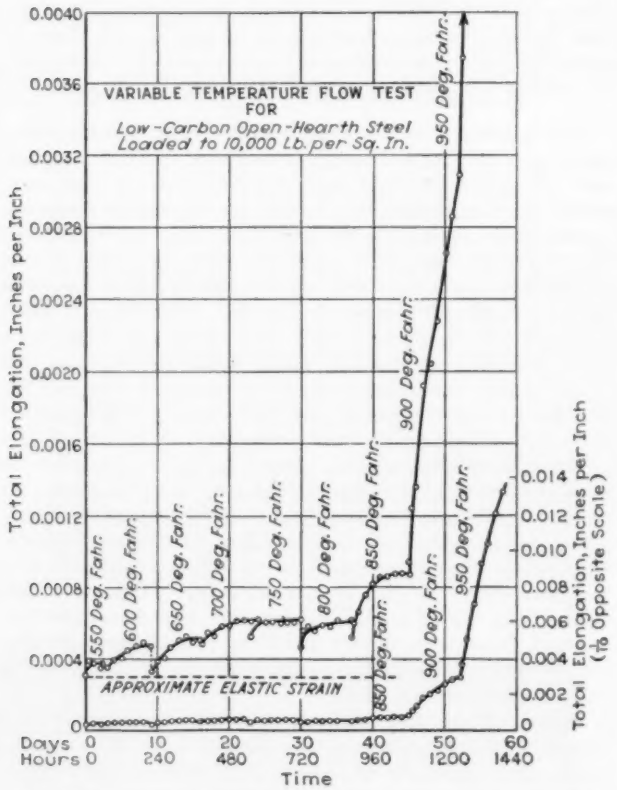


FIG. 53 VARIABLE TEMPERATURE FLOW TEST FOR LOW-CARBON OPEN-HEARTH STEEL

It does not seem entirely necessary to set up separate stress constants for expressing so-called initial and secondary flow. Inasmuch as flow or creep so seldom occurs at an ideally constant rate in the temperature ranges where we must recognize creep, flow-stress constants might be somewhat simplified if all were expressed in terms of total deformation for given periods of time. The troublesome decision between values in the highly important transition ranges would then be avoided.

R. L. PEEK.[19] The writer is wondering if, had the rate of creep been computed in the way described for a constant stress intensity, it would not be found that the rate of creep is either constant or decreasing, except perhaps in the immediate vicinity of the point of rupture. The point of this is that it seems likely that an increasing rate of flow under constant stress intensity indicates some failure of the material, and hence incipient rupture. If this is true, stress intensities producing such increasing rates are obviously unsafe, and if for lower stress intensities the rate decreases with elongation (a conclusion that the writer has suggested as following from the data), it may be possible to use the distinction between the two ranges of stress intensities corresponding to these two conditions as a criterion to determine the safe limits of loading.

ARTHUR MCCUTCHAN.[20] The tests reported in this paper tend to show that the strongest casting alloys for 1000 deg. fahr. service are those with little or no iron, 50 to 80 per cent nickel, and 50 to 20 per cent chromium. The authors suggest the addition of tungsten to improve the ductility of castings made of the composition.

These metals are relatively expensive, and mention is made of difficulty in fabrication. The 18–8 composition while found to have lower load-carrying ability at 1000 deg. fahr. can be formed either hot or cold, although with some difficulty as judged by the price demanded for tubing of this composition. The question arises as to whether the load-carrying ability of the high-nickel alloy at relatively high temperatures such as 1000 deg. fahr. is enough better to warrant its use in preference to 18–8 alloy. This has reference to superheater headers and tubes, piping, valve bodies, etc., operating at 1000 deg. fahr. Specifically, if tubing of 18–8 composition KA2S costs 8 to 10 times as much per pound as carbon steel, should not this alloy of superior load-carrying ability cost commemsurately more?

It is desirable to have data as to the relative sulphur attack on this alloy as compared with that on ordinary KA2S or similar composition having 3 to 4 per cent silicon. The latter material has been reported as having a high resistance to corrosion from hot sulphur dioxide, which might be present, for example, in the flue gases of a high-temperature superheater. In general, the high nickel-chromoum alloy described corresponds to Nichrome which shows good resistance to SO_2 attack.[21]

The question of the advantage or disadvantage of using very low carbon 18–8 alloy to avoid possible embrittlement due to carbide precipitation, as against the better load-carrying ability of high-carbon contents, is one that is very perplexing to the consumer at present. Especially as some difficulty seems to exist in producing the 18–8 alloy with carbon under 0.07 per cent.

The value given for stress to produce 0.1 per cent elongation in 1000 hours with 18–8 material (designated as E1912) chances to agree with that found by Professor Norton at 1100 deg. fahr., although considerably lower at 1000 deg. fahr. As to the different shape of the curve for 0.1 per cent elongation noted in several figures from the general form of stress-temperature flow curves, is it not possible that the shape of these curves might be considerably altered if tests had been made at intervening temperatures?

In many instances a creep rate one-tenth of the lowest reported in this paper has been made the basis for comparing the resistance to creep of metals at elevated temperatures. It is wondered if such extrapolation of test results is considered unwarranted in view of the methods of measuring deformation now employed.

---

[19] Bell Telephone Laboratories, New York, N. Y.

[20] Detroit Edison Company, Detroit, Mich.

[21] See paper on "Heat Resisting Steels," by Dr. W. H. Hatfield, presented before the Iron and Steel Institute, May, 1927.

FSP-53-10

Experiences in Chlorinating Condenser Circulating Water

By VINCENT M. FROST,[1] NEWARK, N. J., AND W. F. RIPPE,[2] KEARNY, N. J.

This paper gives the experiences of the authors over a period of two years in an endeavor to better the heat transfer and reduce condenser cleaning costs at the Kearny, N. J., power station of the Public Service Electric & Gas Company. The condensing apparatus there consists of five 50,000-sq. ft. condensers of the two-pass type, each having 10,000 one-inch tubes 19 ft. 2½ in. long. Chlorine control apparatus was installed, and the paper gives details regarding its operations, and includes water analyses, as well as data on condenser tests, amount of chlorine required, etc. The advantages and disadvantages of chlorination are set forth, together with the reasons for adopting it for the station in question.

IN AN ENDEAVOR to better the heat transfer and reduce the cost of cleaning condensers in the power stations of the Public Service Electric & Gas Company, it was decided to experiment with the use of chlorine gas in treating the circulating water. A cooperative arrangement was made with the Wallace & Tiernan Company, specialists in the application of chlorine control apparatus, and it was decided to try out the method at Kearny Station. This paper is intended to describe the results of our experiences to date along these lines.



V. M. FROST

Kearny Station is located in New Jersey on the west bank of the Hackensack River about 1½ miles from Newark Bay. Slime troubles seriously affected the vacua obtainable in the condensers, and it was found necessary to clean the condensers with rubber plugs at least once a week in the warm weather and twice a month in midwinter. Various methods of brushing and boiling the tubes are also used to improve the heat transfer. These methods, though not entirely satisfactory, were the best available. The character of the river water is shown by Table 1, which represents analyses of water samples covering a full day in the summer. These samples were taken every two hours. The analyses were made by the Department of Sewage Disposal, College of Agriculture, New Brunswick, N. J.

The average of some of the pertinent items is herewith given:

Total solids, p.p.m.	19,410
Suspended solids, p.p.m.	52
pH	7.2
Chlorides, p.p.m.	9,550
Common salt, p.p.m.	15,760
Dissolved oxygen, p.p.m.	1.0

It can be seen from the low oxygen content that the water is very polluted. The salt content indicates about one-half sea water.



W. F. RIPPE

The condensing apparatus at Kearny consists of five 50,000-sq. ft. condensers. They are of the two-pass type, with vertically divided water boxes, and each one has 10,000 one-inch tubes of No. 16 gage Admiralty mixture 19 ft. 2½ in. long. Each condenser is provided with two single-speed circulating pumps of approximately 45,000 g.p.m. capacity, so connected that one pump can supply either half or all of its own condenser or of any other condenser in the station. During normal operation with relatively cold water each condenser is operated individually with one of its own pumps.

The arrangement of the canals, pumps, and screens is shown in Fig. 1. Each condenser has its own intake canal which extends from the face of the dock, passing through the foundations of the bunker and boiler house to the suction chamber located beneath the turbine foundations. The canals are of rectangular cross-section, 9 ft. 6 in. deep and 6 ft. 6 in. wide. Each canal is provided with revolving screens located in the bunker-house basement, as shown on section *C-C* of Fig. 1. The screens are about 200 ft. from their respective suction chambers. The velocity of the water in the canal is 1.6 ft. per sec. with one pump in service, and 2.9 ft. per sec. with two pumps in service.

It was decided to try chlorinating one unit, and No. 4 unit was chosen arbitrarily. This unit, which is the same as No. 2 unit in every respect, consists of a 47,250-kw. Westinghouse turbo-generator unit and a Westinghouse radial-flow condenser equipped with two steam-jet air pumps and one hydraulic air pump. Since the two units are identical in every respect and operate under practically the same load conditions, this gives a fair means of comparison.

After deciding on what unit to install the equipment, the next question to come up was where to put the equipment. The choice lay between injecting the chlorine in the canal at the screen or in the suction pipe of the pumps. It was finally decided to install the equipment in the screen room and inject the chlorine solution immediately behind the screens for the following reasons:

1 The chlorine apparatus should be kept out of the heart of the plant, as it is a hazard.

[1] Asst. to General Superintendent of Generation, Public Service Electric & Gas Co. Mem. A.S.M.E. Mr. Frost was graduated from the Lawrence Scientific School of Harvard University, receiving his degree in Mechanical Engineering in 1902. Following his graduation, he was employed by Westinghouse, Church, Kerr & Co. and the Westinghouse Machine Company on steam-turbine and condenser work, including erection, operation, and repair. He served as assistant engineer and watch engineer at the Williamsburg Power Station of the Transit Development Company, Brooklyn, N. Y., and as chief engineer of the Perth Amboy Plant of the American Smelting and Refining Company. Since 1913, he has been with Public Service Electric and Gas Company.

[2] Test Engineer, Kearny Station, Public Service Electric & Gas Co. Jun. A.S.M.E. Mr. Rippe was born in New York City, on October 19, 1900. He entered Sibley College of Mechanical Engineering at Cornell University in 1918 and was graduated in 1922. Upon graduation he entered the employ of the Public Service Electric and Gas Company as a cadet engineer, completing the course in 1925, and has been with this company since that time.

Presented at a Meeting of the Metropolitan Section of the A.S.M.E., New York, November 21, 1929.

TABLE 1 ANALYSIS OF HACKENSACK RIVER WATER—KEARNY STATION, SEPTEMBER 17, 1929
(See Appendix for description of analyses.)

Sample No.	Tide elev., ft.	Direction of tide	Solids, p.p.m. Total	Solids, p.p.m. Suspended	pH	Acidity, p.p.m. $CaCO_3$	Alkalinity, p.p.m. $CaCO_3$	Chlorides, p.p.m.	Common salt, p.p.m.	Dissolved oxygen, p.p.m.	Biochemical oxygen demand, p.p.m. Straight	Biochemical oxygen demand, p.p.m. Dilution 50%	Biochemical oxygen demand, p.p.m. Dilution 25%
1	93.8	High	20,310	25	7.2	10.0	114.0	10,850	17,890	1.00	..	7.9	16.8
2	93.3	Ebb	21,530	14	7.3	11.5	114.0	9,250	15,250	1.00	..	6.0	2.8
3	91.6	Ebb	19,940	27	7.2	13.0	118.0	10,200	16,820	0.95	..	1.7	14.8
4	89.8	Ebb	17,930	39	7.2	13.0	119.0	8,800	14,510	0.90	..	5.7	2.7
5	89.4	Flood	16,700	175[a]	7.2	11.0	129.5	8,500	14,010	1.20	0.70	6.3	11.4
6	92.0	Flood	18,530	68	7.2	11.5	125.0	9,300	15,330	1.10	0.50	4.5	5.7
7	94.2	Flood	21,130	72	7.2	12.0	123.5	10,400	17,150	0.30	..	1.7	4.7
8	93.7	Ebb	22,490	131[a]	7.2	12.0	120.5	10,400	17,150	1.55	0.30	0.6	3.8
9	91.7	Ebb	20,180	74	7.2	11.5	119.0	10,250	16,900	1.10	..	2.9	4.7
10	89.9	Ebb	18,450	65	7.1	15.5	128.5	9,250	15,250	0.55	..	2.7	2.4
11	89.0	Low	17,320	320[a]	6.9	17.0	120.5	8,170	13,470	1.25	..	8.1	12.0
12	91.5	Flood	18,410	88	7.1	11.0	128.0	9,250	15,250	1.15	..	6.8	6.6
Avg.	91.7	...	19,410	52	7.2	12.5	122.0	9,552	15,760	1.00	0.12	4.6	7.4

[a] See Appendix, under "Suspended Solids."

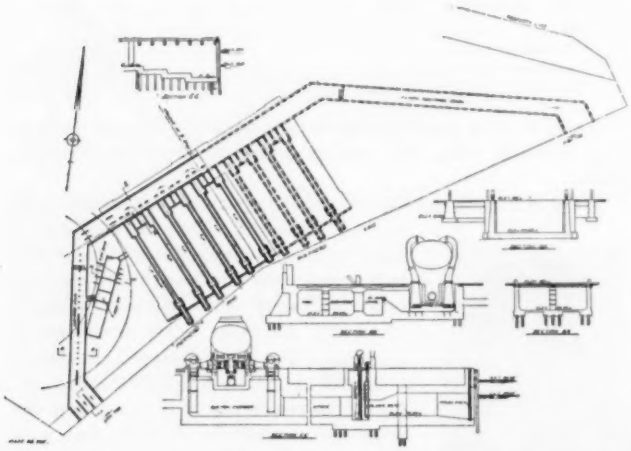


FIG. 1 ARRANGEMENT OF CANALS, PUMPS, AND SCREENS, KEARNY POWER STATION

2 Due to the comparatively small suction chamber and the isolation of units there would be no appreciable dilution.

With one pump in service, it takes $2^1/_2$ min. for the water to travel from the screen through the pump to the condenser, while with two pumps in service it requires $1^1/_2$ min. The velocity of the water through the condenser tubes is 4.8 ft. per sec. with one pump in service and 8.6 ft. per sec. with two.

This investigation is based on the theory that slime fouling of surface condensers is the result of definite biological growth and is dependent upon conditions of temperature and the presence of normal water organisms in the circulating supply. Slime is a conglomerate of inorganic matter, namely, mud and silt bound together by algae growths of a jelly-like nature. If the chlorine dosage destroys the algae and prevents growth, there will be nothing to hold the inorganic matter together and slime will not form. On this basis it was believed that intermittent dosage at correct intervals would prevent the growth and reduce the total amount of chlorine to be used. Patents covering this intermittent process have been granted to Dr. J. Baker, of the Wallace & Tiernan Company.

The equipment installed in No. 4 screen room is shown in Fig. 2 and consists of three chlorinators, each capable of handling 720 lb. of chlorine per day with a battery of 150-lb. chlorine tanks, a program clock, and the necessary piping. The operation of this equipment is as follows: The clock trips a relay which operates a valve in the water line supplying water to the injectors on the chlorinators. The discharge of chlorine solution is carried from each machine down into the canal by means of hose. The program clock can be so set as to give any desired frequency and length of dosage. The amounts of chlorine fed and the frequencies are shown in Fig. 3.

Figs. 4 and 5 give the story of the investigation. The data plotted are weekly averages of hourly readings taken over the two-year period, with corrections for slight load variations. The air leakages of both units during the period under discussion were about the same, so this item has been neglected.

Unit No. 4 was taken out of service in January, 1928, for general overhauling, and during this period was given a thorough cleaning so that when it went back into service the following April it showed a better heat transfer than No. 2. It gradually became worse, however, and by August was about the same as No. 2. The big difference between the two units in June, 1928, was caused by tests being run on No. 4 at an intentionally reduced vacuum.

Chlorination of No. 4 was started on August 10, 1928, after both No. 4 and No. 2 had been thoroughly cleaned by baking

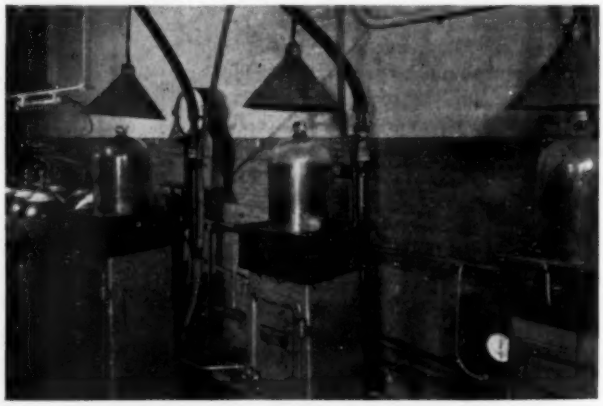


FIG. 2 EQUIPMENT INSTALLED IN NO. 4 SCREEN ROOM

and rubber plugs. After chlorination started, the increase in vacuum of No. 4 over No. 2 was very marked, and frequent examinations showed the absence of slime in the tubes. This gain amounted to 0.2 in. in the middle of October, 1928, but some of this gain may be ascribed to better original cleaning of No. 4 than of 2, although an endeavor was made to get both condensers in the same condition. It was not expected that there would be such a sudden gain in heat transfer, which is the reason for the foregoing statement. During this period No. 2 slimed badly as usual, and even the weekly use of rubber plugs could not hold it near No. 4 in performance.

The dosage chart, Fig. 3, shows that the amount of chlorine used per day varies from 320 lb. to 72 lb. Due to lack of data on the intermittent method of treatment, the dosage was purely experimental. The reduction in the use of chlorine was effected by reducing the percentage of time the chlorine was being injected rather than the concentrations. Whereas at the start

the chlorine was being injected for 2 min. every 10 min., by the middle of October it was found that injection for 3 min. every hour was sufficient to keep the condenser free of slime. It must be borne in mind that it was desired to keep the chlorine dosage sufficiently high to prevent the least indication of slime, and for that reason some of the dosages may be excessive.

In October, 1928, since it was apparent that the slime could be prevented, it was decided to investigate the possibilities of cleaning a very dirty condenser with chlorine. Chlorination of No. 4 was therefore stopped on October 19, 1928, and the condenser was allowed to become dirty. Chlorination was again started on November 26, 1928, but for a while there was no apparent benefit and No. 4 became worse than No. 2, as shown in the first week of December, 1928. It is believed that the chlorine would eventually have brought No. 4 back to its original advantageous condition, but owing to its heavy accumulation of slime this would have been very slow work. It was decided therefore to clean the tubes of both condensers as thoroughly as possible and start another comparative run.

An examination of the tubes of No. 4 condenser at this time showed a hard scale, which it was not expected that the chlorine would remove. Therefore on December 15, 1928, this condenser was boiled on the water side with chemicals, making it as clean as it was conceived to be possible. This resulted in increasing the coefficient of heat transfer from 185 to 270, and brought the vacuum to a point 0.35 in. Hg better than that in No. 2. Immediately after the boiling, chlorination was continued with the hope of keeping the condenser as near as possible to its condition immediately after the boiling. The advantage was increased to 0.4 in. Hg in the latter part of January, 1929, when it was decided to put No. 2 on a par with No. 4 by boiling it also. This was done and there was then very little difference between it and No. 4. On February 1, 1929, unit No. 2 was taken out of service for its annual overhauling, but the chlorination of No. 4 was continued with the idea that when No. 2 was put back into service the two units would start on an even basis. On April

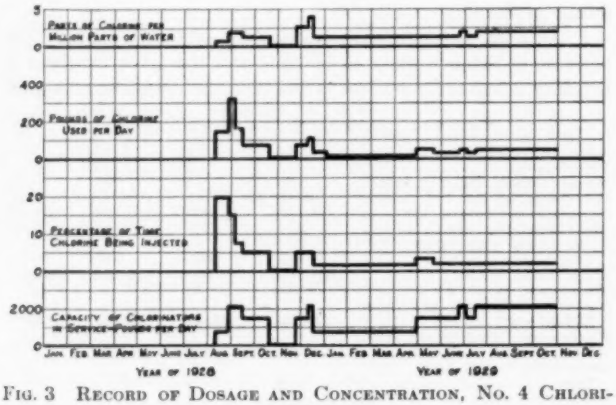


FIG. 3 RECORD OF DOSAGE AND CONCENTRATION, NO. 4 CHLORINATOR

10, 1929, No. 2 was put back in service with a very clean condenser and practically no difference in vacuum between it and No. 4.

Upon referring to Fig. 3 it will be seen that from January 1 to May 1, 1929, it was possible to keep the condenser free of slime with only 12 lb. of chlorine per day. This was done by operating one chlorinator for three minutes every hour for an eight-hour period each day. Here again the concentration of chlorine was maintained, and the reduction in the amount used was effected by cutting down on the percentage of time that the chlorinator was in service.

The dosage was increased on May 1 when the water temperature reached 50 deg. fahr., as the growth of algae increases rapidly above this temperature. It is questionable whether or not this method of treating the water for an eight-hour period of the entire day is the most efficient, but suffice it to say that in this case it was effective. The quantity of chlorine used was

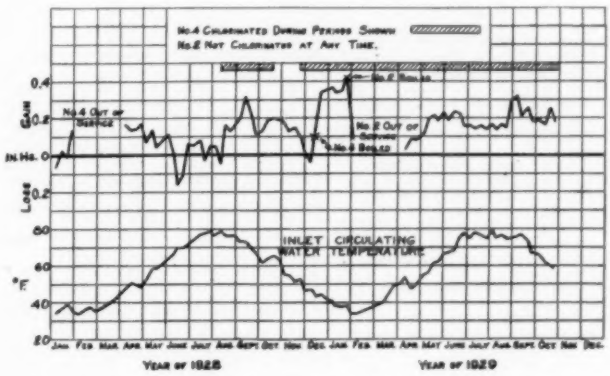


FIG. 4 GAIN IN VACUUM OF NO. 4 CONDENSER OVER NO. 2 CONDENSER

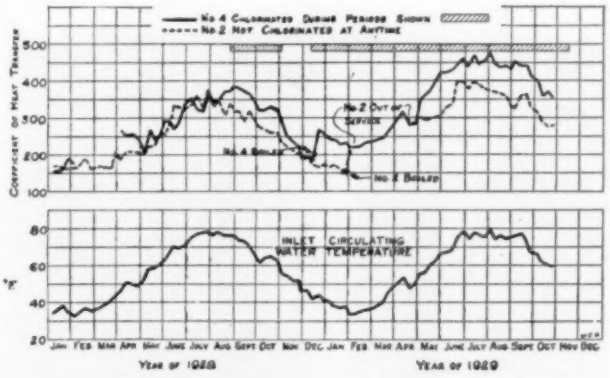


FIG. 5 HEAT TRANSFER OF NO. 2 AND NO. 4 CONDENSERS

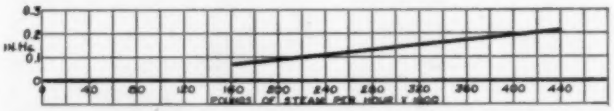


FIG. 6 INCREASE OF VACUUM OF NO. 4 CONDENSER AFTER HAVING BEEN CHLORINATED FOR ONE YEAR

so small that any reduction would be immaterial in this particular instance.

At this point, the most significant part of the entire investigation is presented. The two condensers started out on an even basis, but in a month No. 4 was 0.2 in. better than No. 2, even though No. 2 was being cleaned regularly with rubber plugs. A gain in vacuum of approximately 0.2 in. was maintained. The big increase on September 1, 1928, was caused by No. 2 not being cleaned with rubber plugs for three weeks, but as soon as this was done it was brought down to the 0.2-in. difference again. Comparing No. 4 with other units during the period that No. 2 was out of service shows a gain of about 0.15 in.

Condenser performance tests were made on No. 4 condenser with weighed condensate in June, 1928, and July, 1929, the data of which show the gain in heat transfer due to chlorination. These data are given in Tables 2 and 3. The higher flow of circulating water in the 1929 tests is accounted for by higher tides, which cut down on the suction lift of the pumps and thereby increased

TABLE 2 TEST OF NO. 4 CONDENSER

Date, 1928	6/26	6/26	6/25	6/25
Duration, hr	3	3	3	3
Room temperature, deg. fahr	87	83	82	87
Barometer, in. Hg	29.88	29.88	29.82	29.83
Generator net output, kw	21,840	28,080	38,500	41,900
Main steam pressure, lb. gage	352	356	353	352
Main steam temperature, deg. fahr	715	713	712	711
Total steam, lb. per hr	231,000	286,480	379,490	420,310
Water rate, lb. per kw-hr	10.58	10.20	9.86	10.03
Vacuum at neck of condenser (30 in. bar.), in. Hg	28.68	28.60	28.41	28.21
Condensate temperature, deg. fahr	86	88	91	95
Circulating-water inlet, deg. fahr	71.7	69.9	68.3	70.4
Circulating-water outlet, deg. fahr	76.7	76.1	76.6	78.7
Circulating-water rise, deg. fahr	5.0	6.2	8.3	8.3
Circulating-water flow, g.p.m	91,200	88,900	86,300	95,860
Velocity of water in tubes, ft. per sec	9.8	9.5	9.2	10.3
Heat transfer, B.t.u. per sq. ft. per hr	4490	5480	7140	7960
Mean temperature diff. (log.), deg. fahr	13.2	16.4	21.0	22.7
Mean temperature diff. (arith.), deg. fahr	13.4	16.5	21.4	22.9
Coefficient of heat transfer:				
B.t.u. per sq. ft. per hr. per deg. fahr. (log.)	340	334	340	351
B.t.u. per sq. ft. per hr. per deg. fahr. (arith.)	335	332	334	348
Vacuum at air pump, in. Hg	29.03	29.01	28.93	28.92
Air leakage, cu. ft. per min	12.0	8.3	6.2	8.2

REMARKS: 1. Turbine operated non-bleeding. 2. Two circulating pumps in service. 3. Air leakage measured by nozzle in air-pump discharge.

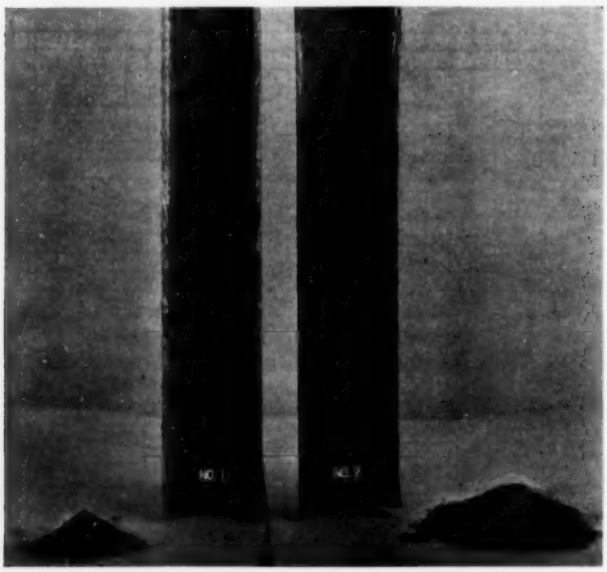


FIG. 7 CONDITION OF TUBES FROM CHLORINATED (NO. 1) AND UNCHLORINATED (NO. 2) CONDENSERS

the flow. In order to have a fair means of comparison the tests were corrected to the same water flow and temperature; the difference is shown in Fig. 6. The average load of this unit gives about 300,000 lb. of exhaust steam per hour, which corresponds to a gain of 0.15 in. Hg of vacuum.

Fig. 7 shows the condition of two tubes removed from No. 2 and No. 4 condensers in August, 1929, after No. 4 had been chlorinated for one year. The piles at the bottom show the relative amounts of deposits on a four-foot length. The picture tells its own story.

One of the most important drawbacks in chlorinating circulating water is the fact that it is a hazard. Chlorine gas, although not poisonous, has a choking effect and will drive men out of the plant if a large enough quantity is allowed to escape. The equipment shown in Fig. 2 gave considerable trouble, especially in the early stages, due to leaks and various mechanical difficulties. Because of the nature of the gas, the policy has been to provide that all future installations shall be outside the main building, for if a tank of chlorine were to be discharged in the main building it would drive every one into the open and cause a shutdown.

With this in mind, when it was decided to extend the experiment, the layout shown in Fig. 8 was installed on unit No. 3 in July, 1929. The point of injection of the chlorine solution is the same as on No. 4, i.e., immediately behind the screens, but the essential differences are as follows:

1 All equipment is installed in a separate building outside

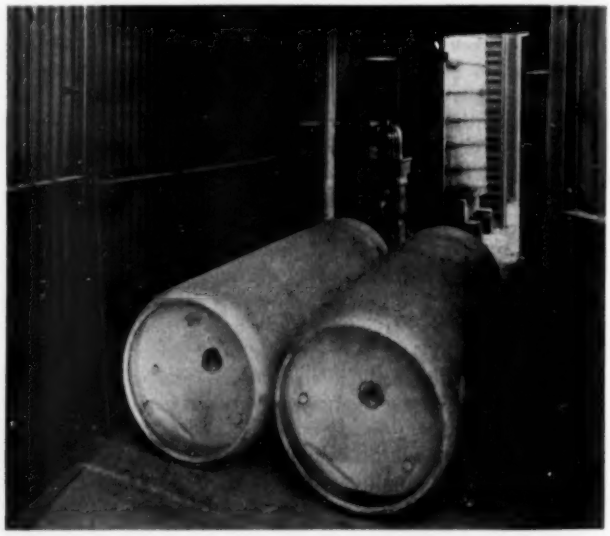


FIG. 8 INSTALLATION OF ONE-TON CHLORINE CONTAINERS IN BUILDING OUTSIDE MAIN STRUCTURE

the main structure. This lessens the hazard in case of leaks or failure of the apparatus.

2 One-ton containers are used instead of 150-lb. tanks. The chlorine is cheaper when so bought, and the labor involved in handling a one-ton container is less than the equivalent 13 small

TABLE 3 TEST OF NO. 4 CONDENSER

Date, 1929	7/16	7/17	7/15	7/11
Duration, hr	3	2	3	3
Room temperature, deg. fahr	80	78	81	87
Barometer, in. Hg	30.18	30.15	30.32	30.40
Generator net output, kw	14,550	23,060	30,780	39,320
Main steam pressure, lb. gage	362	355	358	352
Main steam temperature, deg. fahr	686	686	701	713
Total steam, lb. per hr	166,340	244,100	316,120	397,930
Water rate, lb. per kw-hr	11.43	10.58	10.27	10.12
Vacuum at neck of condenser (30 in. bar.), in. Hg	28.66	28.51	28.28	27.99
Condensate temperature, deg. fahr	86	89	93	98
Circulating-water inlet, deg. fahr	76.8	77.4	77.9	79.8
Circulating-water outlet, deg. fahr	81.0	83.1	85.4	89.0
Circulating-water rise, deg. fahr	4.2	5.7	7.5	9.2
Circulating-water flow, g.p.m	77,600	81,100	79,800	81,800
Velocity of water in tubes, ft. per sec	8.3	8.7	8.5	8.8
Heat transfer, B.t.u. per sq. ft. per hr	3270	4650	5980	7500
Mean temperature diff. (log.), deg. fahr	9.3	10.9	14.2	16.5
Mean temperature diff. (arith.), deg. fahr	9.3	11.3	14.6	17.0
Coefficient of heat transfer				
B.t.u. per sq. ft. per hr. per deg. fahr. (log.)	350	425	420	454
B.t.u. per sq. ft. per hr. per deg. fahr. (arith.)	350	412	410	441
Vacuum at air pump, in. Hg	28.81	28.74	28.63	28.40
Air leakage, cu. ft. per min	14.6	10.3	9.6	8.7

REMARKS: 1. Turbine operated non-bleeding. 2. Two circulating pumps in service. 3. Air leakage measured by nozzle in air-pump discharge.

tanks. The fewer connections with the one-ton containers lessens the possibility of leaks and reduces the labor of connecting up tanks.

3 The chlorinator injects the solution continuously into the tower filled with limestone, and the pump operates intermittently, taking solution from the tower and pumping it into the canal. Due to the continuous operation of the chlorinator less capacity is required, and the chlorinators function better when in continuous service. The disadvantage is the extra pump and tower necessary for this particular feature.

At the time the new equipment was ready for operation, the condenser was very dirty and an effort was made first to determine whether or not the chlorine would clean it. Apparently it would eventually clean it of all slime, but the ideal method seems to be to start with as clean a condenser as possible, with the expectation that the chlorine will maintain that condition.

The question of corrosion of condenser water boxes and tubes has been brought up. There has been no sign of corrosion of No. 4 condenser after being chlorinated for a period of over a year. The amount of chlorine in the water is so small and the time it is in the water is so short that it is not believed any serious trouble will arise from this source. Repeated tests have usually shown no residual chlorine in the water entering the condenser, and none at any time leaving the condenser during dosing periods. The maximum residual chlorine found at any time at the inlet to the condenser was 0.3 part per million.

There is no doubt of the ability to keep the condensers at Kearny free of slime by the use of chlorine. Since No. 4 was boiled out in December, 1928, no cleaning has been done except for the chlorine treatment and the removal of loose matter in the tube ends. This fact alone proves the treatment's effectiveness. That there is a saving due to the better vacuum maintained than when cleaned only with rubber plugs, has also been shown. The estimated average gain in vacuum due to chlorination on a unit at Kearny operating at a 75 per cent load factor is 0.15 in. Hg, amounting to almost 1 per cent gain in steam consumption.

The amount of chlorine required to chlorinate a condenser at Kearny is based on the following data:

River-water temperature, deg. fahr.	Pounds per day	Number of days	Pounds of chlorine for period
32–50	40	121	4,840
50–70	60	152	9,120
70–80	75	92	6,900
Total		365	20,860

The advantages obtained by chlorination are:

1 Gain in vacuum
2 Elimination of rubber-plug cleaning
3 Elimination of boiling of condensers
4 Elimination of necessity of taking a condenser out of service to clean it, thus increasing the service availability of the unit. This also eliminates the extra cost of steam used for starting turbines.

The disadvantages of chlorination are:

1 Cost of chlorine
2 Maintenance of chlorinating equipment
3 Fixed charges on additional investment
4 Hazard.

These items for and against chlorination have been weighed for the conditions obtaining at Kearny Station, and it seems advantageous to install the system. In different plants the various items will vary in importance due to differences in character and temperature of water, loading conditions, etc.

Acknowledgment is due Dr. J. Baker and W. J. O'Connell for their cooperation and assistance in the work described.

Appendix

EXPLANATION OF WATER ANALYSES

Total Solids. Two analyses were made on each sample. The high results are due to the fact that the samples were sea water. The total solids then, for the most part, represent soluble material.

Suspended Solids. Three analyses, two for check, were run on each sample. Those marked (*a*) were evidently high because of the error in sampling, i.e., dragging of the "thief."

pH. Each sample was checked by two different indicators.

Acidity. This was determined by using phenolphthalein as an indicator. Therefore the acidity is due to CO_2 or mineral acids, or to both.

Alkalinity. This was determined by using methyl orange as an indicator. As there is no phenolphthalein alkalinity, the alkalinity in this test represents bicarbonates *only*.

Chlorides. Three analyses were run on each sample.

B.O.D. (Biochemical Oxygen Demand.) These results are liable to discrepancy due to the samples being sea water. The B.O.D. was a five-day test. The straight samples (no dilution) after incubation at 20 deg. cent. for five days showed no decrease in D.O. except in the three cases shown. The other dilutions, 50 per cent (50 per cent dilution water and 50 per cent sample) and 25 per cent (75 per cent dilution water and 25 per cent sample), also show a variety of results. This can be accounted for by the presence of chlorides, and more likely of nitrates, of which latter there may be a large quantity.

Discussion

BERT HOUGHTON.[3] Certainly every man operating condensers with harbor water will appreciate the difficulty of cleaning them by mechanical means. If the chemical method described will produce the desired result without attacking the condenser metal, it will indeed be welcome.

However, the writer is reminded of a similar experiment tried several years ago at the Hudson Avenue Station. In that case it was not condenser circulating water, but well water which was circulated through the generator air coolers, and it was about 15 per cent salt water. These air coolers became heavily coated with slime after a few months of operation. The water was analyzed bacteriologically and found to contain algae. The recognized method for the prevention of algae growth, the writer was advised, was chlorine treatment. The system at Hudson Avenue was such that the chlorine could be introduced at the discharge of the pumps, the treated water flowing into a large storage tank, in which there was considerable turbulence. It was felt that the conditions were particularly advantageous for thoroughly mixing chlorine with the water. An average of three pounds of chlorine per million pounds of water was used, the chlorine being fed continuously. This experiment was carried on for several weeks, but as the slime did not disappear, it was dropped.

The results obtained, however, coincide to some extent with those of the authors. They mention the fact that, starting with a dirty condenser, they were not able to eliminate algae growth. They have found, however, that starting with a clean condenser they can prevent algae growth. The interesting feature to

[3] Operating Superintendent, Brooklyn Edison Company, Brooklyn, N. Y. Mem. A.S.M.E.

the writer, in view of his experience, is that this growth could be prevented and the condensers kept clean with so small an amount of chlorine, of the order of 0.1 to 0.2 lb. per million pounds of circulating water. (The intermittent rate is higher—this refers to the average.)

The writer has made a comparison of tests on the condenser of

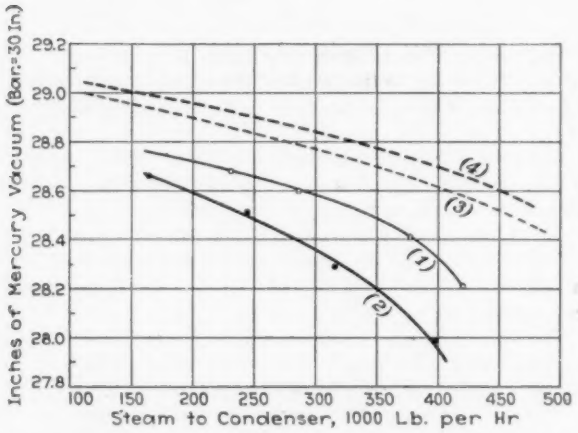


FIG. 9 COMPARISON OF CONDENSER PERFORMANCE AT KEARNY AND HUDSON AVENUE STATIONS

(1) Kearny No. 4, 1928: 50,000 sq. ft., 47,250 kw.; unchlorinated; circulating-water inlet temperature, 70 deg. fahr.; velocity, 9.7 ft. per sec.
(2) Kearny No. 4, 1929: 50,000 sq. ft.; 47,250 kw.; chlorinated; circulating-water inlet temperature, 78 deg. fahr.; velocity, 8.6 ft. per sec.
(3) Hudson Ave. No. 3: 70,000 sq. ft.; 50,000 kw.; old tubes, rubber plugged; circulating water temperature, 72 deg. fahr.; velocity, 8.4 ft. per sec.
(4) Hudson Ave. No. 2: 70,000 sq. ft.; 50,000 kw.; new tubes; circulating-water temperature, 72 deg. fahr.; velocity, 8.4 ft. per sec.

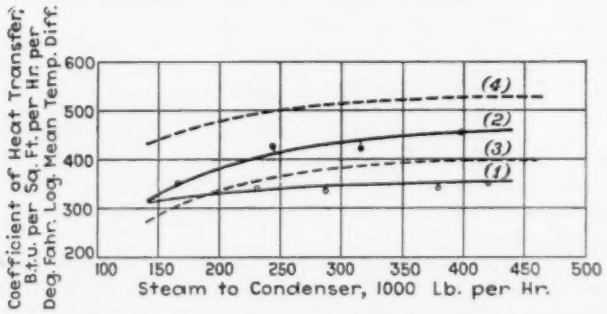


FIG. 10 COMPARISON OF CONDENSER PERFORMANCE AT KEARNY AND HUDSON AVENUE STATIONS

[(1), (2), (3), and (4): See notes under Fig. 9.]

unit No. 4, Kearny Station, with tests on units No. 2 and No. 3, Hudson Avenue Station. All three condensers are of the Westinghouse two-pass radial-flow type. Those at Hudson Avenue have a more liberal surface-to-kw.-capacity ratio, namely, 1.4 as against 1.06 for the Kearny No. 4 condenser. The two Hudson Avenue units are identical in design, and each has two circulating pumps with a combined capacity of 100,000 g.p.m. The Kearny No. 4 condenser has two circulating pumps with a capacity of 90,000 g.p.m.

Since the condensers in the two stations are similar in design, even though those at Hudson Avenue have surfaces of 70,000 sq. ft. each while Kearny No. 4 has only 50,000 sq. ft., a comparison can be made both directly and by applying adjustment factors to the results obtained in the two stations. There are two comparisons to be made, namely, No. 4 Kearny using chlorinated circulating water, with No. 2 Hudson Avenue with new tubes; and No. 4 Kearny without the use of chlorinated circulating water, with No. 3 Hudson Avenue after cleaning old tubes with rubber plugs. A few graphs will illustrate the comparisons.

The dottled lines on all graphs refer to Hudson Avenue units, the heavy ones being No. 2 with new tubes, the lighter ones, No. 3 with old tubes rubber plugged. The solid lines refer to Kearny Unit No. 4, the heavy ones after use of chlorinated circulating water. Data for the solid curves were taken from Tables 2 and 3 of the paper.

In Fig. 9 the curves are, in order of highest vacuum, Hudson Avenue No. 2 with new tubes; Hudson Avenue No. 3 with old tubes rubber plugged; Kearny No. 4 with unchlorinated circulating water; and Kearny No. 4 with chlorinated circulating water. The better performance of No. 2 and No. 3 Hudson Avenue is partly due to their having larger surfaces. This is particularly noticeable at high rates of steam flow; at 400,000 lb. per hr. there is a difference in vacuum of 0.3 in. of mercury

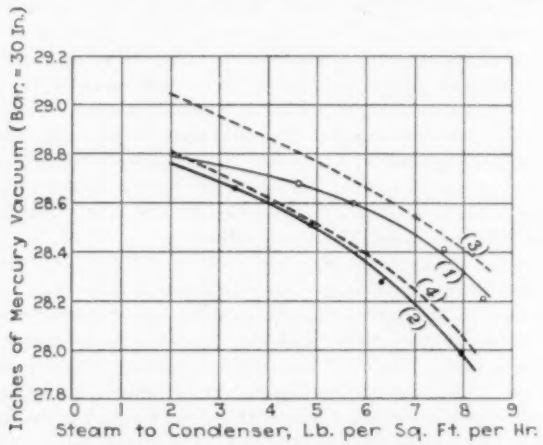


FIG. 11 COMPARISON OF CONDENSER PERFORMANCE AT KEARNY AND HUDSON AVENUE STATIONS

[(1), (2), (3), and (4) same as for Fig. 9, except that for (3), temperature is 70 deg. fahr. and for (4), is 78 deg. fahr.]

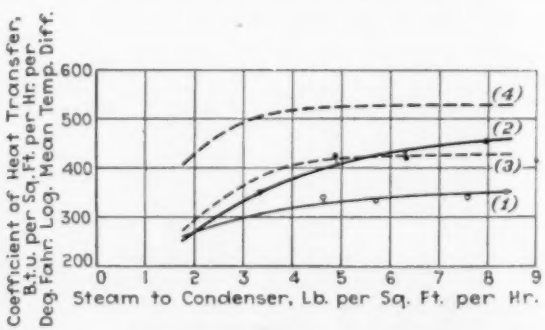


FIG. 12 COMPARISON OF CONDENSER PERFORMANCE AT KEARNY AND HUDSON AVENUE STATIONS

[(1), (2), (3), and (4) same as for Fig. 9, except that for (3) velocity is corrected to 9.7 ft. per sec., and for (4) to 8.6 ft. per sec.]

from the unchlorinated-water test, and 0.7 in. of mercury from the chlorinated-water test. The higher vacuum obtained with unchlorinated circulating water at Kearny is explained by the fact that the circulating-water-inlet temperature was 70 deg. fahr. as against 78 deg. fahr. for the chlorinated water.

Fig. 10 shows the heat-transfer coefficient of the condensers under the same conditions as in Fig. 9. Here the Hudson Avenue No. 2 with new tubes shows the highest coefficient, followed in order by Kearny No. 4 with chlorinated circulating water, Hudson Avenue No. 3 with old tubes rubber plugged, and Kearny No. 4 with unchlorinated water. The difference

between the two Kearny No. 4 curves is larger than it would be if the circulating-water temperatures had been the same.

In order to eliminate the effect of the sizes of the condensers, comparisons were made on the basis of pounds of steam per square foot of condenser surface per hour. The results of the Hudson Avenue tests were also corrected to conditions comparable to the tests on the Kearny Unit No. 4.

Thus Fig. 11 shows curves for Kearny No. 4 merely reduced to pounds per square foot per hour, while Hudson Avenue No. 2 has the test with the new tubes corrected to 78 deg. fahr. inlet temperature, which was that of the chlorinated water. Hudson Avenue No. 3 with old tubes rubber plugged was corrected to 70 deg. fahr. circulating-water temperature to correspond with the unchlorinated circulating water at Kearny. This graph shows the vacuum with the new tubes at Hudson Avenue to be only about 0.05 in. of mercury better than with chlorinated circulating water at Kearny. The chlorinated water is thus apparently very effective in improving the vacuum.

Hudson Avenue No. 3 with old tubes rubber plugged shows considerably improvement in vacuum over the Kearny No. 4 unit with unchlorinated water. There would actually be a larger difference, however, because the velocity at Hudson Avenue was only 8.4 ft. per sec. as against 9.7 ft. per sec. at Kearny. This shows that rubber plugging gives better results at Hudson Avenue than at Kearny.

Fig. 12 shows the heat-transfer coefficients under the corrected conditions. The comparison of Kearny No. 4, chlorinated water, with Hudson Avenue No. 2, new tubes, shows the latter to have the higher coefficient. Moreover, the Hudson Avenue curve is for 8.6 ft. per sec. velocity and 72 deg. fahr. temperature, while the Kearny unit is for 8.6 ft. per sec. velocity and 78 deg. fahr. inlet-water temperature. If these temperatures were the same, the difference between the curves would be increased, due to the factor of mean temperature difference increasing much faster than the additional heat transferred.

In the case of Hudson Avenue No. 3 and Kearny No. 4 with unchlorinated water the velocities are both 9.7 ft. per sec.; with the former the water-inlet temperature was 72 deg. fahr. while with the latter it was 70 deg. fahr.

The comparison would be more satisfactory if the Kearny tests had been made at more nearly the same conditions of circulating-water temperature and velocity. Nevertheless, the results of graph Fig. 11 indicate that further investigation is well warranted. A study of the effect of the chlorine on the life of the condenser metal would be of great interest.

M. H. Curley.[4] It would appear the authors have conclusively proved that condenser tubes can be kept clean from slime by use of chlorine, especially when such slime is a result of sewage pollution of the circulating water. The writer was particularly interested, as chlorine is being tried as a means of controlling mussel growth at the Northport Plant of the Long Island Lighting Company. He had an opportunity of examining the condenser under treatment in April, 1929, noting particularly the absence of slime on the interior of the tubes.

Bengough and May, of the Institute of Metals in London, England, are probably the world's foremost investigators of condensers and condenser tubes. In one of their reports issued early in 1925 under the sub-subject of "Water Treatment," they have this to say: "Slight chlorination of cooling water seems effective in killing or preventing the growth of organisms upon the tube wall, making it unnecessary to employ means of tube cleaning which promote corrosion." Apparently, while it has been known for some time that the use of chlorine would keep the tubes clean, no practical method of applying the chlorine treatment was available, and it is due to the development of practical apparatus by Wallace & Tiernan that full use of this treatment is made possible.

On first hearing about the experimental use of chlorine in the circulating system at Kearny, the writer was particularly interested, having in mind its use in control of mussel growth, as mussels in the circulating-water ducts of the company with which he is connected are far more serious than is the fouling of tubes. For the past two years, he has been investigating tube failures from inlet erosion-corrosion and has found that no trouble from this source exists where heavy sewage pollution is present. The sewage pollution, however, always results in bad slime on the tubes. The answer to an inlet erosion-corrosion, where a heavy coating of slime is present, is, no doubt, due to either or both of two reasons: namely, the slime coating protects the surface of the tubes against the action of the oxygen in solution in the circulating water; or the nitrates in the sewage consume the dissolved oxygen. When the oxygen content is low or nil, no inlet erosion takes place. It is also noticeable that no mussel growth is present where there is a heavy slime coating. Mussels of course cannot exist in water low in oxygen. In Table 1 of the paper, the circulating-water analyses show a very low oxygen content and it is of interest to know that there is no evidence of inlet erosion-corrosion on any of the tubes at the Kearny Station.

In view of the above, it would be well to point out a possible danger in the removal of slime from tubes by the use of chlorine, but not from the standpoint of any damage that may result from the chlorine itself.

In certain waters having possibly slight sewage pollution, but not sufficient to reduce to dissolved oxygen below the danger point, it is easily possible that considerable cleaning is required to remove the slime from the tubes. If the chlorinating system is applied, there is possible danger of cleaning the tubes so thoroughly that inlet erosion will be started. This danger can probably be avoided by using only enough chlorine to clean the tubes so as to leave a slight scum on the surface, this scum acting as protection against inlet erosion attack.

Near the beginning of the paper the authors give their reasons for injecting chlorine in the canal at the screen instead of at the suction pipe at the pump and later they bring up the question of the corrosion of condenser water boxes and tubes by the chlorine, also stating that the maximum residual chlorine found at any time at the inlet to the condenser was only 0.3 part per million. The fact that practically no residual chlorine actually entered the condenser eliminates in itself any question of danger of the corrosion of the condenser water boxes and tubes, and would also seem to indicate that it is most advisable to inject the chlorine in the stream of water going to the condensers as far from the inlet to the condenser as is practical.

The action of the chlorine is, apparently, to kill the algae growth, and this can be done in the stream of water on the way to the condenser as readily as it can be done after forming in the tubes.

The paper would have been much more conclusive had it given some figures of the savings effected in dollars and cents, comparing the cost of other methods of cleaning with the cost of chlorine treatment. Taking the figures given by the authors, it would appear that at the factor given and the size of the unit, the monthly output is 24,500,000 kw-hr. Assuming 10 lb. steam per kw-hr., this gives 245,000,000 lb. of steam. A betterment in the water rate of 1 per cent will show a saving on the above monthly output from this unit of approximately 2,500,000 lb. of steam. At 20 cents, the approximate fuel cost per 1000 lb. of steam shows a saving of $490 on the above monthly output. The summer condition of 75 lb. of chlorine per day, 30 days in a

[4] Long Island Lighting Co., Bay Shore, N. Y. Assoc-Mem. A.S.M.E.

month, gives 2250 lb. of chlorine, which, at 6 cents per lb. shows a cost of $135 for raw material. Assuming an investment of approximately $5000, at 16 per cent this gives about $66 per month for interest and depreciation charges, making a total cost of roughly $200 a month without any labor expense. This indicates a saving in use of the chlorine method of cleaning of approximately $300 per month over the method of cleaning by the use of rubber plugs. It would be of interest to know if the authors can give us the approximate monthly cost of cleaning condensers other than the one in which the chlorine is in use. This question of cleaning condensers is one about which the writer feels we know very little. The average plant plans to maintain an average vacuum and cleans whenever, in some one's opinion, it is necessary to do so to maintain the desired vacuum, irrespective of whether it is more costly to shut the unit down than it would be to maintain it in service over a longer period of time at the reduced efficiency due to the dirtier condition of the tubes. It is very rare to find the actual cost of cleaning condensers billed against the savings effected by such cleaning.

The Public Service Company and the Wallace & Tiernan Company are to be commended for taking the initiative in the use of chlorine for cleaning condenser tubes, and we should all be grateful to the authors for giving us the results of their work, which are bound to be very useful.

FSP-53-11

Electrical-Conductance Measurements of Water and Steam, and Applications in Steam Plants

By MAX HECHT[1] and D. S. McKINNEY,[2] PITTSBURGH, PA.

In this paper a conductance method is presented for determining the concentration of salts in steam-plant water supplies. The method is applied to determine the purity of steam, of supplies used for boiler-feed purposes, and the concentration of boiler and evaporator water. The conductance data are refined by applying both a water and a CO_2 correction. The method for establishing the CO_2 correction is based on the preparation of a standard "synthetic" water, and eliminates the errors accompanying the chemical analysis of "pure" water. The conductance meter used may be calibrated in terms of dissolved salts. The relation of dissolved salts to conductance is discussed. Simple commercial equipment is used for measuring conductance. The data are obtained rapidly by the steam-plant personnel, and the results may be applied easily to the control of water conditions.

OPERATORS appreciate the necessity of excluding any contamination from the steam and water cycle in steam power plants. Circulating water introduced via leaky surface condensers contaminates the water cycle. Erratic control of concentration of water in steam-generating apparatus and carry-over therefrom contaminate the steam cycle. The importance of entrainment rather than moisture should demand the critical attention of station operators. The entrainment of boiler-water salts in steam causes scaling and burn-outs in superheater tubes, and if the entrainment is carried into the turbine, blade deposits contribute to loss in turbine capacity and efficiency. The entrainment is also a factor in the deterioration of blade material because of erosion. In a review of a discussion of corrosion of iron, Whitney states that once corrosion of the metal of turbine blades is started, the water condensed in the steam passing through the turbine removes the corrosion products rapidly. The corrosion is influenced by the oxygen contained in the boiler feedwater. The deterioration is not noticeable in the initial turbine stages, which contact with dry and superheated steam, but is in evidence in the stages where the temperature of the steam is 100 deg. cent. (3)[3]. The possibility of one other factor which may influence the erosion of blade material is suggested: The rate of formation of water droplets may be accelerated in the lower pressure stages of the turbine because of the presence in the vapor of anhydrous dusts. The dusts are produced when the soluble salts and insoluble products discharged with the steam from the boiler drum are dehydrated in the superheater. It may be considered that the individual dust particles, which may be electrically charged, act as nuclei for the condensation of vapor surrounding them when favorable temperature conditions are reached in the lower-temperature stages of the turbine. The water-enveloped particles may induce a more severe erosive action than are caused by droplets free from dust.

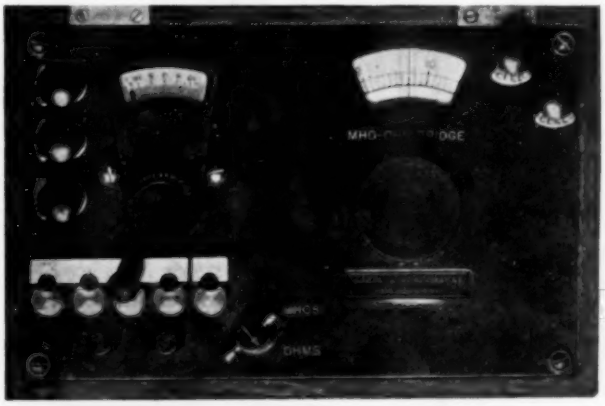


FIG. 1 PORTABLE CONDUCTANCE BRIDGE

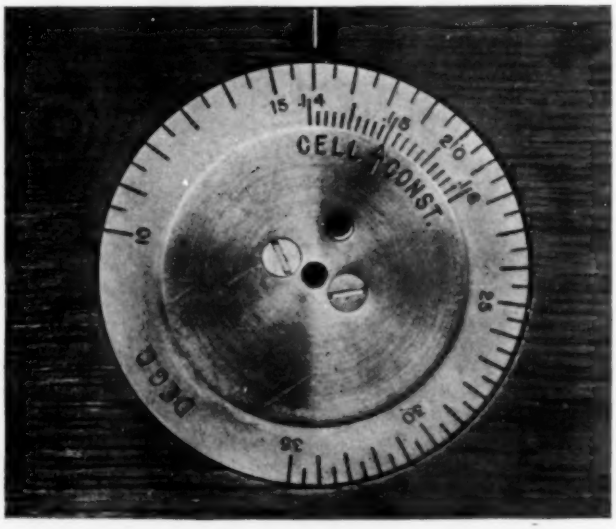


FIG. 2 TEMPERATURE AND CELL COMPENSATORS

SCOPE OF METHOD

The conductance method may be applied for all purposes where the relation between the "pure" and the contaminating supply is desired, or where concentration of soluble salts is to be controlled.

---

[1] Chief Chemist, Power Stations Dept., Duquesne Light Co.

[2] Chemical Engineer, Power Stations Dept., Duquesne Light Co.

[3] Numbers in parentheses refer to bibliography at end of paper.

Contributed by the Joint Research Committee on Boiler-Feedwater Studies and presented at the Semi-Annual Meeting, Detroit, Mich., June 9 to 12, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

The conductance method is applied herein specifically to boiler-room problems. The paper includes a presentation of the conductance method for determining the per cent by weight of entrainment of boiler water in steam, the control of concentration of water in boilers and evaporators, the detection of carry-over from evaporators, and related studies to demonstrate the precision of the conductance method. Condenser-leakage detection by the conductance method is omitted as complete descripitons appear in the technical literature. See bibliography items (4) to (14); item (10) is of specific interest.

Theory

It will suffice for this discussion to state that the conductance of an aqueous solution depends only on the ions of the dissolved substances between the electrodes of a conductance cell. Furthermore, the specific conductance of a solution is a measure of the total of ionized substances, and depends on the kinds and quantity of the ions present. The theory of conductance in aqueous solutions is discussed by Creighton (15).

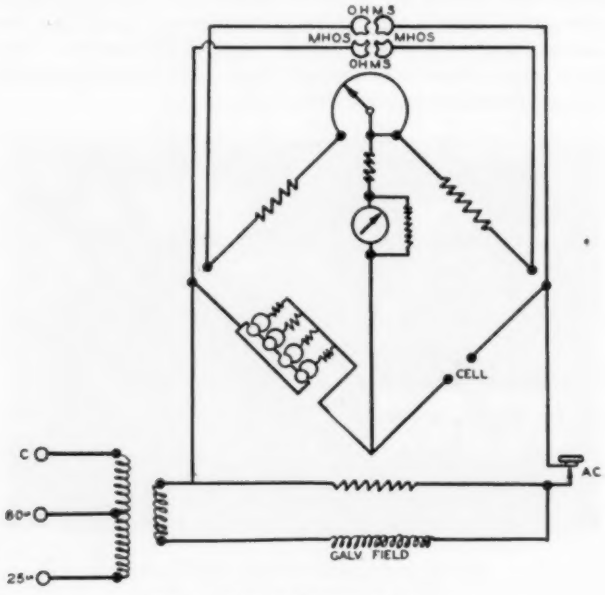


Fig. 3 Bridge Wiring Diagram

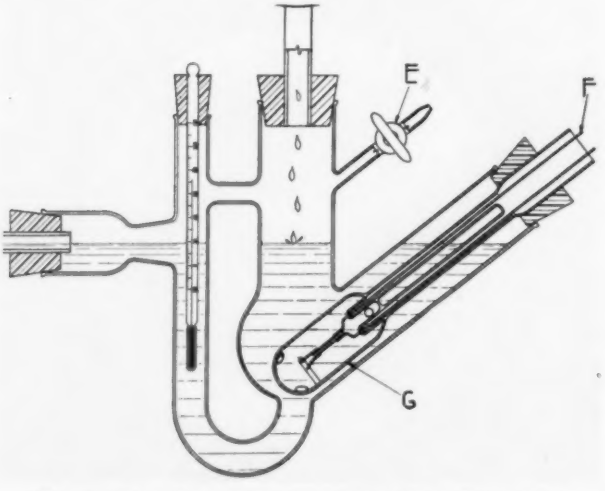


Fig. 4 Flow Cell and Conductance-Cell Electrode

Fitze (16) applied the conductance method for estimating moisture in steam leaving the boiler drum. He bases his application on the assumption that "impurities in the steam are due to entrained moisture carried over from the boiler, which, it is logical to assume, will carry over solids in the same proportion as the boiler water." In Fitze's method a per cent moisture curve was produced by measuring the conductance of boiler water and condensed steam, the latter generated in the boiler under observation. The conductance meter was calibrated in terms of the per cent of moisture by weight. This method may be considered to produce relative or apparent moisture results.

It is believed that the entrainment of boiler water in steam may be more correctly expressed if both the conductance of steam leaving the drum and the conductance of boiler water in the drum are coincidently measured, and the conductances after applying a water correction (conductance of pure water) and a correction for CO_2, are expressed as a ratio. It should be noted that hydrogen sulphide (H_2S) and ammonia (NH_3) also increase the conductance, and if present, corrections must be applied. If the ratio is multiplied by 100, this expression will show the per cent by weight of boiler water entrained in the steam.

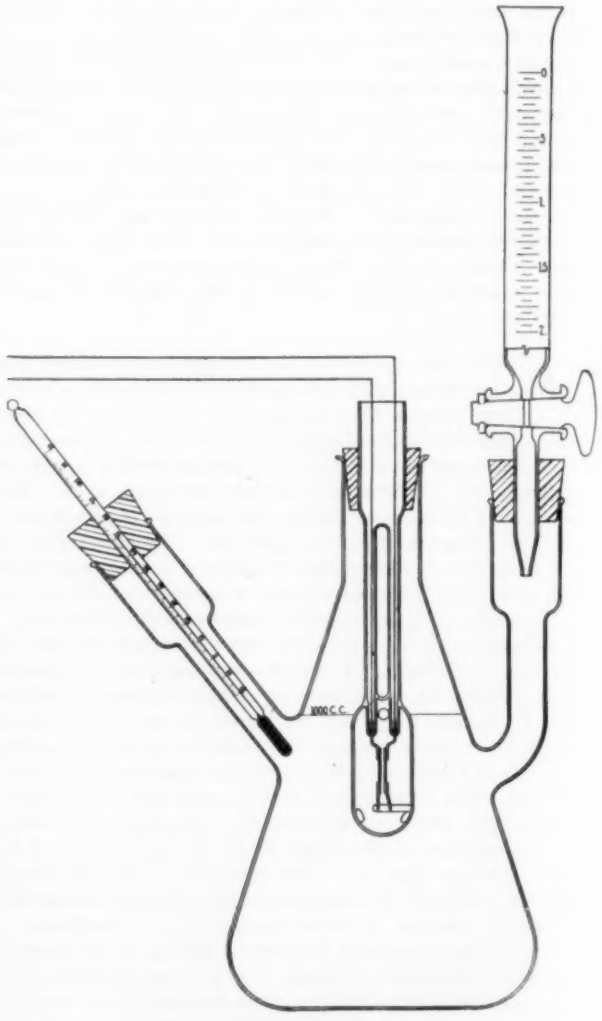


Fig. 5 Conductance Flask

$$\frac{\text{Conductance of steam}}{\text{Conductance of boiler water}} \times 100 = \text{Per cent by weight of entrainment of boiler water in steam}$$

It is not necessary to produce a calibration curve as described by Fitze.

Fitze and the authors observe similar conditions: the entrainment values are greater in magnitude than the moisture values obtained by the calorimeter method. It should be considered that the calorimeter furnishes data for estimating moisture, whereas the conductance data measure dissolved salts. For the specific purposes of obtaining data for estimating cleanliness of steam, it is apparent that the entrainment method is the appropriate one to apply.

Apparatus

The apparatus used for the investigation included a conductance meter, flow cells, a conductance cell, and a flask.

Conductance Bridge. This is a portable mho-ohm Wheatstone bridge operated on either 25- or 60-cycle alternating current. A special and very useful feature incorporated in the bridge is a temperature and cell-constant compensator. A general view of the bridge is shown in Fig. 1, and of the compensators in Fig. 2. The bridge has a range in specific conductance from 0.01 to 0.000001. The slide-wire range is from 1 to 12 mhos, each mho being divided into 10 divisions. The temperature compensator covers a range from 5 to 45 deg. cent. (41 to 113 deg. fahr.). The standardization point is 25 deg. cent. (77 deg. fahr.). The conductance-cell-constant compensator has a range from 0.07 to 0.11. The manufacturers state the instrument is calibrated to an accuracy of 1 per cent up to the 0.000001 ratio. Above that point the error may be as much as 5 per cent. The wiring diagram of the bridge is shown in Fig. 3.

Conductance Cell. The glass conductance cell fitted with platinized platinum electrodes is inserted in the flow cell or conductance flask. The cell was calibrated before the test runs were made; the cell constant was 0.861. This calibration is described in the appendix.

Flow Cell. This type of cell permits the measurement of a flowing supply, and prevents atmospheric contact and pollution. Two forms of flow cell were used. The one furnished with the apparatus is shown in Fig. 4. The improved flow cell, as shown in Fig. 6, was designed by the authors. In this design all of the water entering the cell must pass between the electrodes before discharging from the cell. The conductance flask (Fig. 5) was used in the correlated studies which follow the entrainment presentation.

All of the above apparatus, furnished by the Leeds and Northrup Company, is portable, and amply rugged and sensitive for both laboratory and plant use.

Sampling Connections on Boilers. An A.S.M.E. steam-sample nozzle was installed in a vertical run in the superheater-inlet steam lead. A connection for boiler-water samples was installed in the drum. The steam and water supplies were cooled under pressure in individual copper cooling coils, and the cooled supplies were passed through the flow cell, Fig. 4 or 6. A diagrammatic layout of the sampling system is shown in Fig. 7.

The importance of feedwater treatment and the control of boiler-water concentration justifies consideration by both the boiler manufacturer and operator in selecting proper locations and installing facilities on the boiler for procuring representative samples. Evans (17) discusses the variations in concentration in boilers due to rating, location of sampling point, and method of feed, and points out that variations in daily analyses are a source of great annoyance to the operators. One of the authors made a water transverse (18) across a cross-drum boiler and showed the wide variation in concentration.

TABLE 1 DESCRIPTION OF BOILER UNIT NO. 22

Item No.		
1	Type	B. & W. cross drum
2	Method of firing	Pulverized fuel
3	Water cooling applied	Side and rear walls Horizontal water screen
4	Drum diameter, in.	60
5	Drum length, ft.	34
6	Method of introducing feedwater	Continuous submerged feed line
7	Tubes	
	a Number of tubes wide	47
	b Number of tubes high	20
	c Arrangement of tubes in deck	6 rows lower deck 14 rows upper deck
	d Length, ft.	24
	e Outside diameter, in.	4
	f Gage, B.w.g.	7
8	Superheater location	Interdeck
9	Steam pressure, lb. per sq. in.	280
10	Total steam temperature at 200 per cent rating, deg. fahr	600
11	Superheat, deg. fahr.	190
12	Feedwater temperature, deg. fahr.	355
13	Heating surface of boiler excluding water screen, area sq. ft.	27,680
14	Water-holding capacity of boiler to drum center, lb	120,000

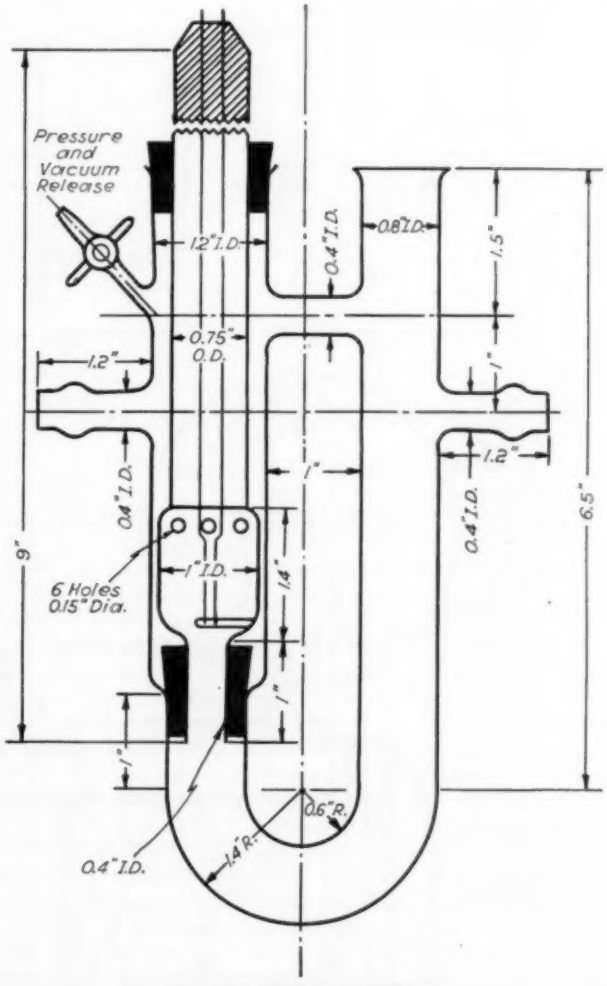


FIG. 6 REVISED FLOW-CELL

Entrainment Tests

The description of the boiler on which the entrainment tests were made is given in Table 1. The baffle arrangement in the drum is shown in Fig. 8. The steam is conducted to the superheater header by means of two leads from the drum. A throt-

tling calorimeter was installed in the "A" lead. The steam sample for conductance measurements was drawn from a connection to the throttling calorimeter on the "B" steam lead, at which the pressures before the calorimeter and the temperature at the calorimeter orifice were recorded. The recorded data were not used in these tests. The boiler-water-sample line was a $1^1/_2$-in. perforated pipe extending across the drum, above and parallel to the continuous feedwater pipe; the discharge holes in the latter are directed downward. The sample line was projected out from the drumhead via an unused extended water-

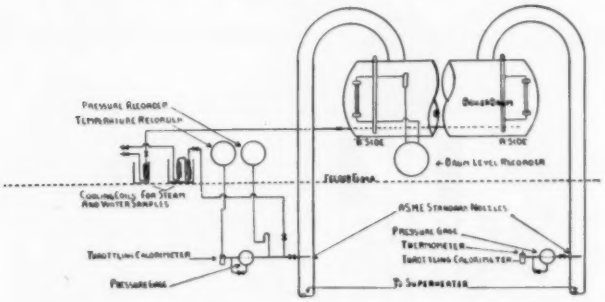


FIG. 7 SAMPLING SYSTEM AND ARRANGEMENT ON BOILER

included evaporation rates, water level, and steam quality by calorimeter, were observed once each 5 min. Conductance of steam was measured once each minute, and conductance of boiler water at the start and end of each test run.

METHOD FOR ESTIMATING ENTRAINMENT

Before each test run, the cooling coils were flushed, the discharge valves of the cooling coils were throttled, the flow cell was connected to the coil, and the cooling water was supplied at a rate to reduce the temperature of the condensed steam to within the range of the temperature compensator. Condensate was delivered from the steam coil at the rate of one quart per minute, and the boiler water from its coil at the rate of $2^1/_2$ quarts per minute. Conductance of the boiler water was observed prior to the start of the test, and when the readings became concordant, the values were recorded. The flow cell was connected to the discharge from the steam coil until consecutive readings were consistent, which gave assurance that the flow cell and conductance cell were free from boiler water. The conductance of the condensed steam was read once each minute, first adjusting the compensator for the observed temperature. Since the readings occurred at the low end of the bridge scale, a considerable travel of the slide wire was required to show a readable deflection of the galvanometer. In order to compensate for a possible error

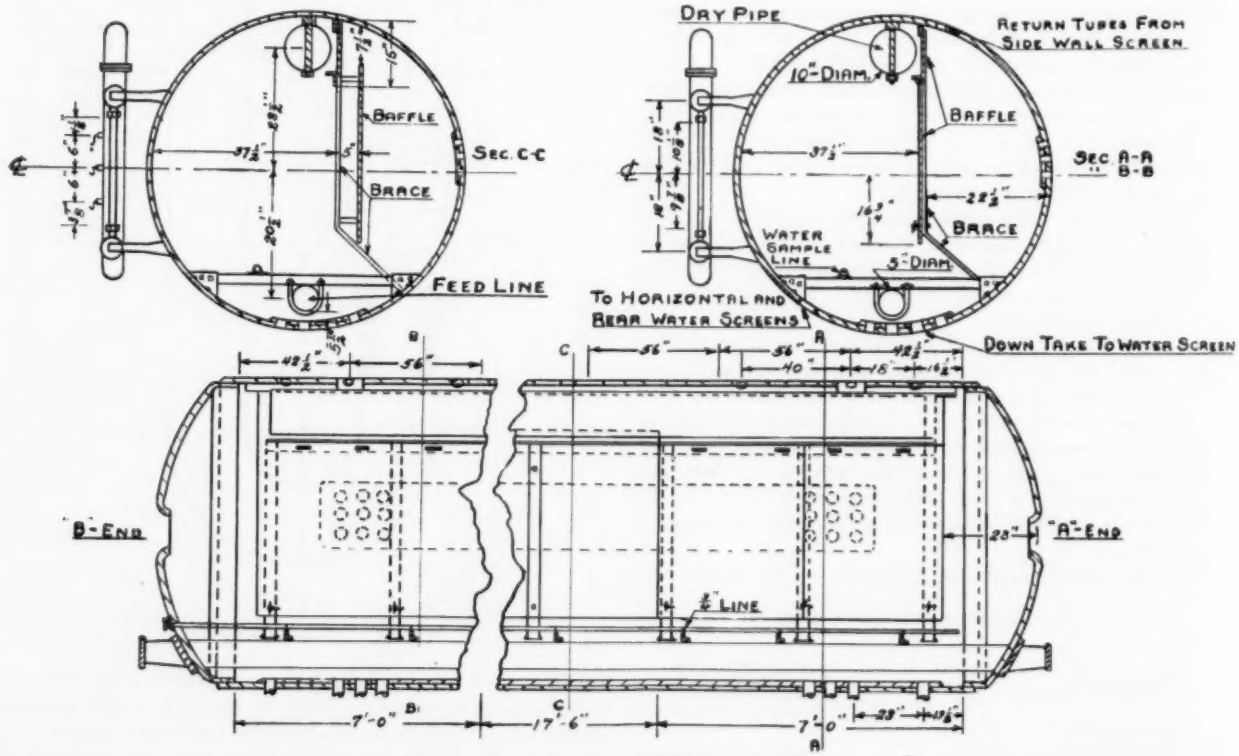


FIG. 8 BAFFLING ARRANGEMENT IN BOILER DRUM

glass connection. All external lines from the steam- and water-sample connections, including the cooling coils, were copper. No evidences of copper salts were found in samples which were completely analyzed, hence it was assumed that for the period the steam and water were in contact with the metals, the conductance introduced by their salts was negligible.

Data Observed. The boiler was under observation for a four-month period, and during this period 22 test runs were made to obtain simultaneous data on entrainment and steam quality, varying the rating and the water level. The data, which

in reading, the slide wire was rotated clockwise to show a positive deflection, and then was rotated counter clockwise until the galvanometer approached zero. This procedure was reversed for consecutive readings. After a test run was completed, the conductance of the boiler water was again measured. The average of the initial and final boiler-water readings was taken to represent the conductance of the boiler water during the $^1/_2$-hr. test run. At least $^1/_2$ hr. lapsed between tests to permit the stabilization of water and steam conditions.

A water correction of 0.9×10^{-6} mhos per cm. was deducted

from all conductance readings of steam and boiler water. Creighton (16) refers to Kendall (19), who has shown that by distilling tap water to which a few cubic centimeters of Nessler's solution has been added in Jena glass vessels and using a block-tin condenser in contact with air, water having a specific conductance of 0.9×10^{-6} mhos per cm. at 25 deg. cent. can be obtained in one distillation. The authors have observed a conductance of 1.49×10^{-6} mhos per cm. at 25 deg. cent. for distilled Pittsburgh domestic water. (Distilled in a Barnstead still having a block-tin-lined condenser, and collected and stored in large glass carboys.) This value is considerably lower than they were able to detect at the station where conductances of condensed steam, condensate, and distilled make-up were measured. In the absence of conductance data for water in contact with iron and copper, the Kendall value was adopted as the "water correction."

Each reading of conductance of steam and boiler water was corrected by deducting this correction, and each steam value was then divided by the boiler-water conductance. The quotient when multiplied by 100 is termed "entrainment in per cent by weight of boiler water in the steam." In a later section of this paper, it will be shown that the conductance of gases, and in particular CO_2, must be considered and a correction must be made in the application of conductance measurements to "pure" water supplies. The correction values are shown in the chart, Fig. 13, developed for Allegheny River water contamination of

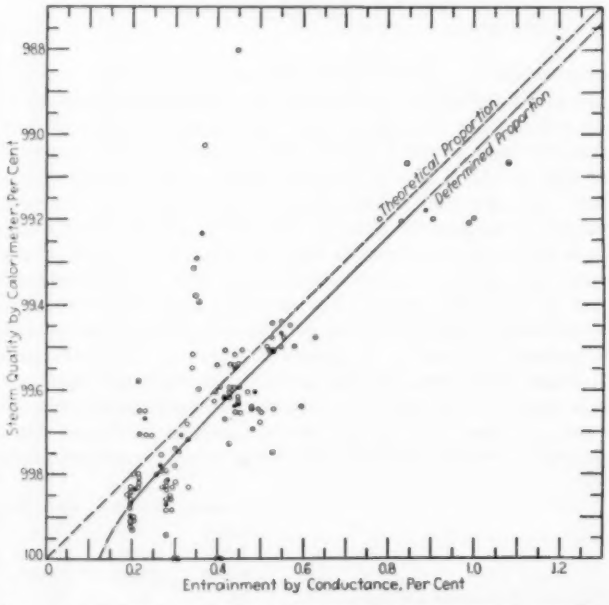


FIG. 9 RELATION OF STEAM QUALITY BY CALORIMETER AND ENTRAINMENT BY CONDUCTANCE

boiler feedwater. The CO_2 content of the condensed steam in these tests measured 0.75 part per million (p.p.m.). No correction was applied to the data since the error is large in estimating so low a CO_2 value. Where a considerable amount of CO_2 is present in the water, a deduction must be applied to the conductance of steam.

RELATION BETWEEN MOISTURE AND ENTRAINMENT

Efforts to plot the observed data for steam quality and entrainment against the other variables, water levels, and ratings indicated no direct relations. It is believed that this is due to the few number of test runs, and to the effects of other factors which were not observed. For example, variations in boiler-water concentration for the various test runs, fluctuation in concentration of boiler feedwater, normal fluctuation in flow rate of feedwater, and finally deviations of observed and actual water levels. A relationship is apparent between the steam quality by calorimeter and entrainment by conductance, and is shown graphically in Fig. 9. The steam-quality data plotted represent one reading each 5 min. and the simultaneously observed entrainment data represent the average of five one-minute readings. The 45-deg. diagonal passing through the origin represents the theoretical identity of the percentage of moisture in the steam and entrain-

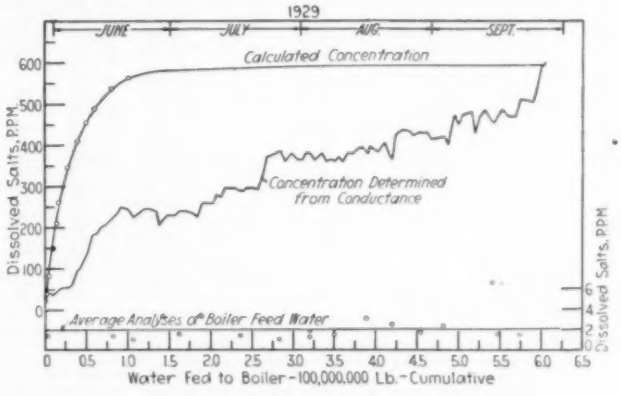


FIG. 10 CONCENTRATION OF BOILER WATER

ment. In the areas included between the diagonal and the abscissa, the per cent entrainment exceeds the per cent moisture by calorimeter. Observations plotted in this area indicate a possible throttling action by the dry pipe of the steam leaving the drum. In the area between the diagonal and the ordinate, the per cent moisture by calorimeter exceeds the per cent entrainment by conductance. Since the steam cannot carry moisture in excess of the amount entrained therein, values in the area above the diagonal would indicate an impossible condition. The observed values are represented by the circles for all runs. The dots represent the average per test run. The average for the calorimeter normal, test 12, and for tests 16, 17, and 18, in which attempts were made to observe steam quality under no steam-flow conditions, are shown on the 100 per cent quality line. If these four values are not considered, it will be observed the related data parallel in general the theoretical proportionality graph, but fall slightly below it. It is further noted for the same group of determinations that the distribution of points is much wider on the steam-quality scale than on the entrainment scale. This indicates that the entrainment method is more accurate than the calorimetric procedure. The wide divergence of entrainment values from steam quality, for very low and no steam-flow conditions, may be interpreted to represent non-equilibrium conditions as respects the quality of steam, because of the rapid reduction in evaporation rates to a condition suitable for establishing a calorimeter normal. For the observed conditions, it appears that the entrainment in steam is greater in magnitude for the lower steam-flow conditions than is found under the normal steam output of the boiler. Agreement is found with Fitze concerning the difficulties of establishing the calorimeter normal, even at lower boiler pressures. During the service run of the boiler a daily observation was made for concentration, using the conductance method. These values were translated to dissolved salts, parts per million, using the calibration curve *A-A*, in Fig. 12. The interpolated dissolved salts were plotted in Fig. 10. The calculated concentration curve was also plotted, based on the average per cent entrainment, 0.35 per cent, column 7, Table 2. The development and application of the formulas

TABLE 2 STEAM QUALITY FROM CALORIMETER, CONDUCTANCE AND CHEMICAL DATA

1	2	3	4	5	6			7			8	9				10	11	12
					Steam quality by calorimeter			Entrainment by conductance				Specific conductance mhos per cm. at 25 deg. cent.						
												Condensed steam		Boiler water				
Test run no.	Date of test (1929)	Duration of run, hr.	Evaporation lb. per hr.	Water level in drum, in. from center	Min. per cent	Max. per cent	Avg. per cent	Min. per cent	Max. per cent	Avg. per cent	Entrainment from chemical data Avg. per cent	Min. 10^{-6}	Max. 10^{-6}	Avg. 10^{-6}	Avg. 10^{-6}		Dissolved salts in steam from entrainment, avg. p.p.m.	Dissolved salts in boiler water from conductance, avg. p.p.m.
1	7-6	0.5	241,400	−3.2	99.48	99.64	99.51	0.45	0.63	0.53	0.97	1.92	2.71	2.29	430		1.1	215
2	7-6	0.5	241,300	+2.7	99.07	99.21	99.18	0.78	1.08	0.90	..	3.26	4.52	3.80	424		1.9	215
3	7-6	0.5	242,100	−5.6	99.44	99.50	99.47	0.53	0.58	0.55	0.99	2.20	2.45	2.30	420		1.6	210
4	7-6	0.5	204,800	−2.9	99.51	99.59	99.55	0.40	0.52	0.44	..	1.67	2.18	1.85	422		0.9	210
5	7-6	0.5	198,600	−2.7	99.52	99.67	99.62	0.39	0.45	0.42	..	1.60	1.85	1.74	413.5		0.9	205
6	7-6	0.5	154,000	−2.6	99.59	99.73	99.64	0.43	0.50	0.45	..	1.80	2.09	1.86	411.5		0.9	205
7	7-6	0.5	30,700	−1.2	99.61	99.75	99.66	0.43	0.53	0.49	..	1.80	2.33	2.05	420		1.0	210
8	7-10	0.25	22,700	−4.9	Normal	Normal	99.53	..	..	..	..	..	..	..	..		..	..
9	7-13	0.5	239,100	−2.8	99.60	99.69	99.64	0.36	0.54	0.44	0.69	2.14	3.19	2.59	594		1.3	300
10	7-16	0.5	239,700	−3.1	99.80	99.83	99.82	0.27	0.30	0.28	0.73	1.52	1.79	1.62	562		0.8	285
11	7-16	0.5	242,700	−1.5	99.76	99.82	99.78	0.26	0.28	0.27	0.74	1.46	1.57	1.52	553.5		0.8	280
12	7-16	0.5	11,200	−2.6	Normal	Normal	99.19	0.28	0.31	0.31	..	1.58	1.77	1.73	564		0.8	285
13	7-16	0.5	237,100	+0.5	99.83	99.94	99.87	0.28	0.29	0.28	0.72	1.58	1.70	1.63	572		0.8	290
14	7-16	0.5	232,900	+2.9	99.55	99.83	99.71	0.30	0.34	0.32	..	1.71	1.91	1.81	570.5		0.9	285
15	7-16	0.5	242,800	+4.6	98.80	99.52	99.23	0.34	0.45	0.37	..	1.87	2.60	2.07	556		1.0	280
16	7-21	0.5	....	−1.0	Normal	Normal	99.37	0.40	0.42	0.41	..	2.85	3.04	2.96	715		1.5	360
17	7-21	0.5	....	−3.1	Normal	Normal	99.96	0.39	0.42	0.40	..	2.76	2.97	2.87	715		1.4	360
18	7-21	0.5	....	−4.9	Normal	Normal	99.42	0.27	0.31	0.30	..	1.94	2.25	2.15	715		1.1	360
19	9-20	0.75	250,100	−0.7	99.58	99.71	99.67	0.22	0.24	0.23	..	2.37	2.57	2.45	1079		1.3	545
20	9-20	0.25	....	−2.1	Normal	Normal	98.81	..	..	..	..	..	..	..	..		..	..
21	9-20	0.75	248,000	−1.9	99.80	99.86	99.83	0.20	0.22	0.21	0.36	2.31	2.55	2.43	1166		1.2	585
22	9-20	0.75	249,500	−3.9	99.84	99.91	99.89	0.19	0.21	0.20	0.36	2.22	2.42	2.34	1163		1.2	585
Average	..	..	....	...	....	....	...	..	..	0.35	..	..	..	2.20	....		1.1	...

Notes: Values in columns 9 and 10 corrected for conductance of pure water.
Values in column 8 calculated from formula in appendix.
Values in column 11 calculated from column 7 × column 12.
Values in column 12 interpolated from calibration curve Fig. 12.
Run 16—Burners off 12 min. before run started. Feedwater valves and steam valves shut.
Run 17—Burners off 70 min. before run started. Feedwater valves and steam valves shut, closed non-return valves.
Run 18—One burner lit, 10 minutes before run started. Feedwater valves and steam valves shut, closed non-return valves.
After run no. 15, calorimeter was disassembled, and mill scale removed which partly obstructed orifice.

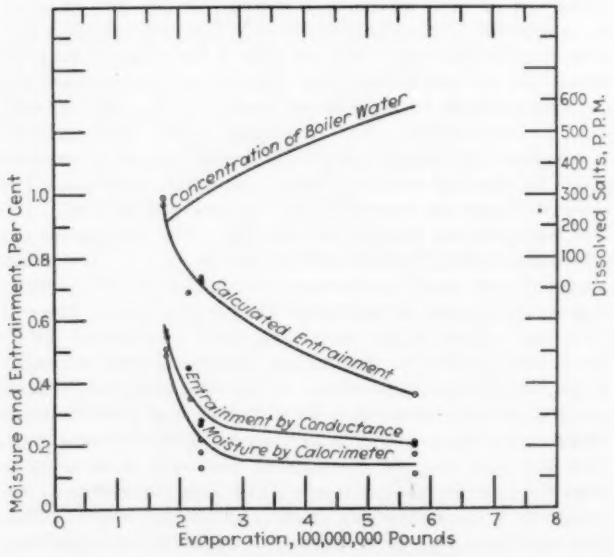


FIG. 11 STEAM QUALITY BY VARIOUS METHODS

for this calculation are discussed in the Appendix. The value used for entrainment (0.35 per cent) is the average of test runs 1, 3, 9, 10, 11, 13, 21, and 22, in which the evaporation rates were maintained between 230,000 and 250,000 lb. per hr. This output is normally maintained by the boiler for 20 hr. daily. The water levels in the drum were maintained below the drum center, which conforms to regular operating practice. The average of dissolved salts of the feedwater supplied to the boiler was 2.08 p.p.m. This value is read from Fig. 13, the calibration curve developed for dissolved salts in "pure" water in the presence of CO_2. The tabulated values for the calculated concentration plotted in Fig. 10 are given in Table 8. It will be observed by reference to Fig. 10 that the concentration at infinite evaporation (no blowdowns) is 594 parts per million (p.p.m.), and at 600,000,000 lb. evaporation the concentration determined by conductance was found to be 585 p.p.m. (The boiler was removed from service for inspection purposes.) The agreement of the calculated and determined value for concentration supports the view that the entrainment value used, 0.35 per cent, correctly expresses the average carry-over from the boiler drum. It is shown in the Appendix that the entrainment may be calculated, using the concentration of the salts in the feedwater and in the boiler water. The acceptance of the interpolated dissolved-salt values of boiler water based on conductance measurements is justified by the agreement shown in Fig. 12 of curves *A-A* and *B-B*. Similarly the use of the interpolated dissolved-salt values of feedwater is justified by the agreement shown between the dissolved salts in feedwater and in condensed steam shown in column 8, Table 7. A summary of the significant values is:

	Laboratory analysis		Field test
	Boiler feedwater	Condensed steam	Condensed steam
Number of samples averaged.....	22	2	20
Dissolved salts, calculated from the analysis, p.p.m.....	2.63	1.63	...
Dissolved salts, interpolated from conductance, p.p.m.....	2.08	2.50	1.1[b]
Specific conductance, mhos per cm. × 10^{-6} at 25 deg. cent.....	5.45[a]	6.10[a]	2.20

[a] Time interval between sampling and analysis, one week.
[b] Calculated from entrainment data.

The interpolation of conductance to dissolved salts is made from the graph in Fig. 13. The entrainment was calculated using the boiler-feedwater interpolated value of 2.08 p.p.m. The comparison between the determined and calculated entrainment and steam quality by calorimeter plotted against total evaporation is shown in Fig. 11; at the maximum evaporation agreement is observed between the calculated and determined entrainment. The tabulated data are shown in Table 3. The agreement as

TABLE 3 STEAM QUALITY BY VARIOUS METHODS

Col. 1 Test run number	2 Evaporation of boiler feedwater to date of test, pounds	3 Concentration of dissolved salts in boiler feedwater, p.p.m.	4 Steam quality calculated from calorimetric data, per cent	5 Entrainment calculated from entrainment data, per cent	6 Entrainment calculated from chemical data, per cent
1	181,400,000	215	99.51	0.53	0.97
3	181,400,000	210	99.47	0.55	0.99
9	218,400,000	300	99.64	0.44	0.69
10	234,000,000	285	99.82	0.28	0.73
11	234,000,000	280	99.78	0.27	0.74
13	234,000,000	290	99.87	0.28	0.72
21	585,000,000	585	99.83	0.21	0.36
22	585,000,000	585	99.89	0.20	0.36

Column 6—Formula used. Entrainment = $\frac{c \text{ Concentration boiler feed}}{C_2 \text{ Concentration boiler water}}$

c = 2.08 p.p.m. = average from Table 7, Col. 8.

TABLE 4 COMPARISON OF DISSOLVED SOLIDS AND DISSOLVED SALTS BY CONDUCTANCE AND CHEMICAL METHODS

1	2	3	4	5	6	7
—Chemical method—			—Conductance method—			
Dissolved solids by analysis, p.p.m.	Dissolved salts by analysis, p.p.m.	Dissolved salts calculated, p.p.m.	Specific conductance at 25 deg. cent. observed, mhos per cm.	Specific conductance at 25 deg. cent. calculated, mhos per cm.	Dissolved salts interpolated from obs. cond. plot, p.p.m.	Dissolved salts interpolated from calc. cond. plot, p.p.m.
Data from boiler fed with "pure" feedwater						
289	215	223	4.45×10^{-4}	4.40×10^{-4}	223	230
366	303	307	5.62	5.95	283	310
377	305	321	6.20	6.02	313	315
365	305	330	6.64	6.39	335	330
455	379	398	7.54	7.82	380	405
393	337	367	7.20	7.25	363	375
415	337	348	7.08	6.74	358	350
467	407	429	7.80	8.30	395	430
517	449	468	8.60	9.10	435	465
488	440	469	8.20	8.80	415	455
507	457	502	9.42	9.69	478	500
506	438	476	9.00	9.16	455	475
569	513	531	10.06	10.21	510	525
526	474	499	9.65	9.62	488	495
Data from boiler fed with 100 per cent zeolite-softened water						
2745			46.5×10^{-4}		2200	
1720			33.5		1600	
1520			29.0		1350	
1548			31.8		1500	
1589	1645		33.8	37.8×10^{-4}	1450	1500
2313			44.5		2100	
1988			40.5		1900	
1714			37.5		1750	
1849	1858		44.7	46.1	2100	1800
1914			40.5		1900	
1776			38.2		1800	
1733			39.2		1850	
1704	1744		44.2	46.4	2100	1800
1995	2011		49.6	55.4	2350	2200

Column 1 Evaporated residue of 500 cc. boiler water, residue dried at 120 deg. cent.

2 Difference between column 1 and sum of colloidal materials by analysis of column 3.

TABLE 5 AVERAGED ANALYSES OF ALLEGHENY RIVER WATER

Number of samples averaged	22
Dissolved ion:	parts per million
Calcium (Ca)	19.5
Magnesium (Mg)	3.9
Sodium (Na)	17.9
Bicarbonate (HCO_3)	16.8
Sulphate (SO_4)	68.5
Chloride (Cl)	13.4
Nitrate (NO_3)	0.3
Sum of dissolved ions	140.3 (*a*)
Colloidal salts	
Organic matter	6.4
Silica (SiO_2)	9.0
Oxides, iron and alumina (R_2O_3)	4.2
Sum of colloidal salts	19.6 (*b*)
Dissolved solids ($a + b$)	159.9

Note: The colloidal salts are considered to have a negligible effect on the conductance of water.

The weights of chemically pure reagents were calculated from the dissolved ion values in Table 5, to produce when dissolved, the "synthetic" water containing 1000 p.p.m. of dissolved salts. The weights are shown in Table 5-A.

shown between the determined entrainment and moisture by calorimeter was anticipated from the concordance observed in Fig. 9.

In Fig. 10 a wider divergence is observed during the initial part of the run between the observed dissolved salts (interpolated from conductance) and the theoretical concentration, but at the end of the run the two curves coincide. Similarly in Fig. 11 closer agreement is observed at the end of the run between the steam quality by various procedures than is found in the early part of the run. It is further observed that the steam quality apparently improved with the increase in concentration of boiler water and total evaporation. Hence it may be noted that for the observed case some condition existed in the more dilute water which is unfavorable to the production of dry and clean steam.

Boiler- and Evaporator-Water Concentration Measured by Conductance

In attempts to measure concentration of boiler water by conductance an electrode was designed by other investigators (private communications) for insertion through the drum of a boiler. The difficulties encountered included the deterioration of the insulation of the cell element by the boiler water, and inability suitably to locate the element in a zone of uniform water concentration and temperature. Relatively large errors in measurements may be caused by local variations in concentration and temperature when fresh feedwater is introduced into the drum adjacent to the location of the cell. These observations apply as well to evaporators, and in the following discussion it should be remembered that concentration in evaporator salines is included, even though not specifically discussed.

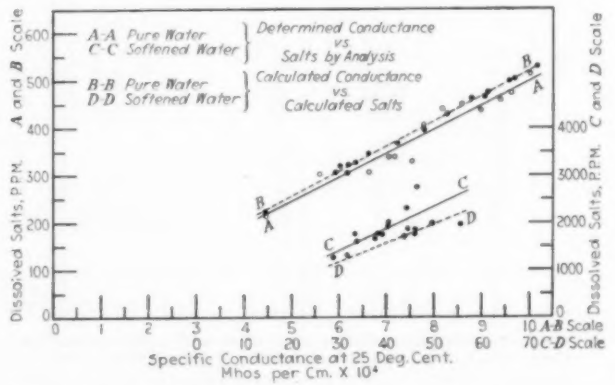


FIG. 12 RELATION OF SPECIFIC CONDUCTANCE TO THE DISSOLVED SALTS IN BOILER WATER

These difficulties were avoided by adopting the simpler procedure of withdrawing water through an internally installed perforated pipe extending across the drum. The water was cooled under pressure to prevent "flashing," and the conductance was measured as the water passed continuously through a glass flow cell (Fig. 6) in which the conductance cell was installed. Compensation was made for temperature observed before each

TABLE 5-A WEIGHTS OF REAGENTS FOR "SYNTHETIC" RIVER WATER

Reagents, chemically pure grade:	Weight in solution milligrams
Calcium sulphate, $CaSO_4.2H_2O$	596.4
Magnesium sulphate, $MgSO_4.7H_2O$	281.6
Sodium bicarbonate, $NaHCO_3$	164.5
Sodium sulphate, Na_2SO_4	66.2
Sodium chloride, NaCl	157.4
Sodium nitrate, $NaNO_3$	2.8
Sum of dissolved salts	1268.9

The weights of reagents were dissolved in laboratory distilled water, (boiled to free the water of atmospheric gases, cooled, but excluding CO_2). 1 cc. of the solution contains 1 p.p.m. dissolved salts (anhydrous basis).

conductance reading. The readings were reported in specific conductance at 25 deg. cent.

As previously stated, the conductance of water varies directly as the dissolved salt content, provided the relation of the ions in the water remains the same. For each supply of water or steam in a specific station, the relation of conductance and dissolved salts will be a constant; however, the constants will differ from one another for various supplies. When desired, the calibration of the conductance meter is made in terms of dissolved salts, parts per million, the adopted procedure for stating concentration in plant control (20). The calibration, once established, is permanent, and is applicable to all boilers in the station unless a major change is made in the quality of feedwater or chemical treatment.

TABLE 6 EFFECT OF DISSOLVED SALTS AND CARBON DIOXIDE ON THE CONDUCTANCE OF LABORATORY DISTILLED WATER

1	2	3	4	1	2	3	4
Test number	Salts of "synthetic" water added, p.p.m.	Observed carbon dioxide, p.p.m.	Observed specific conductance, mhos per cm. at 25 deg. cent.	Test number	Salts of "synthetic" water added, p.p.m.	Observed carbon dioxide, p.p.m.	Observed specific conductance, mhos per cm. at 25 deg. cent.
1	0	0.82	1.46×10^{-6}	5	0	1.18	1.45×10^{-6}
	0	3.17	3.07		8	1.18	19.88
	0	5.05	3.99		8	5.88	20.93
	0	8.81	5.41		8	7.05	21.73
					8	10.58	22.38
					8	15.04	23.26
2	0	1.18	1.54×10^{-6}	6	0	1.20 [a]	1.49×10^{-6} [b]
	1	1.18	3.91		16	1.18	37.93
	1	3.17	5.12		16	4.70	38.74
	1	4.47	5.85		16	7.64	39.46
	1	9.52	7.44		16	9.17	39.94
					16	16.80	41.22
3	0	1.65	1.56×10^{-6}	7	0	1.20 [a]	1.49×10^{-6} [b]
	2	2.00	6.00		32	0.82	74.59
	2	6.11	7.58		32	3.76	75.15
	2	7.64	8.56		32	7.17	75.63
	2	9.75	9.46		32	9.89	75.87
	2	10.93	9.86		32	15.98	76.27
4	0	1.18	1.43×10^{-6}	8	0	1.20 [a]	1.49×10^{-6} [b]
	4	1.41	10.91		64	1.18	143.56
	4	5.05	12.19		64	4.35	143.56
	4	7.05	12.99		64	11.63	144.34
	4	9.99	13.71		64	23.03	145.96
	4	11.75	14.28		64	37.37	145.96

[a] Used average of carbon dioxide values observed in tests numbers 1 to 5.
[b] Used average of sp. cond. values observed in tests numbers 1 to 5.

The data required for calibrating the meter include a series of conductance readings, and dissolved salt concentrations, the latter determined by analysis, over the range of concentration anticipated in the service run of a boiler or an evaporator. The smooth curve drawn through the plotted points establishes the desired relation for the boiler water under observation, and permits the ready conversion of conductance units to dissolved salts. When the conductance is to be expressed in specific-conductance units, it must be referred to a stated temperature, and the cell constant must be known and applied. Temperature correction and cell-constant calibration are discussed in the Appendix. In the laboratory analysis of water, the weights are obtained both of the dried residue after evaporating a measured volume of filtered water and of the constituents of the residue. The analysis report gives the weight in parts per million of colloidal and non-ionized materials which include organic matter, silica, and oxides of iron and aluminum, and of ionized materials which may include calcium, magnesium, sodium, carbonate, bicarbonate, hydroxide, sulphate, phosphate, chloride, and nitrate. The evaporated residue is termed "dissolved solids." When the sum of the colloidal and non-ionized materials is subtracted from the dissolved solids, the difference is termed "dissolved salts."

Conductance by Calculation

The conductance may be calculated from the ionic analysis by multiplying each ion present by the respective equivalent ionic-conductance factor. The sum of the products equals the conductance of the water expressed in specific-conductance units. It may be considered that practically complete ionization of salts occurs in water at the concentrations normally met with in the various supplies in the steam plant. Neglecting the colloidal substances, which lack conductance characteristics, the water contains metallic and non-metallic ions, each possessing a distinct ionic equivalent conductance. The factors for converting the ions to conductance, and the application of the factors to a typical boiler-water analysis, are shown in Table 9. The calculation of the ionic analysis to hypothetical combination to determine the dissolved salts is both unnecessary and erroneous (20). The comparison of the measured and calculated conductance provides a means of determining the relation between dissolved salts and conductance and for checking the analytical procedure.

As previously stated, observations for steam quality were made on a boiler during its service run. The feedwater supplied to the boiler was a mixture of 98 per cent condensate and 2 per cent distilled make-up. The boiler water was sampled periodically. These samples were completely analyzed, and the conductance was both measured and calculated. The data are shown graphically in Fig. 12 and are tabulated in Table 4, showing the relation between conductance and dissolved salts. A similar procedure was used at another station for establishing the relation between conductance and dissolved salts of boiler water. This boiler was supplied with 100 per cent zeolite-softened feedwater. These data are also included in the graph and table.

Conductance Method Produces Close Results

The values are reported as specific conductance at 25 deg. cent. The water correction 0.9×10^{-6} mhos per cm. has a very small effect on the observed values of boiler water, hence the origin of the graphs may be considered at zero conductance and dissolved salts. In Fig. 12 the curves marked *A-A* and *B-B* refer to the station where "pure" feedwater was concentrated, and curves marked *C-C* and *D-D* refer to the station where zeolite-softened water was concentrated. The calibration curves for each station are respectively *A-A* and *C-C*. They represent the relation of observed conductance (column 4) and dissolved salts by analysis (column 2) in Table 4. The curves drawn to check this procedure, *B-B* and *D-D* for each respective station, represent the relation of calculated conductance (column 5) and dissolved salts calculated from the analysis (column 3). The boiler-water concentration in terms of dissolved-salts range for the "pure" water station was from 215 to 474 p.p.m., and for the zeolite-softened water station, from 1100 to 2750 p.p.m. The agreement between the two procedures is shown by the values taken from each curve and shown in columns 6 and 7. It may be concluded that the conductance method produces results well within the error of analysis. It is obvious that the conductance measurement made in less than one minute after the sample is available is preferable as a plant procedure for concentration control, when compared to the many hours required for the completion of the analysis.

TABLE 7 COMPLETE CHEMICAL ANALYSES OF BOILER FEEDWATER AND CONDENSED STEAM

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
		Solids in water p.p.m.			Salts in water p.p.m.									Corrected ionic analysis						
Date supply sampled	Unit sample	Dissolved by anal.	Suspended by anal.	Total by analysis	Calculated from analysis	By analysis	Interpolated from sp. cond.	Organic matter by analysis, p.p.m.	Silica by analysis, p.p.m.	Oxides (Fe + Al) by analysis, p.p.m.	pH analysis	Free CO_2 by analysis, p.p.m.	Specific conductance at 25 deg. cent.	Calcium, p.p.m.	Magnesium, p.p.m.	Sodium, p.p.m.	Bicarbonate, p.p.m.	Sulphate, p.p.m.	Chloride, p.p.m.	Nitrate, p.p.m.
1929																				
Boiler feed average		4.76	0	4.76	2.63	2.39	2.08	0.61	1.53	0.23	6.9	1.4	5.45×10^{-6}	0.21	0.06	0.48	0.67	0.88	0.29	0.02
5–2	1–2	5.40	0	5.40	4.72	4.16	1.50	0.18	0.88	0.18	6.2	1.0	4.00	0.25	0.11	0.92	2.36	0.99	0.05	0.04
5–15	4B	7.18	0	7.18	3.83	3.18	1.20	0.73	3.25	0.02	5.6	2.1	4.00	0.13	0.06	0.99	1.11	0.99	0.55	0
5–16	1–2	2.53	0	2.53	2.46	1.16	1.20	0.48	0.88	0.01	5.4	1.5	3.60	0.18	0.03	0.28	0	0.49	0.48	0
5–22	3A-B	8.75	0	8.75	5.11	4.51	2.80	1.98	2.03	0.23	5.3	1.3	7.00	0.25	0.07	1.23	1.28	1.65	0.50	0.13
5–31	4B	5.10	0	5.10	3.23	3.56	1.50	0.08	1.18	0.28	5.9	1.0	3.90	0.25	0.06	0.79	0	1.07	1.02	0.04
6–16	1–2	5.45	0	5.45	3.69	4.32	1.50	0.10	0.85	0.18	6.2	1.0	4.00	0.25	0.12	0.61	1.71	0.91	0.05	0.04
6–21	3A	3.50	0	3.50	2.37	2.17	1.10	0.20	0.83	0.30	7.5	1.0	3.00	0.36	0.11	0.15	0.79	0.82	0.10	0.04
6–28	4A	9.40	0	9.40	3.15	2.69	3.40	1.40	5.03	0.28	8.4	1.9	8.70	0.40	0.14	0.30	1.08	0.99	0.20	0.04
7–2	1–2	5.98	0	5.98	1.91	2.00	1.60	1.18	1.80	1.00	6.4	1.1	4.30	0.26	0.05	0.28	0	0.99	0.29	0.04
7–8	3A-B	8.15	0	8.15	4.62	3.94	3.30	0.93	2.93	0.35	4.9	1.3	8.00	0.35	0.05	0.89	2.23	0.91	0.15	0.04
7–16	1–2	5.80	0	5.80	2.14	2.25	1.50	1.20	1.80	0.55	5.3	1.4	4.20	0.25	0.06	0.38	0	0.91	0.54	0
7–26	3A-B	2.58	0	2.58	1.22	1.03	1.10	0.45	0.90	0.20	4.8	1.1	3.10	0.13	0.04	0.21	0	0.66	0.18	0
8–2	4B	2.98	0	2.98	1.39	1.42	1.20	0.50	0.88	0.18	...	1.2	3.50	0.20	0.04	0.19	0	0.74	0.22	0
8–2	1–2	2.65	0	2.65	1.28	1.09	1.90	0.43	0.95	0.18	...	0.9	5.00	0.15	0.04	0.21	0	0.66	0.22	0
8–8	3A-B	2.70	0	2.70	1.18	1.09	1.50	0.58	0.90	0.13	...	1.3	4.20	0.13	0.04	0.21	0	0.49	0.31	0
8–16	4A	3.35	0	3.35	1.72	1.60	3.10	0.42	1.25	0.08	7.8	1.0	7.50	0.25	0.05	0.18	0.48	0.66	0.10	0
8–22	3A-B	2.13	0	2.13	1.31	1.20	2.50	0.30	0.53	0.10	5.2	1.0	6.30	0.10	0.04	0.22	0.46	0.49	0	0
8–29	4B	7.10	0	7.10	3.59	3.09	1.80	0.68	3.15	0.18	5.6	2.3	4.80	0.13	0.07	0.92	0.76	1.07	0.60	0.04
9–3	3A-B	2.18	0	2.18	1.33	0.85	2.40	0.45	0.60	0.28	7.0	2.0	6.50	0.10	0.11	0.13	0.28	0.66	0.05	0
9–14	4A	3.20	0	3.20	1.74	1.49	6.70	0.38	1.23	0.10	7.4	2.0	16.00	0.20	0.05	0.24	0.52	0.74	0	0.04
9–16	1–2	5.55	0	5.55	4.40	4.19	1.50	0.20	0.98	0.18	6.0	1.4	4.30	0.26	0.10	0.91	1.77	0.82	0.50	0.0
9–20	3A-B	3.00	0	3.00	1.30	1.40	1.50	0.55	0.90	0.15	6.2	1.4	4.10	0.15	0.04	0.24	0	0.66	0.30	0
Condensed steam to superheaters																				
	Avg.	4.46	0	4.46	1.63	2.04	2.50	0.52	1.83	0.08	...	1.5	6.10	0.22	0.03	0.38	0	0.66	0.39	0
7–9	22	6.47	0	6.47	2.05	2.75	2.40	0.74	2.95	0.03	...	1.1	6.00	0.30	0.02	0.56	0	0.66	0.51	0
9–20	22	2.45	0	2.45	1.20	1.32	2.50	0.30	0.70	0.13	...	1.8	6.20	0.13	0.04	0.20	0	0.66	0.17	0

Column 3 Dissolved solids by analyses, evaporation of 4000 cc. residue dried at 180 deg. cent.
4 Suspended matter estimated by filtering total sample through asbestos mat in crucible.
5 Sum of columns 3 and 4.
6 Sum of determined and corrected ions, column 15 to 21, inclusive.
7 Columns 3 less the sum of columns 9, 10 and 11; these materials are considered colloidal in character and have a negligible conductance.
8 Interpolated from graph, Fig. 13, correcting for CO_2 shown in column 13.
9, 10 and 11. Followed standard methods, American Public Health Assn., 1926 edition.
12 Determined with hydrogen electrode.
13 Used standard methods, A.P.H.A.
14 Conductance measured with mho-ohm Wheatstone bridge.
15, 16, 18, 19, 20 and 21. Used standard methods, A.P.H.A.
17 Sodium is indirectly determined. Evaporated residue, column 3, treated with H_2SO_4 and evaporated, then weighed to determine total sulphate, from which are subtracted silica and oxides of iron and alumina (columns 10 and 11), and calculated sulphate of lime and magnesia. The difference is sodium sulphate, from which the sodium ion is calculated.

TABLE 8 CALCULATED CONCENTRATION OF BOILER WATER AT VARIOUS EVAPORATIONS

Evaporation Lb. × 10^6	$\frac{E\ (\text{Evap.})}{10^{2\cdot303\ K}} = \frac{c\ (1 - E)}{c_1 - (C_2 \times E)}$	Concentration, C_2 of dissolved salts, p.p.m.
1	1.0296	19.3
2	1.0600	35.5
5	1.1570	82.4
10	1.3384	151.6
15	1.5488	211.9
20	1.7914	263.6
30	2.3977	347.3
40	3.2092	409.7
50	4.2954	456.4
60	5.7491	491.3
80	10.299	536.8
100	18.45	562.2
Infinity	Infinity	594.3

CO_2 Correction of Conductance

The authors recently recommended the installation of conductance apparatus for recording the quality of condensate returned to a central heating plant through an extensive condensate-return system. Since it was desired to calibrate the recorded data in terms of actual dissolved salts, a laboratory investigation was made to determine the effect of varying concentration of CO_2 on the conductance of water and the development of a CO_2 correction curve. Also in the case of evaporators, when CO_2 gas is evolved from the water fed to the evaporators, the conductivity of the condensed vapor is not a true measure of dissolved salts unless the conductance is compensated for the influence of CO_2.

A survey of the available "pure" water supplies in the various steam stations and of laboratory-distilled water produced evidence that the laboratory supply prepared in a Barnstead still, and having a block-tin-lined condenser, had a minimum of con-

TABLE 9 CONVERSION FACTORS FOR CALCULATING CONDUCTANCE FROM IONIC ANALYSES

Ions by analysis, p.p.m.	Milli-equivalents per million	Ionic equivalent conductance, mhos per equivalent at 25 deg. cent.	Factors for specific conductance per part per million mhos per cm. at 25 deg. cent.
Hydrogen, H^+	0.9921 ×	350.0×10^{-6}	$a = 347.2 \times 10^{-6}$
Calcium, Ca^{++}	0.0499 ×	60.0	$a = 2.99$
Magnesium, Mg^{++}	0.0822 ×	53.7	$b = 4.41$
Sodium, Na^+	0.0435 ×	50.9	$a = 2.21$
Bicarbonate, HCO_3^-	0.0164 ×	69.4	$d = 1.14$
Carbonate, CO_3^{--}	0.0333 ×	69.0	$c = 2.30$
Hydroxide, OH^-	0.0588 ×	192.0	$a = 11.29$
Sulphate, SO_4^{--}	0.0208 ×	79.0	$a = 1.64$
Chloride, Cl^-	0.0282 ×	75.5	$a = 2.13$
Phosphate, PO_4^{---}	0.0316 ×	71.0	$c = 2.24$
Nitrate, NO_3^-	0.0161 ×	70.6	$b = 1.14$

a "Handbook of Chemistry and Physics," Hodgman and Lange, Chemical Rubber Publishing Company, twelfth edition 1927, p. 777.
b "Electrochemistry," by Creighton and Fink, John Wiley and Sons, vol. 1, Table XXXIV, p. 134. (Values calculated to 25 deg. cent., from 18 deg. cent. by formula 19, p. 133.)
c Landolt and Boernstein Tables.
d No published data in literature. Authors have interpolated this value from ions having approximately the same ionic weight and valence.

APPLICATION OF FACTORS TO WATER ANALYSES

Corrected ions by analysis, p.p.m.	Factor for specific conductance, mhos per cm. per p.p.m.	Specific conductance, mhos per cm. at 25 deg. cent.
Calcium	$4.0 \times 0.0299 \times 10^{-4}$ =	0.1196×10^{-4}
Magnesium	2.5 × 0.0441 =	0.1103
Sodium	690.0 × 0.0221 =	15.2490
Carbonate	75.0 × 0.0230 =	1.7250
Hydroxide	105.4 × 0.1129 =	11.8997
Sulphate	802.2 × 0.0164 =	13.1560
Chloride	178.7 × 0.0213 =	3.8063
Total of calculated conductance		46.1×10^{-4}
Measured conductance		44.7

ductance measured after cooling the boiled water and preventing contact with CO_2. When calibrating the conductance meter by selecting the "best" supply at the station, an error may be unavoidably introduced because the quality of the supply may vary. It is convenient to use a "standard" water for calibrating the conductance meter in terms of dissolved salts, the calibration including a CO_2 correction.

Method for Developing CO_2 Corrections for Conductance

The procedure used in the development of the standard "synthetic" water eliminated the necessity of making a series of complete analyses of distilled supplies. The analytical errors introduced would be greater in magnitude than the actual dissolved salts in the water. The method used was to add increasing increments of dissolved salts to laboratory-distilled water, measuring the conductance, then adding varied amounts of CO_2 saturated water and remeasuring the conductance.

The "synthetic" water was made by averaging 20 analyses of Allegheny River water which include the yearly range in dissolved salts. Allegheny River water supplies three of the steam stations in Pittsburgh operated by the Duquesne Light Company and its subsidiaries. The data obtained are being applied to each station for steam-quality studies.

The necessary volume of the laboratory-distilled water was boiled to free it of atmospheric gases and then cooled, preventing the absorption of CO_2. Various volumes of the "synthetic" water were added to the CO_2-free distilled water to obtain increasing weights of dissolved salts, preparing a 1.1 liter volume of each concentration. The conductance of the solution was measured using the conductance flask, Fig. 5. The CO_2 was estimated in a sample removed from the flask (21). Another sample was saturated at room temperature with CO_2 gas released from a CO_2 tank. Approximately 5-cc. increments of the CO_2 saturated solution were added to the main solution. A conductance reading was observed, following which a 50-cc. portion was removed for estimating the CO_2. This procedure was repeated for the increase in concentration of CO_2 up to the desired maximum.

A curve was drawn for each increment of dissolved salts showing the relation between CO_2 and conductance. These curves were extended to zero CO_2. The conductance of the water at zero CO_2 was then plotted against dissolved salts added, and the curve extended to a conductance of 0.07×10^{-6} (the conductance of pure water at 25 deg. cent.) (22). This point represents zero dissolved salts. The negative of the dissolved salts indicated by this intersection is the concentration of the distilled water. In these data this value was 0.6 p.p.m. (It must be appreciated that accurate determination of this value by chemical analysis would be difficult.) This value represents the water correction, which is added to the data of column 2, Table 6, for construction of the final curve. The final curve so obtained is shown in Fig. 13, marked zero CO_2. From the first family of curves, values of conductance for even parts per million of CO_2 may be read and curves for various CO_2 contents constructed.

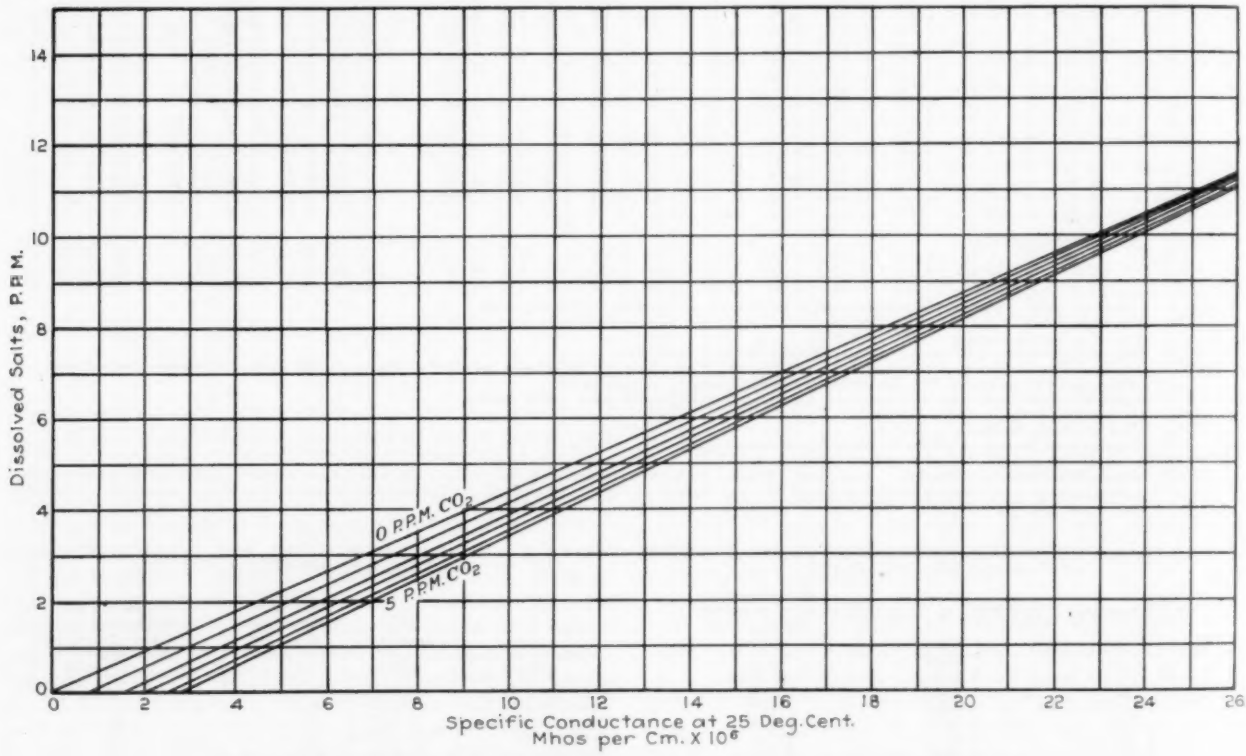


Fig. 13 Dissolved Salts in Feedwater and Correction of Conductance for CO_2

The curve shown in Fig. 13 was used to obtain the dissolved-salt values, column 8, shown in Table 7. The related data shown in the table support the view that the conductance method may be considered accurate for pure-water supplies, as in the chemical procedure unavoidable errors are introduced: first, dust inclusions while the water is evaporated, and second, the analytical errors in estimating small amounts of constituents.

Application of Conductance

In addition to the detection of condenser leakage, the conductance method is valuable for tracing and locating introductions of raw water at other points in the feedwater cycle.

One practical application of conductance equipment will be described in tracing the source of contamination in a large re-

turn system. Two steam plants of the Allegheny County Steam Heating Company supply the central business section of Pittsburgh with steam for heating purposes, and the extensive distribution system includes facilities for the return of the major portion of condensation to the boilers. It is essential that the returns be free of scale-forming contamination, for at one plant these returns provide 80 per cent of the feedwater, and at the other plant the returns will eventually amount to 65 per cent of the feedwater. Boiler outage for cleaning purposes cannot be scheduled during the six-month heating season, throughout which period the heat-transfer surfaces of the boilers must be kept free from scale. To insure this clean condition, a limited amount of treatment is added to the boiler water. Obviously if the contamination in the returns is held to a minimum, the treatment dosage may also be limited, and the generated steam furnished is clean.

The source of contamination is the domestic city-water supply, introduced into the returns by leaks in the customers' water heaters, since the city-water pressure exceeds the steam pressure used for this service. Conductivity recorders are installed at each steam plant, and are equipped with manually adjusted compensators for temperature and CO_2 correction. The recorded data are shown in terms of dissolved salts. The station chemist makes a daily test for CO_2 in the condensate and adjusts the CO_2 compensator. The station operator observes the temperature of the condensate once per shift, and adjusts the temperature compensator. The recorder is provided with alarm contacts which give both a bell and light signal when the dissolved salts reach the maximum setting, which is 25 p.p.m. dissolved salts. The station operator advises the maintenance division when the continuous record above 25 p.p.m. is noted, and a survey is started to locate the point of contamination. Samples of condensate are obtained at junctions of the various branches and the main return line. Measurements are made using the portable conductance bridge to identify the contaminating branch. Samples are then obtained at each customer's premises, and the leaking location is readily found. The customer cooperates by making the repair to the leaky coil, and until it is made the condensate at this location is diverted to the sewer.

In the usual instances where the surveys were made, the detection was made by the above procedure in a single working day. A system of detection lends itself to similar applications in the industrial field where process steam condensed may be returned to the boiler.

Carry-over of evaporator liquor into the vapor requires immediate correction if deposits in boilers are to be held to a minimum. The continuous observation for small amounts of carry-over by chemical means is difficult and time-consuming. The application of a conductance recorder for this purpose provides a visible record, and the operation of the evaporator can be immediately adjusted to eliminate the production of poor-quality vapor. Relatively large amounts of CO_2 are evolved in the evaporator due to the thermal decomposition of the bicarbonate and carbonate present in the evaporator feedwater. Correction must be applied to the conductance in order to obtain the dissolved-salt content. An installation of recorders has been made on evaporators at the Colfax power station for this purpose. The recorders are equipped with CO_2 and temperature compensators. The conductance cells are installed in receptacles through which a portion of the condensed vapor from the evaporator passes. The concentration of liquor is measured using a portable conductance bridge. This system permits a more ready control of quality than has been previously experienced.

The trend in recent boiler-plant practice is toward treatment of feedwater, even in distilled-water stations. Such water conditions demand a more intensive control of concentration in the boilers if entrainment of salts in the steam is to be held at a minimum. Respecting the apparatus, it appears that continuous blowdown systems equipped with heat exchangers are past the experimental stage. However, there is need for improvement in the methods of controlling concentrations within desired limits with the use of such equipment. Conductance methods are readily applicable to the problem of measuring concentration and steam quality. The measurements are easily observed provided adequate sampling facilities are available. The measurements may consist of "spot" readings if a portable conductance meter is used, or, where a continuous record is desired, the conductance of condensed steam and of boiler water may be recorded. The entrainment of salts in the steam is readily calculated from these data.

Sampling facilities are installed on the boilers at the Colfax station for obtaining steam and water samples, and the conductance data are being used for limiting entrainment and control of concentration. The portable conductance bridge and a flow cell are used for obtaining the readings.

It is possible to consider as a practical application a system of indicating or recording conductance apparatus combined with motor-operated valves automatically controlling concentration in steam-generating vessels. The recorder may show actual concentration. If alarm contacts are electrically connected to a controller, a motor-operated valve on a continuous-blowdown system may be regulated to maintain concentrations in the water between suitable limits. It is further considered that a system of automatic control can be applied to evaporators, whereby thermal conditions within the evaporator may be adjusted automatically to permit the production of clean vapor. In this application, conditions of back pressure and dilution of the liquor may be adjusted automatically by motorizing both the vapor-outlet and blowdown valves, the regulation to be made via a controller, energized when the vapor quality exceeds a set maximum value.

Conclusions

The conductance method has been successfully applied to boiler-room problems for observing entrainment in steam and concentration of salts in steam-generating vessels.

The accuracy of the conductance method equals that of chemical analysis when concentrated water supplies are tested. The accuracy of the conductance method if corrected for CO_2 is superior to that of chemical analysis when pure water supplies are tested.

The entrainment method described is shown to evaluate correctly the amount of carry-over from steam-generating apparatus. The speed and simplicity of obtaining conductance data recommends it as a valuable plant method.

Acknowledgment

Acknowledgment is made to the engineering personnel of the Power Stations Department of the Duquesne Light Company for the assistance lent in accumulating the calorimetric data, and for the critical review of the manuscript.

Appendix

Conductance Cell Constant. The cell constant of the conductance electrode should be known so that the observed conductance data will be comparable regardless of the cell used. The constant should be redetermined if the electrodes require cleaning or replatinizing. In commercial practice the electrode areas and spacing are not 1 sq. cm. (0.155 sq. in.) nor 1 cm. (0.3937 in.), respectively. However, the cell constant may be found by determining the conductance of any solution whose specific conductance has been exactly established. Generally a potassium chloride solution is used for this purpose. Creighton gives a

value of specific conductance for 0.1 N potassium chloride of 0.0128988 mhos per cm. at 25 deg. cent. (15). The 0.1 N solution is prepared by dissolving exactly 7.4556 grams of chemically pure potassium chloride in distilled water, in a 1-liter volumetric flask, and diluting the solution to exactly 1 liter. The solution should be prepared at 25 deg. cent. The cell electrode and a thermometer are inserted in a glass vessel, and a volume of the prepared solution is added to the vessel sufficient to submerge the electrodes. The temperature of the solution should be adjusted to 25 deg. cent. ± 1 deg. The observed conductance and corresponding temperatures are plotted, and the conductance at 25 deg. cent. is read from the graph. The value is substituted for c in the equation, which is solved for K.

$$K = \frac{k}{c}$$

where K = cell constant
k = specific conductance of 0.1 N potassium chloride at 25 deg. cent.
c = measured conductance of 0.1 N potassium chloride at 25 deg. cent.

If the conductance meter measures resistance, the reciprocal of the reading should be used for c.

Note 1. Two standard temperatures, 18 and 25 deg. cent. appear in the literature. The 25-deg. standard more nearly approximates room-temperature conditions. This temperature appears in recently published methods.

Note 2. When out of service the electrodes should be submerged in distilled water. If the electrode surfaces have been allowed to dry, they should be rinsed with denatured alcohol and distilled water before using. When first using electrodes, or after replatinizing, a period of 30 min. should elapse before readings are taken.

Note 3. If the electrodes have been contaminated, they should be replatinized. A 3 per cent platinic chloride solution with $^1/_{40}$ per cent of lead acetate is used for platinizing. The electrodes are immersed in the solution and the leads connected to a dry cell supplying sufficient voltage to cause a barely appreciable evolution of gas. The polarity is reversed frequently and the operation continued for 3 to 5 min. until the electrodes are both covered with a velvety black deposit.

Temperature Correction. The specific conductance of aqueous solutions increases with temperature rise. In practice, an approximate temperature correction may be applied, since it is not always convenient to measure the conductance at the 25-deg. cent. standard. Any conductance (L_m) measured at a temperature t_m may be calculated to a standard temperature t_s by substitution in the following formula:

$$L_s = \frac{L_m\,(t_s + K)}{t_m + K}$$

If temperatures are measured in degrees centigrade, K = 20.6; if in degrees fahrenheit, K = 5. The standard temperature used herein is 25 deg. cent. or 77 deg. fahr. The coefficient 2.1 per cent per deg. cent. or 1.2 per cent per deg. fahr. may be used for small temperature differences (10).

Conductance Calculations. Reference is made in the paper to the calculation of conductance from the ions found in water. The factors for converting the parts per million to the specific conductance values are given in Table 9, and an example of the calculation is also shown.

Calculation of Boiler-Water Concentration and Entrainment From Chemical Analyses. The development and application of formulas for checking the rate of concentration, and of entrainment permit verifying the actual concentration, which is shown plotted in Fig. 10, and steam quality and entrainment, columns 6 and 7, respectively, in Table 2.

Rate of Concentration of Boiler Water.

Let F = total boiler feed in lb. × 10^6
C = concentration of boiler water in p.p.m.
c = concentration of boiler feedwater in p.p.m.
K = holding capacity of boiler in lb. × 10^6
E = entrainment as fraction of F.

Then

$$\frac{dC}{dF} = \frac{c - CE}{K}$$

and

$$\int_{F_1}^{F_2} dF = \int_{C_1}^{C_2} \frac{K\,dC}{c - CE} = \frac{K}{E}\int_{C_1}^{C_2} \frac{dC}{(c/E) - C}$$

Then

$$F_2 - F_1 = \frac{K}{E}\log_e (c - C_1E) = \frac{K}{E}\log_e (c - C_2E)$$

$$F_2 - F_1 = \frac{K}{E}\log_e \frac{(c - C_1E)}{(c - C_2E)}$$

Changing to $\log_{10}$ and transposing,

$$E(F_2 - F_1) = 2.303K \log \frac{(c - C_1E)}{(c - C_2E)} \quad \ldots\ldots\ldots [1]$$

Case 1. To Determine if Blowdown Is Necessary to Limit Concentration. It will be appreciated that at infinite evaporation (assuming no scale is deposited) the weight of solids entering the boiler must equal the weight leaving it entrained in the steam, or

$$c = C_2E \quad \text{and} \quad C_2 = \frac{c}{E}$$

This may be derived mathematically as follows: Equation [1] may be written

$$c - C_2E = \frac{c - C_1E}{10^{\frac{E(F_2 - F_1)}{2.303K}}}$$

The limit of the right-hand side of the equation as F approaches infinity is zero, hence when F equals infinity,

$$c = C_2E \quad \text{and} \quad C_2 = \frac{c}{E}$$

If substitution in this formula leads to a value of C_2 below the maximum allowable value, no blowdown is necessary. Example (Case 1):

c = 2.08, from Table 7, average of column 8
E = 0.35 per cent = 0.0035 from Table 2 average of column 7

$$C_2 = \frac{2.08}{0.0035} = 594.3 \text{ p.p.m.}$$

From experience, the maximum allowable concentration of boiler water has been empirically set at 4250 p.p.m. Hence no blowdowns are necessary for control of dissolved salts. Blowdowns may be made if suspended solids are present in the water, provided the boiler has been banked. The purpose of the blowdown is to prevent the sedimentation on the lower tube-bank surfaces.

Case 2. Change in Concentration With No Blowdowns.

$F_2 - F_1$ = Evaporation
c = C_1
F_1 = K

Equation [1] then becomes

$$\frac{E(F_2 - F_1)}{2.303K} = \frac{E(\text{evap.})}{2.303K} = \log \frac{c(1 - E)}{c - C_2E} \quad \ldots\ldots [2]$$

Example (Case 2): The concentrations calculated at various evaporations with no blowdowns, using Equation [2], are shown in Table 8, and these values are plotted in Fig. 10. The values used for the calculations are:

c = 2.08, from Table 7, average of column 8
K = 0.12 million lb., from Table 1, item 14
E = 0.35 per cent = 0.0035, from Table 2, average of column 7.

The steps in the calculations are illustrated when calculating the concentration for the evaporation of 10,000,000 lb. Substituting in Equation [2],

$$\frac{0.0035 \times 10}{2.303 \times 0.12} = \log \frac{2.08(1 - 0.0035)}{2.08 - C_2 \times 0.0035}$$

$$0.1266 = \log \frac{2.08(1 - 0.0035)}{2.08 - C_2 \times 0.0035}$$

$$1.3384 = \frac{2.08\ (1 - 0.0035)}{2.08 - C_2 \times 0.0035}$$

$$C_2 = 151.6$$

Case 3. Change in Concentration With Continuous Blowdown. By substituting in Equation [2], the maximum allowable value for C_2, F_2 can be calculated. Blowdown should commence at this point.

Let B = blowdown in lb. $\times 10^6$
C_3 = concentration of boiler water after blowdown and refilling to original water level.

Then

$$C_3 = C_2 \frac{K - B}{K} + c \frac{B}{K}$$

Substitute C_2 in Equation [2] and recalculate F_2; call this value F_3. Then, $F_2 - F_3 + B$ = amount of water (in lb. $\times 10^6$) that may be fed to the boiler before the next blowdown. Since each succeeding cycle will be identical, this is all the calculation required.

Case 5. Scale or Sludge Formation. In case scale or sludge is being formed in the boiler, the calculated value of C_2 will exceed that actually attained. In this case c should be decreased by the concentration of scale forming materials in the boiler feedwater.

Case 6. Change in Water Level. Calculate C_2 using Equation [2]. Let K be holding capacity at the original water level, and K_1 be the holding capacity at the new water level. Let C_3 be the concentration at the new water level. Then

$$C_3 = C_2 \times \frac{K - K_1}{K}$$

The various holding capacities of the boiler must be calculated for corresponding water levels from the boiler dimensions.

Note: In general, knowing any two of the quantities, c, C_2, and E_1 the third quantity may be calculated for any value of F_2 using Equation [2]. It should be noted that very slight errors in the determination of E (or of c if the boiler feed is of high purity) will make large differences in C_2.

It should also be noted that the equation leads to a value of C which assumes uniform concentration in the boiler. Since the concentration at various points in the boiler varies considerably (18), sampling arrangements must be made to secure as near an average sample as is possible.

Bibliography

(3) "Corrosion of Iron," by W. R. Whitney, *Industrial and Engineering Chemistry*, vol. 17, no. 4, April, 1925, pp. 385–389.

(4) "The Examination of Water by Electrical Methods," by W. Pollard Digby, *The Electrician*, London, England, July 29, 1910.

(5) "Simplified Chemical Method of Detecting Surface-Condenser Leakage," by W. E. Caldwell, *Power*, vol. 54, no. 4, July 26, 1929, pp. 141–142.

(6) "The Electrical Method of Detecting Surface-Condenser Leakage," by W. E. Caldwell, *Power*, vol. 54, no. 6, Aug. 9, 1921, pp. 217–219.

(7) "Measurement of Surface-Condenser Leakage by Electrolytic-Conductivity Method," by Earl A. Keeler, *Power*, vol. 35, no. 4, Jan. 24, 1922, pp. 126–128.

(8) "Electrolytic Method for Testing Condenser Leakage," by A. E. Grunert, *Electrical World*, vol. 89, no. 26, June 25, 1927, pp. 1384–1385.

(9) "Measuring Condenser Leakage With a Megger," by H. E. Edwards, *Power*, vol. 60, 1924, pp. 907–908.

(10) "Conductivity Methods of Measuring Condenser Leakage," by Henry C. Parker and Willard N. Greer, *Power*, Sept. 27, 1927, pp. 476–481.

(11) "Electrical Measurements," by P. A. Borden, American Institute of Electrical Engineers Journal, April, 1925.

(12) "Test Code for Condensing Apparatus," *Mechanical Engineering*, vol. 46, 1924, pp. 291–296.

(13) "Condenser Leakage Tests," *Mechanical Engineering*, November, 1925.

(14) Prime Movers Committee Publications, National Electric Light Association:

a Pub. No. 67–12, Jan., 1927, pp. 7–8 and 51.
b Pub. No. 278–65, July, 1925, pp. 15–16.
c Pub. No. 278–78, July, 1928, pp. 19, 22, 29 to 31.

(15) "Principles and Applications of Electrochemistry," by H. J. Creighton, vol. I, John Wiley and Sons, 1924.

(16) "Determination of Moisture in Steam by Electrical Conductivity," by M. E. Fitze, *Power*, Sept. 18, 1928, pp. 484–485.

(17) "The Effect of Rating on Boiler Water Samples," by J. N. Evans, *Power*, Oct. 1, 1929, pp. 520–521.

(18) "Corrosion, Causes and Prevention," by F. N. Speller. McGraw-Hill Book Co., 1926, pp. 403–404, Fig. 99.

(19) Kendall, Journal American Chemical Society, vol. 38, 1926, p. 2460.

(20) Power Stations Chemistry, Part II, Prime Movers Committee, N.E.L.A., Publication No. 289–114, August, 1929, pp. 3–11, p. 13.

(21) Standard Methods of Water Analyses, American Public Health Association, Sixth Edition, 1925, Section XVI, item 7a, pp. 36–37.

(22) See reference 15, page 83.

Discussion

Sidney Cornell.[4] The theory of conductance depends upon the ions dissolved, and the accuracy of the measurements gets back on final analysis to the determination of total solids in distillate, condensate, or mixed feed. Some serious errors can occur in the assumed simple matter of this determination of total solids by direct evaporation of a liter of distillate and subsequent weighing.

The accuracy of measurement of conductance is high, a calibration to an accuracy of 1 per cent being mentioned by the authors, but in the simple matter of total solid determination errors often of startling amount may be encountered, thereby throwing the accuracy of the entire measurement out.

Calculations of per cent of entrainment or of carry-over contamination can readily be made from the data, as illustrated by the authors, but for calibration or for calculation of total solids in any of the liquids measured by conductance a determination of total solids by direct weight must be made.

[4] Consulting Chemical Engineer, Griscom-Russell Company, New York, N. Y.

Unfortunately no standardized methods have as yet been proposed for this important matter. Serious contamination can occur in distillate samples in the sampling, in the holding, and in the evaporation to dryness. More serious errors can be encountered in the every-day matter of weighing. The make-up of synthetic waters in turn is exposed to marked errors from the same sources, and the following may be taken as a tenative standard for the determination of total solids in distillate.

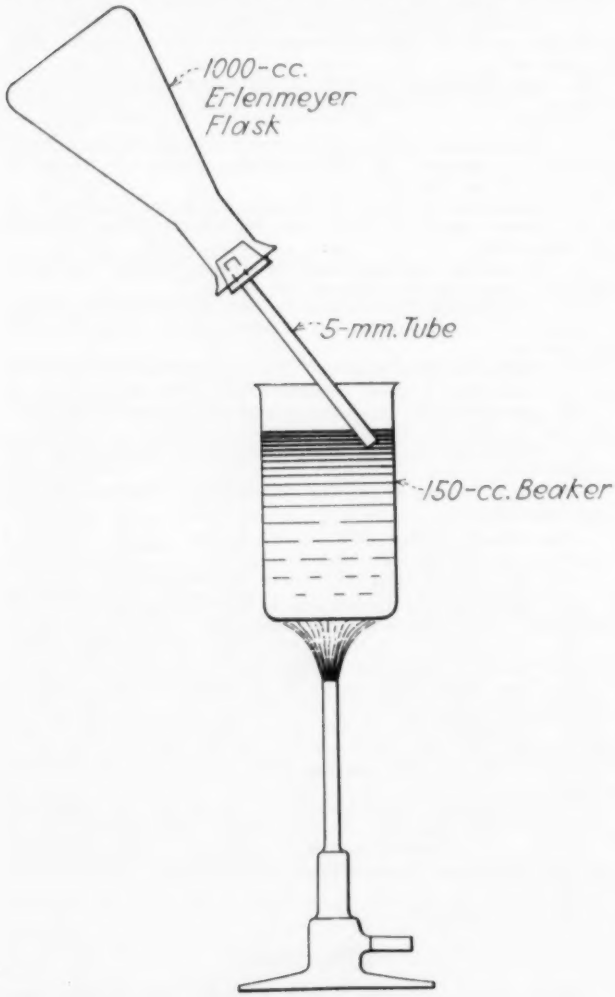


FIG. 14

Distillate from evaporators may run to as low as 1 p.p.m. total solids and to range upward to as high as 10 p.p.m. for condensate, and the laboratory evaporation of a liter sample may take four or five hours to dryness. A liter sample can be taken as a standard quantity for this test; no smaller because of the measurement by weight involved, and no larger because contamination from outside sources will upset the advantage. Hence an Erlenmeyer Pyrex 1000-cc. flask calibrated and marked should be made the sampling vessel. Fit to this flask a XXXX cork with single hole, and 5-mm. glass tube 5 in. long. Avoid a rubber cork as contamination will always occur, and a glass stopper cannot be fitted to this size neck. This sample is inverted over a 150-cc. Pyrex beaker, the sample flowing down until the end of the tube is covered and the beaker is on the hot plate. Obviously continuous feed takes place, with no possibility of outside contamination, as might occur with beaker transfer. Avoid all washing with distilled water, as often laboratory distilled water contains 2 p.p.m. total solids. If condensation takes place on the Erlenmeyer flask and tends to flow back into the beaker, place the flask at an angle, as shown in Fig. 14, supporting it on the chemical stand.

Naturally the sample flask, the beaker, and the tube must be chemically clean to start with and the beaker be correctly weighed.

A 150-cc. beaker will weigh approximately 50 grams. Very few laboratories possess correct weights; in fact, daily usage causes a rapid deterioration in their accuracy. Further, heat, light, draft, etc., affect the accuracy of a chemical balance. It is admitted that a few hours' time in the weighing of assays causes a marked variation. Thus if it were possible the sample should be evaporated instantly to dryness and the beaker be weighed before and after. Obviously this is impossible. To avoid error it is very necessary to weigh the sample beaker by the tare method, using a metal dish with sand or shot therein and getting final balance with the rider. The rider type of balance is superior for this sort of work, and the rider must be correct, and one with Bureau of Standards certificate is probably the safest to use. The beaker should have been heated on a hot plate or in an oven to 125 deg. cent. for 10 min., placed in a desiccator until cool, and weighed. When the sample is evaporated to dryness, the beaker should again be brought to 125 deg. cent. for 10 min., cooled in the desiccator, and weighed, the rider over the beam being the only weight difference. In fact a 1 mg., a 5, and a 10 are the weights used.

Obviously the balance case must be dry, it must have the usual pot of sulphuric acid or other dehydrating means therein, and it should be equipped with a thermometer. The volume of a 150-cc. beaker is in itself one to impose some variation by temperature alone, and as 25 deg. cent. is mentioned on the third page of the paper under discussion as the standardization point for the cell, this might be taken for the weighing.

The writer has noted errors of 100 per cent by different chemists in the matter of total solids in distillate, and when full appreciation of the fact that a measurement by direct weight of 1 part in 50,000 is necessary, some betterment in technique may be evolved. That is, the beaker mentioned weighs 50 grams. The total solids in 1 liter of distillate may be as low as 1 p.p.m., or 0.001 gram, and of course weighing on the usual chemical balance to 0.0001 gram.

In any event, no sample should be allowed to rest overnight once evaporation has started, as dust is always falling, and it takes very little contamination to impose wide errors.

The effect of gases, their measurement in small quantity in turn, is a difficult matter, but this matter of weighing of total solids alone requires unusual care to be correct. Based on this correctness rests the calibration of the conductance measuring instrument.

JOE A. BERGIN.[5] The authors have treated the subject of conductance of solutions very thoroughly in this paper, and have given much consideration to applying corrections to make the data presented comparable. When the determinations have been made in a thoroughly scientific manner and the slight corrections for pure water and dissolved carbon dioxide have been applied, the results are valuable in steam-power-station practice. There must be, however, proper precautions used in performing the conductance tests.

Conductance measurements for determining condenser leakage are now quite generally used and are reliable for the purpose. Using the conductance method for determining moisture in the steam and the carry-over or entrainment of mineral salts does

[5] Superintendent, Philo Plant, Ohio Power Company, Philo, Ohio. Mem. A.S.M.E.

not present any unusual difficulties after certain constants have determined and applied. The conventional calorimeter for measuring moisture in the steam is far from dependable, and there is a dire need of a more reliable method since the advent of high steam pressures and boilers operating at increasingly higher ratings.

The control of boiler and evaporator concentrations presents no great difficulties. The conductance method has been in use some time for determining the time for blowing down. It does not seem impossible to control motor-operated blow-off valves by the conductance of the boiler or evaporator water and to automatically control the concentration.

To properly use and apply the electrical conductance measurements in any steam-power plant it is necessary to do certain preliminary work, such as preparing graphs for the various conditions of the water used. The graphs prepared for one plant cannot be used in another, and in some plants the characteristics of the water change from day to day with the wind and weather.

The results obtained by the authors compare favorably with those obtained by the time-consuming methods now in use and considered reliable. The method may seem to be a great deviation from standard power-station practice, but electrical conductance measurements may be quickly made and dependable results obtained wth the use of proper equipment and careful sampling.

BERT HOUGHTON.[6] During the past few years carry-over in steam has demanded an increasing interest. This has been due to the increased size of boiler installations and more particularly to the increased steaming rates per unit of steaming area. At these steaming rates, with consequent increase of violent ebullition in the steam drum, greater difficulty has been experienced in designing efficient steam baffles to effectively prevent carry-over. Several cases are on record in which carry-over has been sufficiently serious to cause turbine shutdowns.

It has been customary to determine the quality of the saturated steam leaving the boiler by a throttling calorimeter and to estimate carry-over from the steam quality obtained. This method is indirect, and usually the indications thus obtained are not as high as the carry-over actually present in the steam. The development of a more direct method for the determination of carry-over is therefore timely.

The proposed method for obtaining "entrainment in per cent by weight of boiler water in steam" necessarily assumes an accurate boiler-water sample. The concentration of dissolved solids in boiler-water samples taken simultaneously at different points in a boiler varies widely, at times as much as 400 per cent. Percentages based on data as inaccurate as boiler-water samples can hardly give reliable information unless sufficient points on the boiler are taken to insure comparatively average boiler-water concentration. The writer does not feel that it is necessary to convert "entrainment in steam" to "entrainment in per cent by weight of boiler water in steam" and can only view such a procedure as introducing an unnecessary error.

In Fig. 11 the authors present data indicating that entrainment in steam diminishes as boiler-water concentration increases. This is contrary to fact in ordinary boiler practice. Many investigators have shown that carry-over not only increases but increases rapidly with increasing boiler-water concentrations. The writer believes that if the boiler-water concentration had carried higher than the 600 p.p.m. indicated, the evidence would have been unmistakable and the apparent errors avoided.

The authors make a correction for carbon dioxide in steam. A daily analysis is made and the CO_2 compensator set accordingly. Such a procedure assumes that the concentration of CO_2 and other entrained gases is constant over a period of time. Gases usually found in steam, that is, air and carbon dioxide, vary materially in concentration, thus making accurate compensation difficult. To make the conductivity method reliable, means should be developed to minimize the entrained-gas correction.

The conductance method is applicable to the measurement of dissolved solids in the steam. With the feed-treatment methods in use in many boiler plants, suspended as well as dissolved solids are present in boiler water. These suspended solids are entrained with the steam, and from a practical standpoint will cause the same clogging in superheaters, valves, and turbine blades as the dissolved solids which the authors have determined by the conductance method.

In the appendix, formulas have been developed to calculate the solids in the boiler, and it is brought out that a point of maximum is reached in boiler-water concentration, solids being entrained with the steam as rapidly as they enter with the feedwater. There is another agency that prevents the building up of boiler-water concentrations, and that is "leaky" blowdown valves. The writer has seldom heard of cases in which boiler water concentrations built up but that a "leaky" blow-down valve was involved.

The writer wishes to commend the authors on their work and believes that this conductance method, so long in use for condensates and feedwaters, may yet be developed to the point where it will be a standard method for the determination of carry-over in steam.

J. A. HANCE.[7] The writer has had no experience with the apparatus described, but has reviewed the paper and has also conferred with our electrical and chemical departments and others, and submits the following comments.

Our chemical laboratory makes analyses, from time to time, of various boiler feedwaters. These analyses are made in accordance with conventional methods, and the results obtained show the principal elements that are present, but they have no definite way of determining how these elements are combined in the water. Similar samples sent to other laboratories show divergent results.

Likewise calorimeter tests made to determine the quality of the steam frequently show widely different results.

The methods proposed in this paper appear generally to be based on sound principles, and the writer is of the opinion that the method is of value in determining the dryness of steam, but he is not certain as to its value in determining the suitability of certain waters for boiler use.

It appears that the electrical conductance of water would vary with each element present and also with the quantity of each element that may be present. If one were to consider only the elements that are listed in Table 5 and consider each element varying in quantity from zero to a maximum by small increments, he would obtain an infinite number of combinations, many of which would probably show the same electrical conductance. It would also appear that certain waters may show a satisfactory electrical conductance, but may contain elements that may have corrosive or scale-forming influences which would render them unfit for use.

It is not clear how steam-generating plants could correct the water without resorting to conventional chemical analysis. It would require very careful or trained observers or physicists to make the corrections and the calculations referred to.

H. F. JOHNSTONE.[8] The device which the authors have per-

[6] Operating Superintendent, Brooklyn Edison Company, Brooklyn, N. Y. Mem. A.S.M.E.

[7] Baldwin Locomotive Works, Philadelphia, Pa.

[8] Special Research Associate in Chemical Engineering, University of Illinois, Urbana, Ill.

fected will find a large use, not only in determining carry-over in steam boilers and following impurities in return lines, but also in the control of chemical processes in which there may be a change of conductance due to the reactions involved. The writer is most interested in the observations that the authors make in which "it appears that the entrainment in steam is greater in magnitude for the lower steam-flow conditions than is found under the normal steam output of the boiler," and also, "the steam quality apparently improved with the increase in concentration of boiler water and total evaporation, . . . for the observed case some condition existed in the more dilute water which is unfavorable to the production of dry and clean steam." A systematic study of the steam produced by a boiler by means of the conductance method may lead to some valuable information about foaming, of which so little is known at present. The writer wishes to congratulate the authors and the Society on this very useful paper.

M. D. Baker.[9] The conductivity bridge has proved to be a valuable asset for a comparative study of water and water conditions in central power stations. After a standard cycle has been established so that all the salts in the water are comparatively the same, a very close check can be maintained on regulation of treatment needed. In the cycle at the Springdale Station of the West Penn Power Company, the contamination is mostly Allegheny River water through condenser leakage, and the treatment is trisodium phosphate and sodium hydroxide. The carry-over with the steam and the evaporated make-up is carefully watched so as to have as pure a water as possible in the feedwater cycle. By measuring the conductivity of the water supplies, a control has been established so that a weekly check on boiler waters has become a matter of routine.

We have found that the conductivity bridge can be used with very accurate results only on waters containing less than 20 p.p.m. total solids. When the concentration is above this, results are varying and erroneous and can be used only comparatively. Due to the presence of various ions in the solution, the degree of ionization may vary with the preponderance of any one, and tabulated results have shown such a wide discrepancy that the bridge cannot be considered an accurate indicator of conditions when concentrations are high.

During a protracted study of boiler-water concentrations, in which over 1100 determinations were made, the parts per million by evaporation were plotted against conductance in mhos $\times 10^{-6}$; the resulting curve sheet was a mass of dots with absolutely no relationship between the various boilers. During this test, control conditions were as near ideal as was possible to maintain. Similar results were obtained in an effort to check on parts per million in evaporator concentrate. It was noted that a curve accurate to within 10 per cent could be drawn for each run of a boiler or evaporator, but that this curve would not be satisfactory for any other piece of apparatus and would not be satisfactory for the next run of the same boiler or evaporator.

Entrainment tests were run on a boiler similar to the one described in the article, but it was found necessary to collect entrainment samples and take calorimeter readings on each saturated steam lead. This was caused by the variation in concentration of the two ends of the boiler. Due to the inaccuracy of conductance reading at the higher concentrations, a 1 per cent solution of the boiler water was used. After making all necessary corrections, more consistent results and closer checks as to the moisture content determination of the calorimeter were obtained. Tests were run on boilers having boiler-water concentrations varying from 600 to 1250 p.p.m.

Due to the erratic results obtained with higher concentration in a solution, the writer has been unable to compare any analysis with conductance units. Some comparison has been obtained by using a 1 per cent solution and multiplying this result by 100. An example of this is one sample of Allegheny River water which had a conductance of 210×10^{-6} mhos, a 1 per cent solution of the same having a conductance of 2.56×10^{-6} mhos. Theoretical conductance based on analysis was 249×10^{-6} mhos. This is the closest agreement between theoretical and actual conductance which the writer has ever been able to obtain.

Conclusions: The conductivity bridge is a valuable instrument in tracing and determining contamination and as a guide in controlling feedwater conditions. It is also of value as a method of determining entrainment and for obtaining comparative data when concentrations exceed 20 p.p.m. Its great advantages are speed and the fact that the operators can use the instrument and obtain comparative results.

E. B. Powell.[10] The paper is of special interest at this time, and the writer would appeal to the authors to continue the work which they have so capably taken up.

With the growth in use of high-pressure steam turbines and engines of so-called "back pressure" type, exhausting to industrial process systems having no return or but small return of condensate, steam contamination by entrainment becomes of increasingly serious concern. In plants of this type economic considerations will usually dictate chemical treatment of the boiler-feed make-up, rather than distillation of make-up, wherever the available water supply is susceptible of the required degree of "purification" by chemical means. In general, this will result in much higher solid content in boiler feed, and in consequence much higher rate of concentration of dissolved and suspended solids in the water within the boiler, than occurs in the more conventional type of utility power plant. Moisture entrainment in steam from the boilers of this newer type of plant must therefore be subject to even closer control.

The work which the authors have described and that covered by earlier papers mentioned in their bibliography has been devoted primarily to the development of a mechanism for observation and measurement of entrainment. Obviously this can be looked upon only as an introduction to a larger study. From the laboratory aspect Prof. C. W. Foulk has made admirable progress in the development of the theory of priming. The testing of the elements of his theory by the criterion of the operating boiler plant is still to be undertaken, and, in fact, although priming is one of the oldest problems associated with the steam boiler, the field remains but scantily explored. The authors have made a most valuable contribution in preparing the ground for further analysis. The data of their paper suggest numerous lines of attack leading toward a more thorough understanding of the phenomenon. It is to be hoped that, having carried the work to the extent which they have, the authors may be persuaded to continue their studies and investigations at least to the point of clearing the many elements of uncertainty to which they call attention, such as the increase of entrainment at low rates of boiler steaming, which would seem to cast considerable doubt upon current methods of steam calorimeter calibration and the wide departure of the individual conductivity and calorimetric determinations from the mean relationship indicated by the average of an extensive series of such observations.

E. L. McDonald.[11] The writer's first experience with the

[9] Chief Chemist, West Penn Power Company, Springdale, Pa.

[10] Consulting Engineer, Stone & Webster Engineering Corporation, Boston, Mass. Mem. A.S.M.E.

[11] Efficiency Engineer, Kansas City Power and Light Company, Kansas City, Mo. Mem. A.S.M.E.

electrical conductance method of measuring water contaminations dates back to 1919, when current at a fixed voltage was passed through two carbon electrodes (with fixed distances and exposures) immersed in glass jars of condensed steam, condensate, and river water, giving relative conductivities and, in turn, condenser leakage (properly corrected for temperature). Curves plotted for various temperatures and dilutions facilitated the determinations.

Since 1927 the writer has experimented with electrical recorders, using this same principle and for the same purpose.

More recently he has experimented with the same apparatus and also with the portable conductance bridge type for determining carry-over on 1400-lb. boiler units. The portable instrument is in daily use for determining boiler-water concentrations.

Much of the writer's experience substantiates the findings of the authors; however, in attempting to determine the carry-over on the 1400-lb. boiler units, an inexplainable and as yet unsolved problem has been met which throws doubt into the value of such a method where higher pressures and temperatures are concerned, at least until more experimental work is conducted.

Very briefly, the situation is this:

The conductivity method has been used and proves reliable for determining soluble salts in various waters throughout the plant. Solutions of known quantities have been used to check its accuracy, and evaporative tests have been made on a great number of boiler-water samples to also prove this.

In attempting to determine carry-over in the 1400-lb. boiler units, equipment was installed similar to that used by the authors (A.S.M.E. steam sampling tube and condenser coils), and a great number of both steam and water samples were taken.

Solids were determined on all samples by both the electrical-resistance method and by carefully conducted evaporative tests.

Both of these methods checked reasonably close on the 1400-lb. boiler water, but in no case could a check be obtained on the 1400-lb. steam sample. The total solids in solution by actual weight averaged about seven times the amount indicated by the resistance method. The per cent carry-over of solids by actual weight agreed closely with that indicated by relative M.O. alkalinity.

Space and time will not permit presenting details, but the conclusions drawn to date are:

The electrical-resistance method is based on the complete ionization of solids in solution. Practically all solids in this boiler water were in solution and apparently completely ionized. The check against actual evaporative tests verifies this.

Some chemical change apparently takes place when converting water to steam at high pressures which decomposes or changes the relative conductivity of the constituents in the boiler water (aside from CO_2); otherwise the electrical method, which checks the solids in the boiler water, would also check on the steam samples, which should contain the same solids.

The electrical method appears to indicate the approximate per cent moisture carry-over, but not the total solids carried over. Calorimeter determinations, while not entirely reliable at this pressure, showed moisture carry-over in the same region as the electric figures.

Chemical methods (evaporative tests and M.O. alkalinity) give total solids in solution, but not per cent moisture carry-over.

On the tests in question, there was a rather definite difference between total solids in solution and approximate per cent moisture carry-over, the former being about seven times the latter, but to date the definite reason for this great difference is unaccounted for.

Some additional experiments are being carried out as time permits, which may clear this point, and it is hoped that some comments or suggestions may be evolved from the discussion submitted that will be of value.

In the meantime it is suggested that if the electrical-conductance method is to be used for determining solids or moisture or carry-over on high-pressure equipment, it should be checked against the weight as determined by an evaporative test, which in the final analysis (assuming the sampling is representative) is the actual amount of solids leaving the boiler drum.

No attempt is made to present the vast amount of data collected during these experiments, but merely to outline briefly the difficulty encountered and the final deductions to date.

W. N. Greer.[12] The paper has described the measurements in terms of specific conductances, and it might be well to elaborate on the value of making or, at least, of tabulating conductivity measurements in terms of specific conductances. The specific conductance is equal to the measured conductance multiplied by the cell constant and/or the specific conductance is equal to the cell constant divided by the measured resistance. Since the cell constant of the conductivity cell is a function of the area of the electrodes and their distance apart, it is at once evident that conductivity cells of the same design will seldom have the same cell constant, due to small variations in manufacture, especially those made in the hands of the glass blower. Tabulating the results, then, in terms of specific conductance makes the results comparative and independent of the conductivity cell used.

The modified design of the flow-type conductivity cell described in the paper is no doubt an improvement over that previously described by Messrs. Parker and Greer and referred to in the paper. The improvement is primarily concerned with the position of the electrodes in the flow chamber, since in this new design the solution flows more or less directly through the electrode proper, immediately subjecting it to changes in the solution, whereas in the previous design the solution had a tendency to flow around the electrodes, and the latter was not immediately subjected to small changes of the solution.

Authors' Closure

Questions of like nature have been developed in the discussions presented; hence the authors will answer them as a group rather than directing their reply specifically to the separate discussions. Because of the interest expressed in the discussion and of the value of the supplemental data obtained subsequent to the submission of the manuscript, the authors take the liberty in answering the questions of including the additional data in their closure. In general, the discussion is directed to sampling, standardization of the method, and the interpretation of the conductance data.

Concerning the physical features involved in the boiler-test set-up, the importance of adequate sampling facilities was sufficiently stressed to the engineering personnel, and facilities were provided for sampling both steam and boiler water. In the case of boiler unit 22, steam sampling nozzles were properly located in the two steam leads from the drum to the superheater inlet manifold. Because of previous experience, and the background of the boiler-water traverse test (see reference item 18) boiler-water samples were not secured from the gage-glass water columns as such samples would not be representative of the average maximum concentration in the drum. It was observed that a sampling line installed in and extended across the length of the drum produced samples showing correct concentration of boiler water in the drum. In the traverse referred to, this observation had been checked by installation of a line in the drum as described, and in addition a sample nipple was located at each fifth header-tube cap in the 52 uptake water headers extending across the

[12] General Division Sales, Leeds & Northrup Company, Philadelphia, Pa.

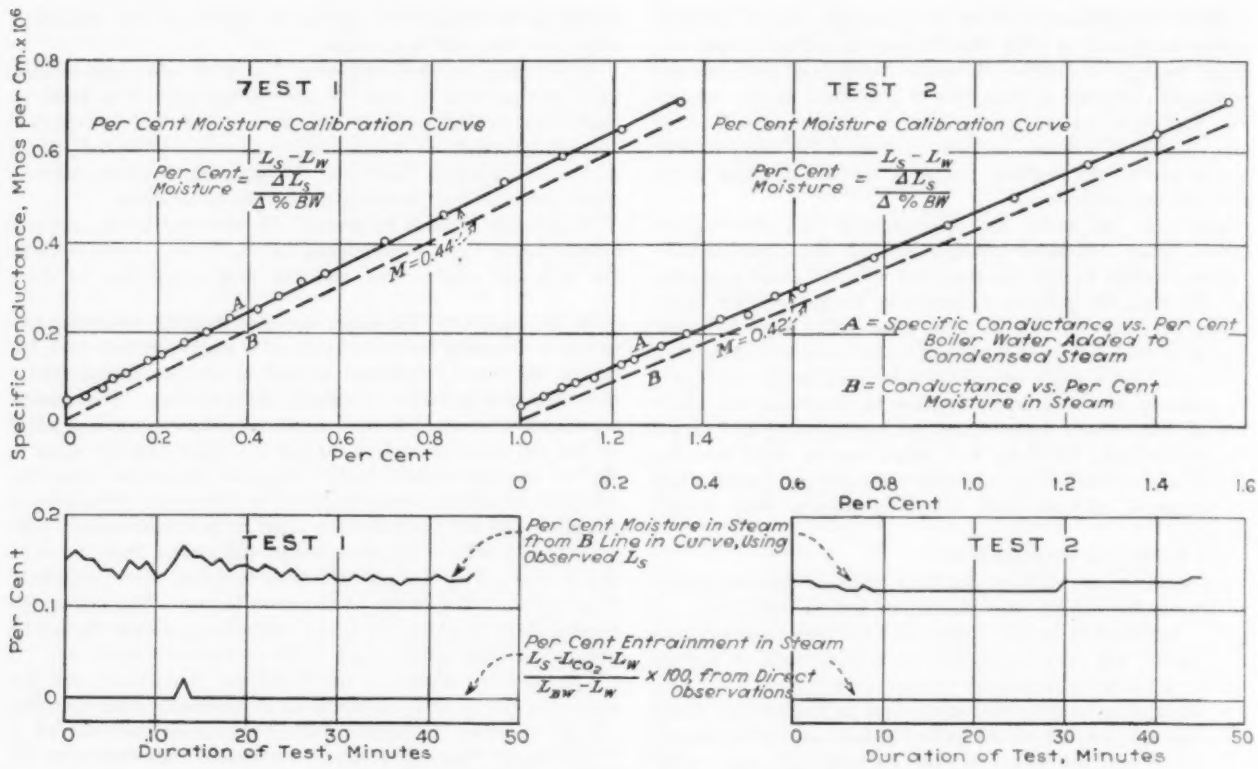


FIG. 15 COMPARISON OF CONDUCTANCE METHODS FOR STEAM QUALITY

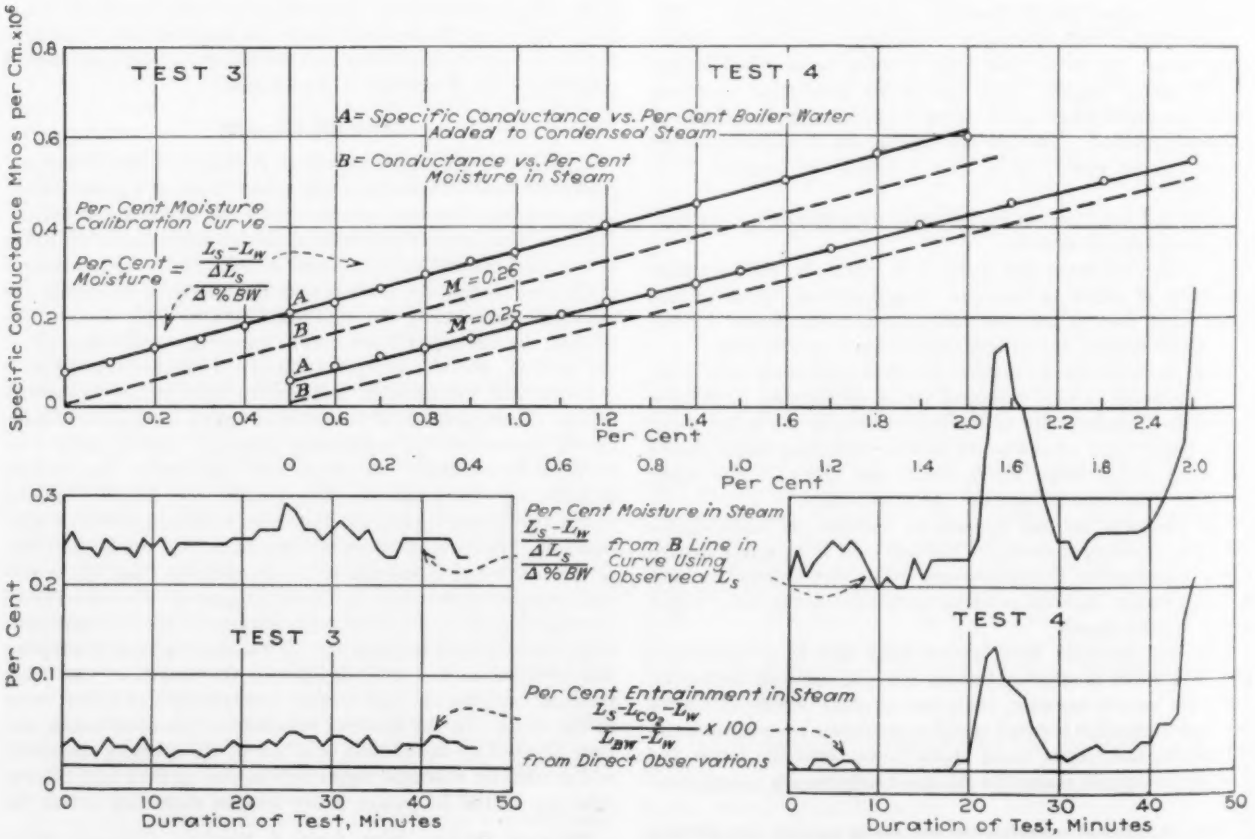


FIG. 16 COMPARISON OF CONDUCTANCE METHODS FOR STEAM QUALITY

boiler. Each of the individual lines was connected to a separate cooling coil. The concentration of boiler water was measured by the conductance method. Samples were taken hourly for 24 hours. The comparisons of concentrations are based on the averaged results for the test period. The average concentration of boiler water from the drum sampling line was found to measure 6.6×10^{-6} mhos per cm., a difference of but 11 per cent from the average concentration of the 10 header samples, 7.3×10^{-6} mhos per cm. It was observed, however, that the location of the two feedwater inlet pipes influenced the concentration of boiler water in the drum. For example, at the drum center the concentration was observed to be at a maximum, 8.5×10^{-6} mhos per cm., 55 per cent higher than the minimum concentration of 5.5×10^{-6} mhos per cm. found directly under the feedwater distributor. For obtaining the data used in the development of the entrainment from boilers where Zeolite-softened water was evaporated, a boiler-water sampling line was located lengthwise in the drum of a multidrum boiler wherein the steam generation was at a maximum. The steam sampling nozzle pointing downward was located adjacent to the drum in the horizontal steam line leading to the main steam header. The physical layout of the plant did not permit locating the nozzle in a vertical run of the pipe; nevertheless the indications of entrainment as shown in Figs. 15 and 16 are considered correct, since evidence of entrainment from this station was detected remote from the plant.

Sampling nozzles of new design for both steam and boiler water have been developed by the authors and the engineering personnel, and a schedule of tests will be started in several multidrum boilers. Comparisons will be made for steam quality by the conventional quality method and the conductance method.

The condenser circulating-water supply at Pittsburgh is considered fresh but polluted in character; hence condenser leakage cannot be readily detected by any chemical test. Immediate stoppage is made if leakage is indicated by the conductance method. The make-up is evaporated. Therefore as "pure" water is supplied to the boilers, no undue concentration of boiler-water salts has been found. At Colfax, blowdown of boiler water is not necessary during the operating service run of the boiler; hence the valves are not operated. The loss of concentration because of leaky blow-down valves is a valuable comment. The valves are, however, checked regularly by touch for indication of leakage while the boiler is in service. The concentration was found to be but 703 parts per million of dissolved salts by gravimetric analysis in an adjacent boiler when the boiler was drained. The boiler was operated over the same period as boiler unit 22 and during its 3565 service hours evaporated 716,000,000 lb. of water.

The conductance measurement of the station supplies can be obtained readily by the operating personnel. The chemist's services are required initially, however, for the preparation of the necessary calibration curves. Once such curves are established for each of the *various* water supplies used in the station, they are permanent, unless a major change occurs in the quality of the water supplies. It is obvious that the relation of *dissolved salts* and specific conductance of steam, raw water, and concentrated boiler water in a single station cannot be plotted on a single chart, since the ion distribution of these waters differs. Unfortunately for the users of industrial water, there is no standard procedure available for the chemical analysis of such supplies. The authors have modified the Standard Methods of Water Analysis (see reference item 21) as per the footnotes included in Table 7. When waters of the "pure" quality are subjected to chemical analysis, a minimum of 4 liters is evaporated for the dissolved solids determination; and as is customary in the majority of industrial laboratories, the evaporation is conducted in platinum over a steam bath to avoid evaporation losses; precautions are exercised to maintain the atmosphere in the vicinity of the evaporation in a clean condition; analytical balances are maintained in proper condition by experienced repair men; precision analytical weights are used, and are calibrated monthly against Bureau of Standards certified weights, used for no other purpose. With the weighing precautions as enumerated, it will be conceded that the gravimetric determination represents the most accurate one the chemist has at hand. Because of the doubt of determining accurately the dissolved salts of a water containing from but two to five parts per million and in spite of

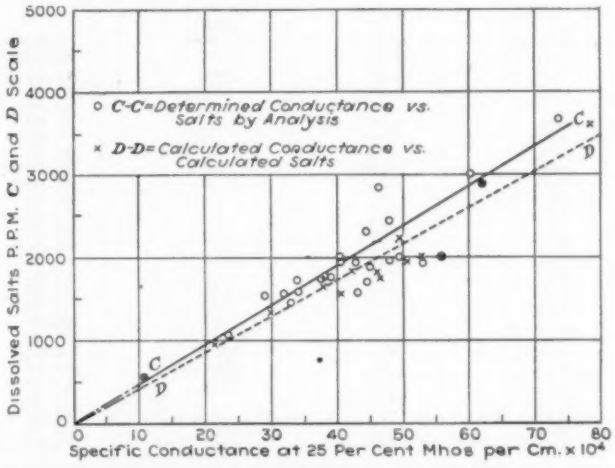


FIG. 17 RELATION OF SPECIFIC CONDUCTANCE TO DISSOLVED SALTS, ZEOLITE-SOFTENED BOILER WATER

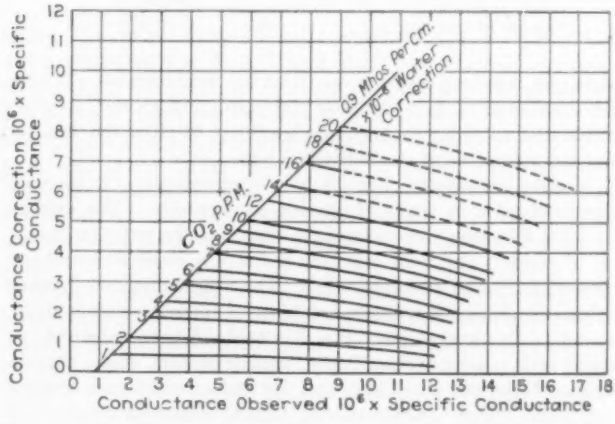


FIG. 18 CONDUCTANCE CORRECTION CURVE, ALLEGHENY RIVER WATER USED FOR STEAM CONDENSATE
(At Colfax power station, Twelfth Street heating plant, and Stanwix heating plant.)

the excellent agreement of the data tabulated in Table 7, column 3, the authors believed the most accurate method of establishing for "pure" water the relation of dissolved salts and specific conductance was by the use of a "synthetic water." The linear relations in the concentrated waters is checked by the plot of the "Calculated Conductance vs. Calculated Salts," Fig. 12, and in the supplemental Fig. 17, in which additional data are supplied. Inspection of the tabulated data and the summary in Table 10 shows that the deviation of the observed and calculated conductance is well within the experimental error of the analysis.

The data in Table 4, boiler fed with "pure" feedwater, and in Table 7, boiler feedwater, have been similarly summarized.

The authors conclude from the foregoing that the submitted comments concerning the non-concordance of the conductance

		Maximum deviation	Average deviation	Deviation of average
	From Table 4:			
Ratio of columns	1 minus 2 divided by column 2	+ 34.4	17.6	+ 17.6
Ratio of columns	3 minus 2 divided by column 2	+ 9.8	5.3	+ 5.3
Ratio of columns	4 minus 5 divided by column 4	+ 7.3	3.6	+ 2.4
Ratio of columns	6 minus 7 divided by column 6	+ 9.7	5.0	+ 3.7
	From Table 7:			
Ratio of columns	3 minus 7 divided by column 7	250.0	127.0	+127.0
Ratio of columns	6 minus 7 divided by column 7	+112.0	18.2	+ 13.9
Ratio of columns	8 minus 7 divided by column 7	+350.0	64.0	+ 22.2

TABLE 10 COMPARISON OF DISSOLVED SOLIDS AND DISSOLVED SALTS BY CONDUCTANCE AND CHEMICAL METHODS

1	2	3	3	4	6	7
Dissolved solids by analysis, p.p.m.	Dissolved salts by analysis, p.p.m.	Dissolved salts calculated, p.p.m.	Specific Conductance at 25 deg. cent. Observed mho per cm.	Calculated, mho per cm.	Dissolved salts interpolated from Observed conductance Plot	Calculated Conductance plot
606	561	563	12.30×10^{-4}	11.00×10^{-4}	500	490
1159	1050	946	23.50	21.35	1100	920
1556	1424	1326	33.00	30.25	1560	1280
1616	1520	..	29.00	..	1380	..
1647	1548	..	31.80	..	1520	..
1737	1553	1532	42.98	40.48	2040	1740
1709	1589	1645	33.80	37.80	1600	1620
1849	1704	1744	44.20	36.40	2100	2000
1866	1714	..	37.50	..	1760	..
1828	1720	..	33.50	..	1600	..
1907	1733	..	39.20	..	1840	..
1935	1776	..	38.20	..	1800	..
1998	1849	1858	44.70	46.10	2120	2000
2062	1914	..	40.50	..	1920	..
2140	1925	1915	53.20	50.41	2520	2400
2092	1933	1850	43.30	42.52	2060	1800
2118	1934	1935	48.20	51.31	2280	2200
2185	1988	..	40.50	..	1920	..
2146	1995	2011	49.60	55.40	2360	2400
2248	2075	2017	56.03	53.14	2520	2280
2378	2243	2238	48.30	49.04	2300	2120
2438	2313	..	44.50	..	2120	..
2887	2745	..	46.50	..	2200	..
3091	2997	2889	60.50	62.13	2880	2640
3882	3675	3605	74.00	78.43	3520	3400

Summary:

Columns	$\frac{(1)-(2)}{(2)}$	$\frac{(3)-(2)}{(2)}$	$\frac{(4)-(5)}{(4)}$	$\frac{(6)-(7)}{(6)}$
Maximum deviation, %	+11.8	−9.9	+11.9	+17.9
Average deviation, %	8.0	2.6	6.3	7.0
Deviation of average, %	+ 8.0	−1.6	+ 1.7	+ 6.6

data and dissolved salts appear to be not well founded. It must be considered that in deriving the calculated conductance the individual value for each sample is subject to all of the errors in the determination of the individual ions.

Another questioned item is the interpretation of the term dissolved salts in terms of specific conductance. The chemist uses the following nomenclature in reporting the results of analysis of water. The total dissolved solids of a water included suspended solids and dissolved solids. The weighed residue obtained after evaporating a measured volume of filtered water is designated as dissolved solids. The dissolved solids includes the colloidal and non-conducting items, organic matter, silica, and the oxides of iron and aluminum, and the ionized substances possessing conductance characteristics, termed dissolved salts. The correlation of conductance may only be made with reference to the dissolved salts. It is probable that with a clearer understanding of these terms a user of the conductance method who has implied that the results are erroneous may be able to readjust the voluminous data compiled by him and make them usable. The authors are not aware of any method other than the gravimetric method for determining rapidly the total dissolved solids in water.

It will be observed from the supplemental data herewith, Table 11 and Figs. 14 and 15, that the carbon-dioxide compensation of supplies containing measurable amounts of carbon dioxide must be made if accuracy in the entrainment value is desired. The frequency of making the CO_2 test is based upon the local plant conditions and the rate of change of CO_2 concentration in the steam discharged from the boiler. The steam condensate returns referred to in the paper possessed a constant CO_2 content; hence a daily test suffices. In the tests on entrainment of steam generated in boilers using Zeolite-softened water, the CO_2 test was made at 10-minute intervals throughout each test run. In these runs the conductance of the steam was measured each minute, and the readings were used to compare the method of test for moisture in steam as published by Fitze (see reference item 16) with the method presented by the authors. The authors believe the following formula correctly expresses Fitze's method:

$$\frac{L_S - L_W}{\frac{\Delta L_S}{\Delta \%BW}} = \text{per cent moisture}$$

Where

L_S = measured conductance of steam

L_W = water correction derived by transposition of curve

$\frac{L_S}{\%BW}$ = the change in the conductance of steam caused by the addition of 1 per cent boiler water

The formula for the authors' method included the constants for water correction and the derived value for the CO_2 correction.

$\frac{L_S - L_{CO_2} - L_W}{L_{BW} - L_W} \times 100$ equals per cent by weight of entrainment of boiler water in steam

Where

L_S = measured conductance of steam

L_W = water correction 0.9×10^{-6} mhos per cm.

L_{CO_2} = carbon dioxide correction

L_{BW} = conductance of boiler water

The curves for obtaining the CO_2 correction are shown in Fig. 17, calculated from the data given in Table 6.

The tabulated data shown in Table 11 also include the entrain-

TABLE 11 COMPARISON OF CONDUCTANCE METHODS FOR DETERMINING QUALITY STEAM

		1		2	
Boiler Number		1	2	1	2
Run Number		1-17-30	1-17-30	12-27-29	12-27-29
Date of Run		45	45	45	45
Duration of run, min		115,000	182,000	101,400	185,200
Steam flow, lb. per hour		245.8	249.4	226.8	235.9
Steam pressure, lb. per sq. in. gage					

Results on Composite Samples Collected During Runs

	Supply sampled	Steam	Boiler water	Steam	Boiler water	Steam	Boiler water	Steam	Boiler water
	Analyses, values in p.p.m.:								
1	Dissolved solids, by evaporation	3.30	2118	3.20	2092	1.93	2140	4.25	1737
2	Organic matter	0.60	107	1.60	97	0.85	104	1.10	109
3	Silica	0.30	73	0.20	58	0.33	106	1.68	71
4	Oxides of iron and aluminum	1.20	4	0.80	4	0.15	5	0.13	4
5	Dissolved salts, by analysis, item 1—(2+3+4)	1.20	1934	0.60	1933	0.60	1929	1.34	1553
6	Calcium (Ca)	0.05	2	0.05	2	0.05	2	0.10	2
7	Magnesium (Mg)	0.04	2	0.04	2	0.04	2	0.04	2
8	Sodium (Na)	0.28	728	0.10	661	0.12	722	0.27	577
9	Bicarbonate (HCO_3)	2.57	0	0.00	0	0.05	0	0.50	0
10	Carbonate (CO_3)	0.00	10	0.00	38	0.00	44	0.00	19
11	Hydroxide (OH)	0.00	151	0.00	77	0.00	144	0.00	118
12	Sulphate (SO_4)	0.58	912	0.49	934	0.49	875	0.58	717
13	Chloride (Cl)	0.21	130	0.00	136	0.00	126	0.00	97
14	Dissolved salts calculated from analysis, sum (6 to 13 incl.)	3.73	1035	0.68	1850	0.75	1915	1.49	1532
15	Carbon dioxide (CO_2)	7.1	0	6.0	0	15.8	0	13.0	0
16	Specific conductance, mhos per cm. 10^6	10.0	4820	8.9	4330	7.0	5320	5.3	4298
17	Specific conductance corrected, mhos per cm. 10^6	6.9	4819.1	5.1	4329.1	0.0	5319.1	0.0	4297.1
18	Entrainment, ratio of dissolved salts, item 5, per cent	0.062[a]		0.031[a]		0.031[a]		0.086[a]	
19	Entrainment, cond. of composite samples, item 17, per cent	0.144[b]		0.118[b]		0.000[b]		0.000[b]	
20	Entrainment, observed readings, average	0.022[b]		00.33[b]		0.000[b]		0.000[b]	
21	Moisture, observed readings, average	0.248[c]		0.282[c]		0.14[c]		0.13[c]	

[a] $\frac{\text{Dissolved salts steam}}{\text{Dissolved salts boiler water}}$

[b] $\frac{L_S - L_{CO_3} - L_W}{L_{BW} - L_W} \times 100$

[c] $\frac{L_S - L_W}{\Delta L_S \text{ per } \Delta\% BW}$

ment in per cent calculated using the dissolved salts found (by chemical analysis) in the steam and boiler water. Considering the value so obtained as a standard, it will be observed that the data secured by the authors' procedure approximate the standard considerably closer than do the data supplied by Fitze, wherein no CO_2 compensation is made.

The authors have not arrived at any additional explanation of the observation, "it appears that the entrainment in steam is greater in magnitude for lower steam-flow conditions than is found under the normal steam output of the boiler," and, "the steam quality apparently improved with the increase in concentration of boiler water and total evaporation." They wish to point out that sufficient evidence by other than conductance measurement is included in the paper to support Fig. 11 for the observed conditions.

The final general factor which is mentioned is the change of conductance because of change in concentration of the individual ions. Considering the experimental data reported in this paper, the authors believe that boiler concentrations up to 5000 parts per million of dissolved salts may be considered as dilute solutions. The graphs shown in the Figs. 12 and 16 are considered linear. It has been noted that the equivalent conductance of salts, such as NaCl, $\frac{1}{2}$ $CaCl_2$, $NaNO_3$, $\frac{1}{2}$ Na_2SO_4, and $\frac{1}{2}$ Na_2CO_3, show a more rapid decrease in equivalent conductance from zero concentration up to approximately 300 parts per million of the salt, and beyond this concentration the slopes of the graphs are linear. The slope for NaOH is linear throughout its range. The authors confess that they do not know how the ions are combined within the boiler at operating temperatures and pressures, and hesitate to correct the data they have accumulated for the small change in the conductance. The substantial agreement that they have found between the observed and calculated relations as shown in Figs. 12 and 16 permits them to conclude that the error they may have made in their calibration of the meter units in terms of dissolved salts may be considered of minor value.

The authors are grateful to the discussors for their presentation of the various aspects of the paper and believe that a large field is open for investigation both in the industry and in the engineering schools

FSP-53-12

Design of Thick-Walled Tubes Subjected to Pressure and Heat Input

By E. W. LUSTER,[1] ELIZABETH, N. J.

With the increases in pressure and oil temperature, which have accompanied the development of distillation and cracking equipment in recent years, has arisen the need for a rational method for the design of thick tubes subjected to internal pressure and heat input. The author has accordingly developed in the present paper formulas for temperature and pressure stresses in such tubes, and for calculating the optimum tube for given conditions. Examples of calculations are given, as well as particulars of experimental investigations, both completed and contemplated, for the determination of the values of the constants employed in the formulas.



WITH the increases in pressure and oil temperature which have taken place in oil-refinery distillation and cracking equipment within recent years, the need of a rational method for the design of thick tubes subjected to internal pressure and heat input has become increasingly urgent.

In presenting this paper no claim is made that a tested and final solution has been found. The question of combined temperature and pressure stresses is a controversial one, a large group of engineers of high standing denying the existence of the temperature stresses. From the mathematical work which accompanies this report, the development of the theory to which the author adheres may be followed. In the laboratories of the Standard Oil Development Company the determination of constants for use in the equations developed is being given intensive study, and, as the ultimate step in rounding out the work, it is hoped by testing tubes under pressure and heat input, to prove or disprove the theory. The ultimate solution of this problem will not be due entirely to the efforts of mechanical engineers as the difficulties facing the designer have a metallurgical as well as a mechanical aspect.

While the present paper was prepared primarily for use in the design of oil-refinery pipe stills and cracking furnaces, it has a direct application in the design of any equipment employing tubes subjected to internal pressure and heat input from the outside, and with a change in mathematical signs will cover the design of tubes in which heat flows outwardly.

[1] Assistant Manager of General Engineering Department, Standard Oil Development Company. Mem. A.S.M.E. The author was graduated from Rutgers University, 1914, with a B.S. degree in electrical and mechanical engineering. Spent one and one-half years in power-plant test work. Joined the Standard Oil Company in June, 1916. Attended first Military Training Camp and served as lieutenant and captain in the Second Regiment of the U. S. Engineers, Second Division of the A.E.F. in France, and later in the Army of Occupation in Germany. Following discharge from the army was resident engineer for the Societa Italo-Americana Pel Petrolio in Genoa, Italy, until the end of 1923, erecting bulk storage and bunkering stations in Italy and North Africa. Returned to the United States the first of 1924 and has since been in charge of the design of process equipment for the Standard Oil Development Company.

Presented at a meeting of the Plainfield Section of the A.S.M.E., Plainfield, N. J., April 3, 1930.

In the design of our earlier cracking units, that is, those built up to 1926, no account was taken of temperature stresses. The allowable stress in the tubes was taken as that given by the A.S.M.E. Boiler Code, and the stress in the tubes was calculated by the thick-wall formulas. As oil temperatures were relatively low, from 850 to 860 deg., and the pressure moderate, being 350 lb. at the outlet tube of the furnace, and as very liberal allowance was made to care for the loss of wall thickness due to corrosion, tube failures were rare, loss of property negligible, and injury to attendants non-existent.

In 1926 cracking units for 750 lb. oil discharge pressures were built which gave an oil pressure at the inlet tube of the furnaces of around 1000 lb. In the design of these units also no temperature-stress allowance was made, though the tubes were designed with a liberal corrosion allowance and of sufficient thickness to withstand at the hottest tube the full pressure which could be developed by the feed pumps, namely, some 1200 lb. By suitable oil flow the tube carrying the hottest oil was not in a section of the furnace where it would be subjected to high heat inputs. For these reasons tube failures in these settings have also been extremely rare.

The next step in the development of cracking furnaces called for a setting which had no zones of low heat input. A furnace was required in which the oil and flue gas traveled in a countercurrent manner, the heat input being as nearly uniform throughout the setting as possible. Such a furnace has the tube carrying the hottest oil in the so-called "radiant" section of the furnace where the heat inputs average 10,000 B.t.u. per sq. ft. of exposed tube surface.

(Note that in this paper heat inputs will be spoken of in this manner instead of the heat input per square foot of projected areas, which is the practice of some designers.)

It was realized that the securing of a fair heat input through a relatively thick tube, through the layer of hard coke which builds up on the inner wall of the tube due to the conversion of the cracking stock to gasoline, and finally through the oil film which exists between the coke layer and the oil, would give rise to metal temperatures greatly in excess of those encountered in boilers. It was also realized that the condition would be aggravated by the fact that the oil receiving the heat would be at an elevated temperature, have a low thermal conductivity, and would not be boiling at least to the extent that water does in a steam boiler. It was estimated that under these conditions the tubes would have an outside wall temperature of from 950 to 1100 deg. or higher, depending upon the various factors mentioned.

The allowable stresses in metals and alloys at such elevated temperatures were very poorly defined, and it was obvious that the initial steps toward a rational design for such tubes had to be, first, the selection of a proper metal, and, second, the determination of safe stresses for this metal.

The temperatures mentioned above are within the range over which a metal is considered to elongate at a uniform rate when subjected to a sufficiently high stress. Although this phenomenon has been the subject of numerous publications, it may not be amiss to mention that this uniform elongation is known as

"creep." To put it a little differently, it may be said that at a given temperature there is some stress which will, in a given period of time, cause the metal to elongate or creep a definite percentage of its original length. The 100,000-hr. creep is the one in most general usage, and is the stress which, at a given

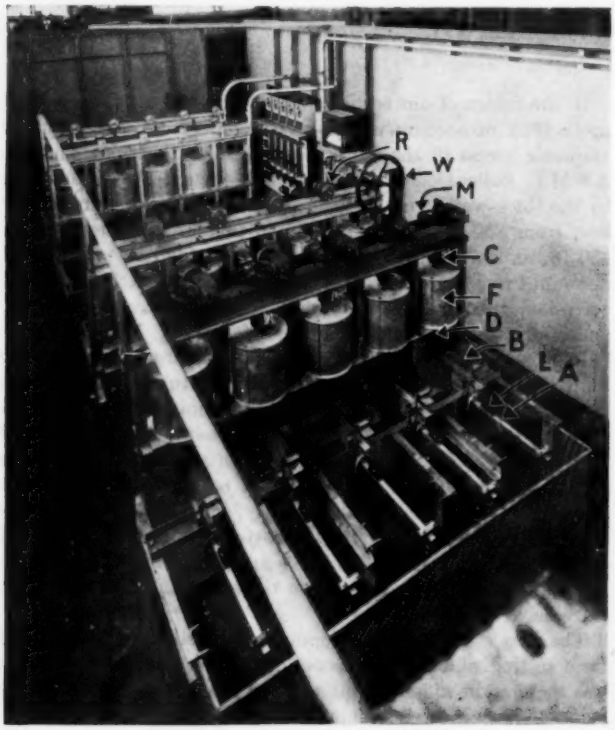


FIG. 1 ELECTRIC FURNACE USED IN CREEP DETERMINATIONS

temperature, causes the metal to elongate 1 per cent of its original length in 100,000 hr. (11.4 years).

Some of the tube manufacturers had already made creep determinations on a few alloys, but because of the many conditions to be met in oil-refinery work it was felt advisable to install equipment and make determinations for those materials for which a possible use could be foreseen. Creep-determination equipment consists essentially of an electric heating furnace, a system for loading the test specimen, a micrometer for reading the elongations, and thermal control equipment for maintaining the furnace temperature at any desired point. Our metallurgical laboratory has available at present twenty-five such furnaces arranged in batteries of five.

The furnace is shown at *F* in Fig. 1, and was constructed by the metallurgical division of the company from information supplied by the Union Carbide and Carbon Corporation. It consists of an alundum tube $1^1/_8$ in. in inside diameter and is space-wound with Nichrome IV wire. Alundum cement is applied over the spaced wire and the tube and wire covered with 4 in. of insulation. The furnaces are $12^1/_2$ in. long and give a very uniform temperature distribution over the middle 6 in. of the specimen. The sample to be tested is screwed into blocks *C* and *D*, Fig. 1, and the load applied by the double lever *A* and *B* which has a multiplying ratio of 100 to 1. By this system stresses up to 75,000 lb. per sq. in. may be applied to the standard 0.505-in.-diameter test specimen. Elongation of the test specimen is measured by means of a large-size micrometer, the gear and worm of which have been calibrated. The micrometer device *M* is turned by wheel *W* when necessary to keep the bubble in lever *L* in the center of the glass. The dial *R* is calibrated to twenty-five millionths of an inch, and readings estimated to the nearest five millionths.

The temperature is controlled within five degrees plus or minus by means of potentiometer controls, relays, and resistors. The control *C* of Fig. 2 operates a relay *X* which in turn cuts in or out a part of resistor *R*-1. Resistor *R*-2 is in series with the furnace, which, after once being set for a given temperature, need not be changed. Fig. 2 shows a control system for five furnaces, each equipped with a controller. The temperature of each specimen is recorded on the potentiometer recorder shown at *T* in Fig. 3. Fig. 3 also shows the control system for a bank of five furnaces operated simultaneously at the same

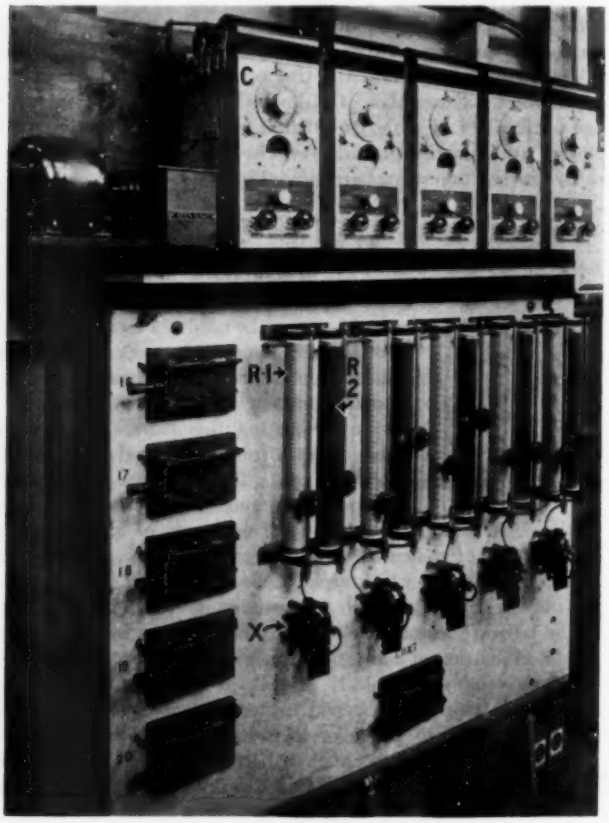


FIG. 2 FURNACE CONTROLS

temperature by a single controller. Rheostats *R*, Fig. 3, are in series with each furnace and compensate for the slight differences in furnace characteristics.

Fig. 4 shows a specimen with the thermocouples attached, the specimen being ready for insertion in the furnace. One thermocouple is connected to the controller and the other to the recorder.

In operation the specimen is given an initial load which is based on previous experience or represents the best guess possible. The application of the load elongates the specimen immediately, after which the rate of elongation decreases, and on a chart in which the elongation is the ordinate and time the abscissa, the curve becomes horizontal, that is, the specimen has ceased to elongate. The load is then increased, and this time a greater period is required before the line becomes horizontal. Increases in loading are continued until the line no longer becomes horizontal but continues to slope upwardly to the right, indicating

a continuous uniform elongation. By properly adjusting the load the rate of elongation can be brought to approximately 1 per cent in 100,000 hr., and after this condition has been established the apparatus is allowed to run from 2000 to 3000 hr. From the elongation which takes place in this 2000- to 3000-hr. period, the actual rate of creep for 100,000 hr. is calculated. In practice two furnaces are used per test, one being set to give a creep of a little over 1 per cent and the other a little less than 1 per cent, so that at the conclusion of the test the 1 per cent point is calculated by interpolation. As a final test a specimen is stressed with the load given by the interpolation, and a check test made for 3000 hr. It can readily be understood that the determination of a complete creep-point curve covering a range of from 900 to 1500 deg. and requiring the determination of at least four points to insure an accurate and smooth curve, occupies a long period of time. For example, assuming 3000 hr. per test, three tests for the determination of each point on the curve, and four points required for a complete curve, a total of 36,000 hr. or 4.1 years would be required to produce this curve if only one machine were available. With twenty-five machines

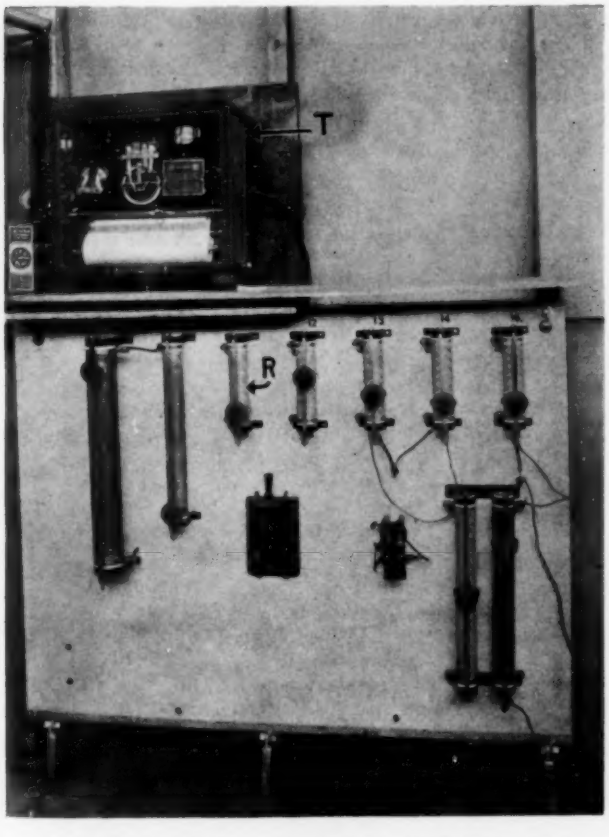


FIG. 3 CONTROL SYSTEM FOR BANK OF FIVE FURNACES

available working in the method outlined above it should be possible to make about six curves per year.

We now have eleven of the furnaces on high chrome-nickel covering the temperature range of from 800 to 1300 deg. in an attempt to correlate the creep limits at 25,000, 50,000, and 100,000 hr. Three machines are on low-carbon steel, six on various chrome steels, and five on special alloys for very high temperatures of from 1800 to 1900 deg. It was mentioned above that an attempt was being made to correlate the creep rates at three different periods; this is being done as the 100,000 hr. is a long time in the oil industry and few indeed are the pieces of equipment which are not scrapped before the end of such a period. One per cent in from five to six years, or, say, 50,000 hr., would appear to be a logical creep limit, as it more nearly represents the life of our equipment.

The full-line curve on Fig. 5 gives the 100,000-hr. creep on KA2S. It extends from 900 to 1300 deg. fahr. The broken-line curve represents one originally supplied by the Babcock & Wilcox Company for Enduro which covers the range from 1000 to 1500 deg. fahr. As these curves are the work of two independent investigators, it is fairly safe to assume that the overlapping portions, at least, may be used for design work. Their close agreement can be noted, in spite of the considerable difference in carbon content: 0.07 per cent for KA2S and 0.43 per cent for Enduro.

Table 1 gives some scattered values for four steels. To date, sufficient points have not been determined to make it possible to draw complete curves.

Following the determination of safe stresses, the next logical step in the solution of the problem was the derivation of a suitable formula with which the actual stresses set up in the tubes could be calculated

The most general case is one in which relatively hot oil is forced through a tube under pump pressure, the tube being located in a suitable furnace whereby heat is imparted through the tube to the oil by radiant brickwork and hot flue gas.

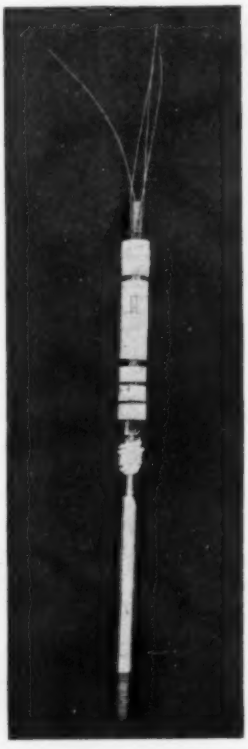


FIG. 4 SPECIMEN WITH THERMOCOUPLES ATTACHED, READY FOR INSERTION IN FURNACE

Under such conditions there should be, according to previous investigators:

1 Stresses accompanying the temperature drop through the tube wall
2 Stresses accompanying the pressure difference between the inside and outside of the tube.

Temperature stresses have been discussed by Lorenz,[2] Dahl,[3] and Orrok.[4] The principal theories of pressure stress are those of Lamé, Barlow, and Clavarino, and are to be found in standard textbooks dealing with this subject.

All the above writers make various assumptions and have worked out many different cases of pressure and temperature stresses. Using their work as a general guide, and starting with the fundamental stress-strain equations, formulas have been

TABLE 1 CREEP VALUES FOR FOUR STEELS

	Lb. per sq. in.	
	800 deg. fahr.	900 deg. fahr.
0.12 to 0.18 per cent carbon steel	15,000	9,500
Low Cr-Ni-Mn	30,000	21,500
KA2S	25,000	20,000

[2] Dr. Ing. Rudolph Lorenz, "Temperaturspannungen in Hohzylindern," *Zeitschrift des Vereines deutscher Ingenieure*, 1907, p. 743.

[3] O. G. C. Dahl, "Temperature and Stress Distribution in Hollow Cylinders," Trans. A.S.M.E., vol. 46 (1924).

[4] Geo. A. Orrok, "High-Pressure Steam Boilers," Trans. A.S.M.E., vol. 49–50 (1928), paper FSP-50-32.

obtained which apply to the conditions in oil-refinery pipe-still and cracking furnaces. The derivation of these is given in Appendix No. 1 of this paper.

When heat flows through the walls of a tube (from the outside to the inside in cracking furnaces), a temperature drop is established. That is to say, any fiber of the material is hotter than the one just inside it. Consequently the thermal expansion of the outer fibers is greater than that of the inner fibers. From this a little consideration will show that the outer fibers are in compression and the inner ones are in tension.

An example may make this clearer. Consider two precisely similar strips of metal, laid one upon the other, with a strip of

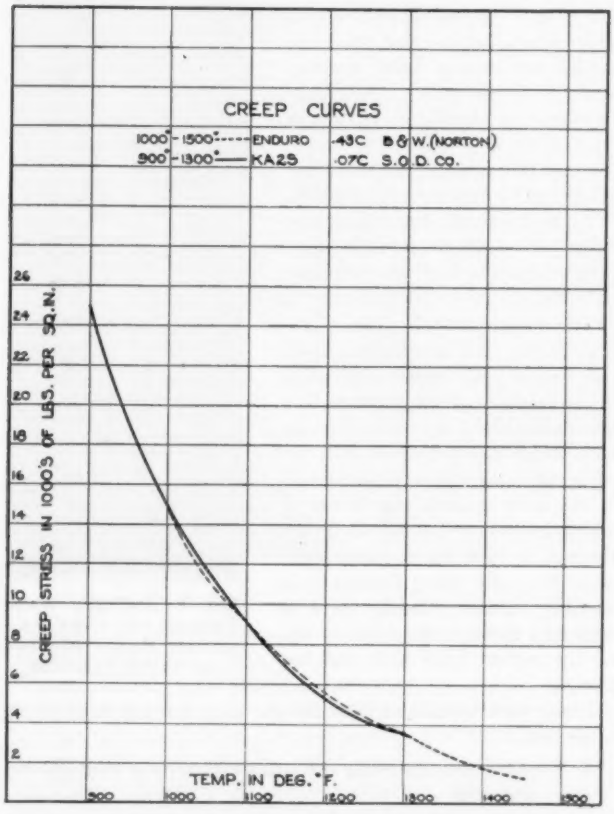


FIG. 5 CREEP CURVES OF TWO ALLOYS

insulating material between them and clamped together. If heat be applied from one side, the exposed strip will tend to become longer. The hotter strip can only expand by stretching the cooler one. The cooler strip is obviously in tension, and its resistance to being stretched produces compression in the hotter strip. If we had a pile of many such insulated strips instead of only two, with a rise in temperature upward from strip to strip, i.e., heat applied from the top, the action described above would take place between adjacent strips. Further than this, any strip in the pile would have all the strips below shortening it and all the strips above stretching it. Strips near the top would be in compression, and those near the bottom would be in tension. This is because a strip close to the top would be acted upon by more strips which tend to shorten it than by strips which tend to elongate it (above it), and vice versa. Finally, there would be a strip somewhere in the pile which would be neither shortened nor elongated, but would remain its original length.

The same remarks apply to the strips considered breadthwise instead of lengthwise. The same stresses are set up, as the strains and temperature changes are identical in the two directions.

Now apply all the above to a set of insulated concentric hoops, forming a laminated ring. We then have a clear conception of what goes on in a metal tube or cylinder with a temperature drop from the outside to the inside. The outside will be in compression, the inside in tension, and one cylindrical surface will remain at its original circumference. This of course applies to the hoop stresses. From the breadth considerations we can see that lengthwise or axial stresses exist in the cylinder.

In the pile of strips, there are no stresses vertically, since in this direction expansion is unrestrained. In the laminated cylinder, however, this is not so. The stretching of the inner hoops and contracting of the outer ones accompany a pressure between them, in a radial direction. There are corresponding distortions affecting the hoop and axial strains. These strains are not now identical, and the hoop and axial stresses are not, in general, equal at any point in the laminated wall. It is, however, obvious that, at the inside and outside surfaces, the radial stress is zero. Here, due to expansion, we should expect the hoop and axial stresses to be equal, and the mathematical analysis given in Appendix No. 1 shows this to be the case.

The principal stresses to be considered are those in the axial and circumferential directions at the inside and outside tube surfaces. They are expressed by the equations (see Appendix No. 1):

Temperature Stresses:

$$[1]\quad f_0 = \frac{m}{2(m-1)}\, aET_0\left(\frac{2}{K^2-1} - \frac{1}{\log_e K}\right)$$ Axial or hoop stress at outer tube surface

$$[2]\quad f_1 = \frac{m}{2(m-1)}\, aET_0\left(\frac{2K^2}{K^2-1} - \frac{1}{\log_e K}\right)$$ Axial or hoop stress at inner tube surface

Pressure Stresses:

$$[3]\quad F_0 = \frac{2P}{K^2-1}$$ Hoop stress at outer tube surface

$$[4]\quad F_1 = P\,\frac{K^2+1}{K^2-1}$$ Hoop stress at inner tube surface

$$[5]\quad F = \frac{P}{K^2-1} = \frac{F_0}{2}$$ Axial stress anywhere

$$[6]\quad T_0 = \frac{\log_e K \times Qr}{k}$$ Temperature drop through tube wall

where a = coefficient of thermal expansion
E = Young's modulus for the material
f_0 = temperature hoop stress at outer surface
f_1 = temperature hoop stress at inner surface
F_0 = pressure hoop stress at outer surface
F_1 = pressure hoop stress at inner surface
K = ratio, outside diameter ÷ inside diameter
k = conductivity
$1/m$ = Poisson's ratio
P = internal pressure
Q_0 = rate of heat input at outside surface
r_0, r_1, r = tube radius, outside, inside, anywhere
T_0 = outside metal temperature — inside metal temperature.

On considering the formulas for temperature stress, we see that they contain the physical quantities a, E, and T_0 in the form of a product. An increase in any or all of them increases the stress.

Taking them in order, an increase in a, the coefficient of thermal expansion, means that the same temperature rise will be accompanied by a greater expansion of the material. The stretching of the cooler portion will then be more, and greater stress will result. A metal of zero thermal expansion would give a zero temperature stress. Such metals are available, but those known lack other essential characteristics.

E, Young's modulus, is defined by the quotient, stress over strain. It is obvious that, for the same strain, an increase in E means an increase in stress.

An increase in T_0, the temperature difference between two points, affects the expansion of the material in the same way as does an increase in a, stress increases for the same reason. T_0 also depends on the conductivity k. For a low k, or small conductivity, a greater temperature difference T is required for the same flow of heat.

From the above considerations, material for tubes should have the following characteristics:

Small expansion due to temperature
Small modulus of elasticity
High conductivity
Great tensile strength at high temperatures.

To facilitate the use of the formulas, Table 2 has been prepared.

Column K gives values of outside diameter ÷ inside diameter

Column A gives values of $\dfrac{2}{K^2-1}$

Column B gives values of $\dfrac{K^2+1}{K^2-1}$

Column C gives values of $\dfrac{2}{K^2-1}-\dfrac{1}{\log_e K}$ (always negative)

Column D gives values of $\dfrac{2K^2}{K^2-1}-\dfrac{1}{\log_e K}$

We then have:

$$\text{Total outside hoop stress} = PA - \frac{maET_0C}{2(m-1)} \quad \ldots\ldots [7]$$

$$\text{Total outside axial stress} = \frac{1}{2}PA - \frac{maET_0C}{2(m-1)} + \text{bending stress} \quad \ldots\ldots [8]$$

TABLE 2 VALUES FOR USE IN FORMULAS

$\left(K = \dfrac{\text{Outside rad.}}{\text{Inside rad.}},\ A = \dfrac{2}{K^2-1},\ B = A+1,\ C = \dfrac{1}{\log_e K} - A,\ C + D = 2\right)$

K	A	B	C	D	K	A	B	C	D
1.01	99.50	100.50	1.000	1.000	1.26	3.40	4.40	0.925	1.075
1.02	49.50	50.50	0.995	1.005	1.27	3.26	4.26	0.922	1.078
1.03	32.84	33.84	0.991	1.009	1.28	3.13	4.13	0.919	1.081
1.04	24.51	25.51	0.987	1.013	1.29	3.01	4.01	0.916	1.084
1.05	19.51	20.51	0.984	1.016	1.30	2.90	3.90	0.913	1.087
1.06	16.18	17.18	0.981	1.019	1.31	2.80	3.80	0.911	1.089
1.07	13.80	14.80	0.978	1.022	1.32	2.70	3.70	0.909	1.091
1.08	12.02	13.02	0.975	1.025	1.33	2.60	3.60	0.907	1.093
1.09	10.63	11.63	0.972	1.028	1.34	2.51	3.51	0.905	1.095
1.10	9.52	10.52	0.969	1.031	1.35	2.43	3.43	0.902	1.098
1.11	8.62	9.62	0.966	1.034	1.36	2.35	3.35	0.900	1.100
1.12	7.86	8.86	0.963	1.037	1.37	2.28	3.28	0.897	1.103
1.13	7.22	8.22	0.960	1.040	1.38	2.21	3.21	0.895	1.105
1.14	6.68	7.68	0.957	1.043	1.39	2.15	3.15	0.892	1.108
1.15	6.200	7.200	0.954	1.046	1.40	2.08	3.08	0.889	1.111
1.16	5.79	6.79	0.951	1.049	1.41	2.02	3.02	0.886	1.114
1.17	5.42	6.42	0.948	1.052	1.42	1.97	2.97	0.884	1.116
1.18	5.10	6.10	0.945	1.055	1.43	1.91	2.91	0.882	1.118
1.19	4.81	5.81	0.942	1.058	1.44	1.88	2.88	0.880	1.120
1.20	4.55	5.55	0.939	1.061	1.45	1.82	2.82	0.878	1.122
1.21	4.31	5.31	0.937	1.063	1.46	1.77	2.77	0.876	1.124
1.22	4.10	5.10	0.935	1.065	1.47	1.72	2.72	0.873	1.127
1.23	3.90	4.90	0.933	1.067	1.48	1.68	2.68	0.871	1.129
1.24	3.72	4.72	0.930	1.070	1.49	1.64	2.64	0.869	1.131
1.25	3.56	4.56	0.927	1.073	1.50	1.60	2.60	0.866	1.134

$$\text{Total inside hoop stress} = PB + \frac{maET_0D}{2(m-1)} \quad \ldots\ldots [9]$$

$$\text{Total inside axial stress} = \frac{1}{2}PA + \frac{maET_0D}{2(m-1)} + \text{bending stress} \quad \ldots\ldots [10]$$

The bending stress depends on the manner in which the tube is supported, and is calculated by the usual formulas. The total axial stress is sometimes important, as cases have occurred in which tubes sagged down under the action of the combined axial stresses.

As an example of the use of Table 2, consider a tube 5 in. O.D., 4.2 in. I.D., with a heat input at the outside of 10,000 B.t.u. per sq. ft. per hr. Take P as 1000 lb. per sq. in. Let the material be KA2S, in which, assume

$a = 11.5 \times 10^{-6}$ inches per inch
$E = 24.5 \times 10^6$ lb. per sq. in.
$K = 5 \div 4.2 = 1.19$
$k = 144$ (B.t.u. per sq. ft. per in. of thickness per deg. fahr.)
$m = 3^1/_2$ or $1/m = 0.3$

$$\text{Then } T_0 = \frac{\log_e(1.19) \times 10{,}000 \times 2.5}{144} = 30 \text{ deg. from [6]}$$

$$\frac{maE}{2(m-1)} = \frac{5 \times 11.5 \times 24.5}{7} = 201$$

From Table 2, for $K = 1.19$, $A = 4.81$, $B = 5.81$, $C = 0.942$, $D = 1.058$.

Outside Hoop Stress:

$1000 \times 4.81 = 4810$ from [7]
$-(201 \times 30 \times 0.942) = -5700$
-890

or 890 lb. per sq. in. compression.

Outside Axial Stress:

-5700 from [8]
$+2405$
-3295

or 3295 lb. per sq. in. compression + the bending stress.

Inside Hoop Stress:

$1000 \times 5.81 = 5810$ from [9]
$201 \times 30 \times 1.058 = 6400$
12,210 lb. per sq. in. tension

Inside Axial Stress:

2405 from [10]
6400
8805 lb. per sq. in. tension + bending stress.

It will be seen from the above that the inside hoop stress is the one which governs design, since the bending stresses can be made very small by proper supports. If we take the 5-in. O.D. tube and vary the inside diameter, we can compute Table 3.

From this table, the curves shown in Fig. 6 are plotted. As indicated on the plot, the inside

TABLE 3 HOOP STRESSES, LB. PER SQ. IN.

K	T_0	Inside hoop stress	Outside hoop stress
1.04	6.8	26,900	23,160
1.06	10.1	19,250	13,990
1.08	13.4	15,780	9,400
1.10	16.5	13,950	6,310
1.12	19.7	12,960	4,050
1.14	23.8	12,440	2,100
1.16	25.8	12,240	860
1.18	28.7	12,200	— 350
1.20	31.6	12,300	—1,410
1.22	34.5	12,500	—2,360
1.24	37.4	12,770	—3,280
1.26	40.0	13,050	—4,020
1.28	43.0	13,480	—4,780
1.30	45.5	13,850	—5,450

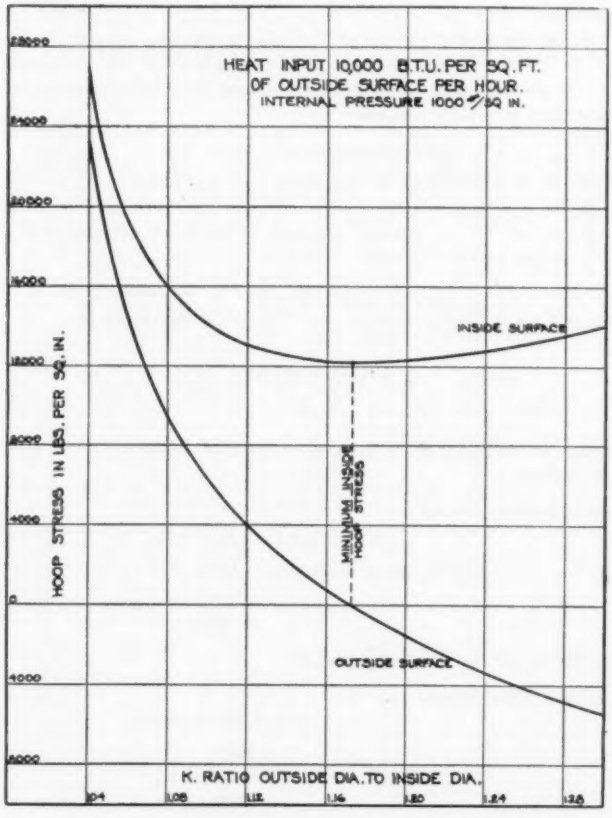


FIG. 6 HOOP STRESSES IN HEATED TUBES UNDER PRESSURE

hoop stress has a minimum value, and for the corresponding value of K the outside hoop stress is zero. As this value occurs at $K = 1.173$, the 5-in. O.D. tube having the lowest inside hoop stress would have an inside diameter of

$$5 \div 1.173 = 4.25 \text{ in.}$$

As pointed out above, the inside hoop stress is the one which governs design. Then by the selection of the proper thickness for the conditions imposed we can secure the minimum hoop stress. This is the optimum tube for the conditions—a tube either thicker or thinner would be weaker. If the inside hoop stress exceeds the safe working stress of the metal, which is usually taken at from 60 to 80 per cent of the 100,000-hr. creep, then the designer must either reduce the metal temperature, the heat input, or the pressure, or else find a metal having a higher creep limit.

For practical calculations, plotting the curves to find the optimum tube involves considerable labor. This can be avoided by the use of Table 4, the mathematical derivation of which is given in Appendix No. 2.

The following example shows its use in the direct calculation of the above-mentioned optimum tube:

$$b = 1.4\,Qr = 1.4 \times 10{,}000 \times 2.5 = 35{,}000 \text{ from [11]}$$

$$2P/b = 2000/35{,}000 = 0.057$$

This lies between $K = 1.16$ and 1.18 in the table. By interpolation,

1.18	0.061	0.057
1.16	0.049	0.049
0.02	0.012	0.008

$$K = 1.16 + 0.02 \times 0.008/0.012$$

$$0.013$$

$$K = 1.173$$

Inside temp. hoop stress =
$35{,}000 \times 1.053 \times \log_e 1.173 = 5900$ from [14]
Inside pressure hoop stress $= 6.30 \times 1000 = 6300$ from [9]

Total inside hoop stress 12,200

Outside temp. hoop stress =
$-35{,}000 \times 0.947 \times \log_e 1.173 = -5300$ from [13]
$5.30 \times 1000 = 5300$ from [7]

Total outside hoop stress 0

These results check with Table 3. It will be noticed that the inside-hoop-stress curve is very flat in the neighborhood of the minimum value. It is therefore quite accurate enough to take K as the nearest value given in Table 4 without resorting to interpolation. This has only been done in the above instance for the sake of completeness.

We might have started with the inside diameter of 4.25 in., to agree with some particular oil velocity. Then for the same total heat input to the tube, we should have had $Q = 10{,}000 \times (5/4.25)$ and b would have been

$$1.4 \times [10{,}000 \times (5/4.25)] \times 2.125 = 35{,}000$$

the same value as before. Then by the use of Table 4, K would have been found to be 1.173, giving an outside diameter of $4.25 \times 1.173 = 5$ in. The table can thus be used to get the optimum tube, if the total heat input, the internal pressure, and either the inside or outside diameter be known. In fact, if any three of these quantities be known, the other can be found for optimum conditions. If we had a tube 5 in. O.D.,

TABLE 4 FOR USE IN SELECTION OF OPTIMUM TUBE

K	$\frac{2P}{b}$	K	$\frac{2P}{b}$
1.02	0.0018	1.52	0.470
1.04	0.0031	1.54	0.505
1.06	0.0071	1.56	0.540
1.08	0.012	1.58	0.585
1.10	0.019	1.60	0.620
1.12	0.027	1.62	0.660
1.14	0.038	1.64	0.698
1.16	0.049	1.66	0.740
1.18	0.061	1.68	0.785
1.20	0.075	1.70	0.830
1.22	0.091	1.72	0.875
1.24	0.105	1.74	0.920
1.26	0.126	1.76	0.970
1.28	0.145	1.78	1.018
1.30	0.165	1.80	1.065
1.32	0.189	1.82	1.110
1.34	0.209	1.84	1.160
1.36	0.235	1.86	1.220
1.38	0.260	1.88	1.265
1.40	0.286	1.90	1.325
1.42	0.317	1.92	1.385
1.44	0.342	1.94	1.430
1.46	0.375	1.96	1.495
1.48	0.405	1.98	1.550
1.50	0.438	2.00	1.614

$b = 1.4\,Qr$ for KA2 Eq. [11]
$= 0.47\,Qr$ for steel Eq. [12]

Temperature hoop stresses *for Optimum Conditions:*
Outside: $bC \log_e K$ Eq. [13]
Inside: $bD \log_e K$ Eq. [14]

P = internal pressure
Q = B.t.u. per sq. ft. per hr. through any cylindrical surface of radius r. For a given heat flow into a length of tube, Qr = const.

$4^1/_4$ in. I.D., K would be 1.173, and $2P/b$, by interpolation, would be 0.057; then

$$b = \frac{2P}{0.057} = 1.4\,Qr$$

With P known as 1000,

$$Qr = \frac{2000}{0.057 \times 1.4} = 25{,}000$$

$$Q = \frac{25{,}000}{2.5} = 10{,}000 \text{ at the outside}$$

with Qr known[5] as 25,000,

$$P = \frac{1.4 \times 25{,}000 \times 0.057}{2} = 1000$$

An example is now given, taken from experience, of the occasional importance of axial stresses.

Carbon-steel tube 4 in. O.D., $3^1/_4$ in. I.D., span 18.58 ft.

Weight of tube and contents, 322 lb. P = 400 lb. per sq. in.

Bending moment, $1/8 \times 322 \times 18.58 \times 12 = 8980$ in.-lb.

Section modulus, 3.55 in.^{3}

Compression in top fiber = $8980 \div 3.55 = 2520$ lb. per sq. in.

Tension at low point of inside wall = $2520 \times (3.25/4) = 2050$ lb. per sq. in.

For the existing conditions,

$E = 17.5 \times 10^6$; $a = 11.35 \times 10^{-6}$; $1/m = 0.3$;
$T = 28°$; $K = 1.23$; $m/2(m-1) = {}^5/_7$.

Total outside axial stress:

$-5/7 \times 11.35 \times 17.5 \times 28 \times 0.933$	=	—3700 from [8]
Bending stress	=	—2520
		—6220
$1/2 \times 400 \times 3.9$ = Axial pressure stress	=	780
		—5440

or 5440 lb. per sq. in. compression in upper fiber.

Total inside axial stress:

$5/7 \times 11.35 \times 17.5 \times 28 \times 1.067$	=	4250 from [10]
$1/2 \times 400 \times 3.9$	=	780
Bending stress	=	2050
Tension at low point of inside wall	=	7080 lb. per sq. in.

The creep stress of the material was 6600 lb. per sq. in. at the temperature of the inner wall, and the above tension exceeds it. The tube actually sagged in service. This instance is a demonstration of the existence of high axial stresses due to temperature effects, as the bending stresses are quite insufficient to account for the sagging.

All the foregoing formulas are based on the supposition that the heat input is uniform all over the outside tube surface. In actual tubes this is not so. The heat input is a maximum at points nearest the source of heat and diminishes to a minimum at points furthest from it. This case is being studied by C. O. Rhys. The work done to date appears to show, however, that the inside hoop stress, as calculated above, would still be the governing factor in design.

A study of the formulas shows that the rational design of thick-walled tubes requires a knowledge of a number of physical properties of the material used. The modulus of elasticity, Poisson's ratio, the coefficient of expansion, and the thermal conductivity all enter into the equations either directly or indirectly, and must all be known. The latter two are usually supplied by the manufacturer of the steel or alloy in question; the first two are seldom given. For the determination of the modulus of elasticity, there has recently been set up in our laboratory an Amsler hydraulic testing machine of 60,000 lb. capacity and an Amsler-Martens mirror extensometer with which changes in length of two-millionths of an inch can be measured. Equipment is also available for the determination of the modulus of elasticity at elevated temperatures as well as at atmospheric temperatures. No equipment is available at the moment for the determination of Poisson's ratio; it is planned, however, to install a torsion machine by means of which the modulus of rigidity may be accurately determined at both atmospheric and elevated temperatures. The modulus of rigidity and Young's modulus are all that are necessary to make possible the calculation of Poisson's ratio, and it is felt that this method, particularly at elevated temperatures, is far more feasible than through the determinations of lateral and longitudinal strains. The importance of Poisson's ratio, which is usually given as 0.3, may be judged from the statement that a reduction of this ratio to 0.2, which is the value given by some investigators for certain alloys at high temperatures, reduces the stresses set up by the heat input by 12 per cent.

One other factor is the tube-wall temperature. While it does not enter into the equations, it is of paramount importance. The tube-wall temperature naturally determines the safe working stress which the tube will withstand, as reference to the creep curve of the material will show very clearly. The prediction of tube temperatures could well form the subject of a separate report, but because of the complexity of this problem it can only be briefly touched upon in this paper.

In general, an oil-refinery furnace is designed to raise a given quantity of oil from a known entering temperature to a known final temperature. The heat-input rates are usually well established from past operating conditions, so that the temperature of the oil at points throughout the setting may be readily calculated. From these data the tube-wall temperature is calculated from the drop through the oil film for which numerous formulas have been given by other investigators; however, allowance must be frequently made for surface cracking, boiling, critical temperature conditions, etc., all of which may set up a gas film at the point of contact with the inner tube wall or with the layer of coke in the tube. In case a layer of coke exists in the tube, due allowance must be made for the temperature drop through this layer, which may vary in thickness from $^1/_{32}$ in. to as much as $^3/_4$ in., depending upon the time-temperature relationships, the cracking rate of the oil in question, and the length of time that the unit has been on stream.

These factors seriously affect the tube-wall temperature, and an attempt is now being made to correlate the data secured from recorded tube temperatures with the theoretical drops through the metal, coke, and oil film. For very high mass velocities of 700 lb. per sq. ft. per sec., with little coke formation, the outer tube wall may be only 60 to 80 deg. hotter than the oil for a heat input of 10,000 B.t.u. per sq. ft. per hr., but with mass velocities of 350 to 400, where coke formation is present, the tubes may be 200 deg. or more hotter than the oil.

As pointed out earlier in this report, the question of the existence or non-existence of temperature stresses is controversial.

The final step in the research program outlined in the second paragraph of this paper is the laboratory check of the stress formulas. While we are quite convinced of the existence of temperature stresses, we hold no brief for the theory and would

[5] $\frac{\text{Total heat input to tube, (B.t.u./hr.)} \times r \text{ (in.)}}{\text{Total cylindrical surface (sq. ft.) at radius } r}$

be the first to reject it could conclusive reasons be found for taking this step. Reference to the examples already given will show that with temperature stresses eliminated the remaining stress, i.e., that due to pressure, is approximately one-half of the sum of the two. This would mean that in some cases the use of thinner tubes would be permissible, with a very considerable monetary saving particularly in the case of high-chrome-nickel alloys. In other cases it would allow the employment of greatly increased heat inputs, with a resulting reduction in the cost of the furnace due to the lesser number of tubes required.

To check the mathematical theory of temperature stresses, equipment has been set up which will allow of testing to destruction tubes of various sizes and materials under varying conditions of temperature, pressure, and heat input.

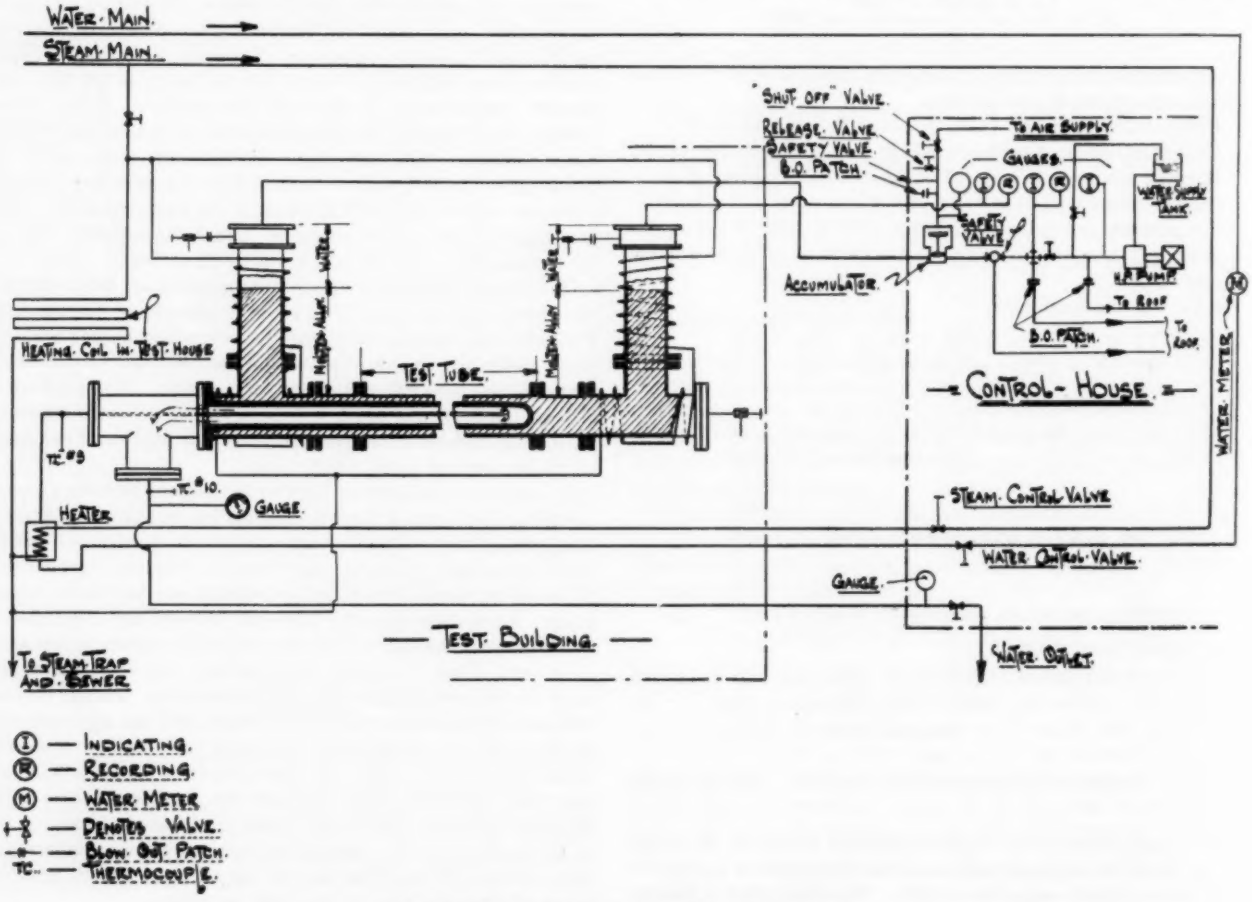


FIG. 7 SCHEMATIC DIAGRAM OF TEST EQUIPMENT

and serves to carry the heat away. Fig. 7 shows a schematic diagram of the test equipment. The pressure medium is a low-melting-point lead-bismuth-cadmium-tin alloy which is contained in the annular space between the test tube and the inner seamless tube, and in two containers which are a part of the high-pressure piping system. This alloy has a melting point of approximately 160 deg. fahr. and is completely liquid at approximately 200 deg. fahr., so that it must be kept hot at all times. For this purpose, the containers are steam jacketed, and a small current can be maintained in the electric-furnace heaters at all times. There are also extra heater plates underneath the fittings to furnish additional heat through the flanges and exposed metal. Pressure is applied on the liquid alloy by means of water from the high-pressure pump, the contact between the water and the metal being in these containers, a second container being used to transmit pressure on the liquid-metal system back through a water medium to gages in the control house. The high-pressure pump is a triplex, motor-driven pump designed for a maximum of 25,000 lb. per sq. in. with a capacity of 250 cu. in. per min. In the system is an air-loaded pressure accumulator with a displacement of approximately 28 cu. in. This system is protected by means of an adjustable 15,000- to 25,000-lb. safety valve as well as by two frangible disks which are now designed to blow at 14,000 lb. The electric furnace consists of nichrome ribbon wound on porcelain insulators, forming a square around the test section. The resistance winding is in four separate sections, the two sides and bottom being connected together in thirds of the furnace length, while the top heater is a separate unit.

The apparatus consists essentially of the test section heated by a radiant electric-resistance heater, with suitable controls and meters, a pressure medium, high-pressure connecting fittings, a cooling system, and a high-pressure pump. The test section is contained in a special reinforced-concrete building, while the pump and all the controls are in a separate building 50 ft. away.

The test section consists of an 8-ft. length of tube of the size under test, mounted in flanges and bolted up to special nickel-chromium alloy fittings designed to operate under a pressure of 3000 lb. at 1200 deg. fahr. under normal working conditions. Supported centrally inside the tube is a one-piece seamless steel tube which is welded closed at one end and flanged on the other end to form a seal between two adjacent fittings. Water circulates inside this tube, being brought in through a $^3/_4$-in. pipe,

Heat is transmitted through the steel tube and through the pressure medium to the inner steel tube, from which the heat is removed by circulating water. The heat flow through the test tube and the temperature of this tube are controlled within certain limits to produce temperature stresses. The outside surface temperatures of the tube are taken by means of eight thermocouples peened in the outer surface. No. 22 Chromel Alumel wire is used to obtain accurate temperature as it is the smallest size considered practicable for use under our particular conditions, and two No. 14 thermocouples are placed alongside two No. 22 to indicate the difference in temperature at the junction due to heat transmission along the thermocouple wires. The No. 14 wire is the size used in our regular service on cracking furnace tubes, and is used in this test merely to check the difference in indicated temperature. The present program covers a series of tests with outside tube temperatures ranging up to 1400 deg. fahr. and with a rate of heat flow through the tube ranging up to 30,000 B.t.u. per hr. per sq. ft. of outside tube surface. The pressure may be carried up to 14,000 lb. per sq. in.

While this equipment will give a rough check on the temperature-stress theory, its greatest value lies in its ability to determine the factor of safety of a tube against rupture. It will show, for example, that a commercial tube operating at temperature T with heat input Q and internal pressure P will not fail until the pressure reaches some multiple of P. It cannot check the equations accurately as these are all based on the elastic properties of the material, whereas this machine will only give bursting conditions.

To check the temperature-stress theory without exceeding the elastic limit of the material, a second testing device is being considered in which a thick tube would be heated from the inside and the strain on the outer wall measured by sensitive extensometers. With such apparatus the temperature stresses could be checked by comparison with those calculated directly from the formulas already derived.

At least part of the discrepancies between results obtained by French and those of the S. O. D. Co. are due to the great differences in the lengths of time in making the tests and the interpretation of the data. In Mr. French's work tests were carried on for 500 hr. or less, while in our own work as long as 6000 hr. was required to obtain the desired result.

Also steel obtained from different sources of supply, and even different heats may produce varying results. In this connection the author quotes from Mr. French's paper the following:

> One important point, which has not been given much consideration in the interpretation of flow tests, is that the numerical values obtained on one bar, lot, or melt of a given alloy may not coincide closely with the values obtained on other bars, lots, or melts. Similar variations in the properties of metals at atmospheric temperatures are well recognized and should likewise be given consideration in the application of flow-test data. The extent of these variations is not now accurately known but it has been shown that the differences in load-carrying ability, from heat to heat of steel, may be large. Without additional data, it is not safe to assume that the reported values represent the average, maximum, or minimum of the range characteristic of a given metal, and safety factors must still be employed.

In the work by Norton, he states that the creep loads may be relied upon to within ±25 per cent, and that relative values should be somewhat closer. Results obtained in our own laboratory may differ from those of other investigators by appreciable amounts and still be as accurate as it is possible to determine at the present time. This is because of the reasons given above.

With a full knowledge of safe stresses, well-defined values for the modulus of elasticity, Poisson's ratio, thermal conductivity, and coefficient of expansion, and laboratory-checked formulas for the calculation of stresses, the designer could proceed with the assurance that his designs were not only safe but economical as well. The program outlined is still far from completion, and for this reason the data contained in this report should be considered as preliminary as the report is not presented as a final one. Until the theory is disproved, however, it seems advisable to accept the existence of temperature stresses, as this is the safer procedure to follow if human life is to be safeguarded and damage to equipment reduced to a minimum.

ACKNOWLEDGMENT

To Mr. C. O. Rhys of the Process Division belongs entire credit for the mathematics given in the appendices to this paper. The derivation of the equations given are an abridgment and simplification of a more complete solution developed by Mr. Rhys during 1929, by which the stresses at any point in a tube may be determined. Mr. R. L. Duff in charge of the Metallurgical Division and his assistants supplied the description of the creep-point equipment and the test results secured with this equipment. Mr. A. J. Ely in charge of the Testing Division supplied the information on the equipment being developed for checking the mathematical theory.

Appendix No. 1—Derivation of Formulas for Temperature and Pressure Stresses in Tubes

TAKE a cross-section near the middle of a long circular tube. Let the temperature at any point be called T. At the inside surface $T = 0$, and at the outside surface $T = T_0$. There is thus a temperature drop through the tube from the outside to the inside. Also let there be an internal fluid pressure P.

Radial Strain. Before the temperature drop is established, let the radius to any point in the wall be called r. When the temperature drop exists, let this radius become $r + u$. Then an element of material, of radial dimension dr will change by du. If a be the coefficient of thermal expansion, and e the radial strain, the radial change of the element will be

$$a\,T\,dr + e\,dr$$

and we can write

$$du = a\,T\,dr + e\,dr$$

or

$$e = \frac{du}{dr} - aT$$

Hoop Strain. When r increases to $r + u$, the circumference, $2\pi r$ increases by $2\pi raT + 2\pi rf$, where f is the circumferential, or hoop strain. We thus have

$$u = raT + rf$$

or

$$f = \frac{u}{r} - aT$$

Axial Strain. When two similar insulated strips are first fastened together and then one of them is heated, compression sets up in one and tension in the other, with consequent bending of the strips. When such strips are replaced by long, thin, concentric tubes (strip length corresponds with tube-axis length) such bending cannot occur, and all fibers stretch axially by the same amount. If therefore v = axial displacement and z = axial length from some origin, it appears that dv/dz would be constant. Calling the axial strain g, we should have, as above,

$$g = \frac{dv}{dz} - aT$$

Strain and Stress Equations. Collecting from above,

$$e = \frac{du}{dr} - aT \quad \text{(Radial strain)}$$

$$f = \frac{u}{r} - aT \quad \text{(Hoop strain)}$$

$$g = \frac{dv}{dz} - aT \quad \text{(Axial strain)}$$

These are the strains in the three directions at right angles to each other. The dilatation is given by $D' = \frac{du}{dr} + \frac{u}{r} + \frac{dv}{dz} - 3aT$ (Love, "Mathematical Theory of Elasticity," p. 61), and the usual stress strain relations of the form

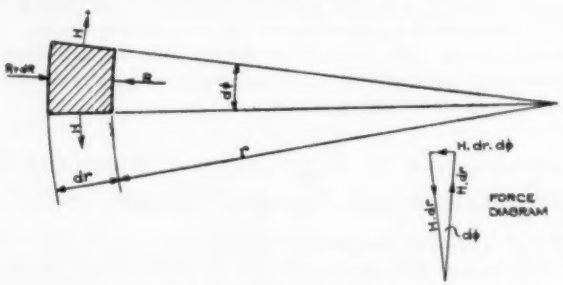


FIG. 8 ELEMENT OF TUBE WALL
(1 in. in length at right angles to section shown.)

$$R = \frac{2GD'}{m-2} + 2Ge$$

$$H = \frac{2GD'}{m-2} + 2Gf$$

$$Z = \frac{2GD'}{m-2} + 2Gg$$

can be written

$$\left.\begin{aligned} R &= \frac{2G}{m-2}\left[(m-1)\frac{du}{dr} + \frac{u}{r} + \frac{dv}{dz} - (m+1)aT\right] \\ H &= \frac{2G}{m-2}\left[\frac{du}{dr} + (m-1)\frac{u}{r} + \frac{dv}{dz} - (m+1)aT\right] \\ Z &= \frac{2G}{m-2}\left[\frac{du}{dr} + \frac{u}{r} + (m-1)\frac{dv}{dz} - (m+1)aT\right] \end{aligned}\right\} \quad [15]$$

where R, H, and Z are the radial, hoop, and axial stresses. G is the modulus of rigidity, or shearing modulus, and $1/m$ is Poisson's ratio.

Equations of Equilibrium. If we take an element of the tube wall as shown in Fig. 8, its equilibrium, in the plane of the paper, is given by

$$(R + dR)(r + dr)d\phi - Rrd\phi = Hdrd\phi$$

or

$$H - R = r\frac{dR}{dr} \quad [16]$$

Radial Displacement. From [15] and [16] we then have, remembering that dv/dz is a constant,

$$\frac{u}{r^2} - \frac{1}{r}\frac{du}{dr} = \frac{1}{m-2}\left[(m-1)\frac{d^2u}{dr^2} + \frac{1}{r}\frac{du}{dr} - \frac{u}{r^2} - (m+1)a\frac{dT}{dr}\right]$$

$$\frac{d^2u}{dr^2} + \frac{1}{r}\frac{du}{dr} - \frac{u}{r^2} = \frac{m+1}{m-1}a\frac{dT}{dr}$$

$$\frac{d}{dr}\left[\frac{1}{r}\frac{d}{dr}(ur)\right] = \frac{m+1}{m-1}a\frac{dT}{dr} \quad [17]$$

To integrate this, T must be expressed in terms of r. For cracking-coil work we require the "steady-state" relation, which is expressed by the well-known heat-conductivity equation

$$\frac{d^2T}{dr^2} + \frac{1}{r}\frac{dT}{dr} = 0$$

of which the solution is, for the boundary conditions

$$r = r_0, \quad T = T_0 \text{ (outside)} \quad r = r_1, \quad T = 0 \text{ (inside)}$$

$$T = \frac{T_0}{\log_e K}\log_e\frac{r}{r_1}, \qquad K = r_0/r_1$$

and

$$\frac{dT}{dr} = \frac{T_0}{\log_e K}\frac{1}{r} \quad [18]$$

This is sometimes called the temperature gradient.

Inserting [18] in [17] we have,

$$\frac{d}{dr}\left[\frac{1}{r}\frac{d}{dr}(ur)\right] = \frac{m+1}{m-1}\frac{aT_0}{\log_e K}\frac{1}{r}$$

$$\frac{d}{dr}(ur) = \frac{m+1}{m-1}\frac{aT_0}{\log_e K}r\log_e\frac{r}{r_1} + C_0r$$

$$u = \frac{m+1}{m-1}\frac{aT_0}{\log_e K}\frac{r}{2}\left(\log_e\frac{r}{r_1} - \frac{1}{2}\right) + \frac{C_1}{r} + C_0r \quad [19]$$

from which

$$\frac{du}{dr} = \frac{m+1}{m-1}\frac{aT_0}{\log_e K}\frac{1}{2}\left(\log\frac{r}{r_1} + \frac{1}{2}\right) - \frac{C_1}{r^2} + C_0 \quad [20]$$

and

$$\frac{u}{r} = \frac{m+1}{m-1}\frac{aT_0}{\log_e K}\frac{1}{2}\left(\log_e\frac{r}{r_1} - \frac{1}{2}\right) + \frac{C_1}{r^2} + C_0 \quad [21]$$

Evaluation of Constant C_1. By inserting [20] and [21] in the first of [15] and simplifying, we have, for the radial stress,

$$R = \frac{2G}{m-2}\left\{\frac{aT_0}{2\log_e K}\frac{m+1}{m-1}\left[m\left(\log_e\frac{r}{r_1} + \frac{1}{2}\right) - 1\right] + mC_0 + \frac{dv}{dz} - (m-2)\frac{C_1}{r^2} - \frac{(m+1)aT_0}{\log_e K}\log_e\frac{r}{r_1}\right\}$$

This is $-P$ (compression) at $r = r_1$, and 0 when $r = r_0$ so that we have, making these substitutions and subtracting:

$$P = \frac{2G}{m-2}\left[\frac{aT_0}{2}\frac{m+1}{m-1}m - (m-2)C_1\left(\frac{1}{r_0^2} - \frac{1}{r_1^2}\right) - (m+1)aT_0\right]$$

$$= 2G\left(C_1\frac{K^2-1}{r_0^2} - \frac{m+1}{m-1}\frac{1}{2}aT_0\right)$$

$$C_1 = \frac{r_0^2}{K^2-1}\left(\frac{P}{2G} + \frac{m+1}{m-1}\frac{1}{2}aT_0\right) \quad [22]$$

Hoop Stress at Outside. At the outside surface we have from [15]:

$$0 = \frac{2G}{m-2}\left[(m-1)\frac{du}{dr} + \frac{u}{r} + \frac{dv}{dz} - (m+1)aT\right]$$

$$H_0 = \frac{2G}{m-2}\left[\frac{du}{dr} + (m-1)\frac{u}{r} + \frac{dv}{dz} - (m+1)aT\right]$$

Subtracting,

$$H_0 = \frac{2G}{m-2}\left[-(m-2)\frac{du}{dr} + (m-2)\frac{u}{r}\right]$$

$$= 2G\left(\frac{u}{r} - \frac{du}{dr}\right)$$

Substituting in this from [20] and [21], putting $r = r_0$,

$$H_0 = 2G\left(\frac{2C_1}{r_0^2} - \frac{m+1}{m-1}\frac{aT_0}{2\log_e K}\right)$$

Inserting the value of C_1 from [22],

$$H_0 = 2G\left(\frac{1}{K^2-1}\frac{m+1}{m-1}aT_0 - \frac{m+1}{m-1}\frac{aT_0}{2\log_e K}\right) + \frac{2P}{K^2-1}$$

$$= G\frac{m+1}{m-1}aT_0\left(\frac{2}{K^2-1} - \frac{1}{\log_e K}\right) + \frac{2P}{K^2-1} \quad \ldots [23]$$

the outside hoop stress.

The term in P is due to pressure only, and the other part of temperature only. We call these F_0 and f_0, respectively.

Hoop Stress at Inside. At the inside surface the radial stress is a compression equal to the internal pressure. We can write

$$-P = \frac{2G}{m-2}\left[(m-1)\frac{du}{dr} + \frac{u}{r} + \frac{dv}{dz} - (m+1)aT\right]$$

$$H_1 = \frac{2G}{m-2}\left[\frac{du}{dr} + (m-1)\frac{u}{r} + \frac{dv}{dz} - (m+1)aT\right]$$

Subtracting,

$$H_1 = \frac{2G}{m-2}\left[(m-2)\frac{u}{r} - (m-2)\frac{du}{dr}\right] - P$$

$$= 2G\left(\frac{u}{r} - \frac{du}{dr}\right) - P$$

Substituting in this from [20] and [21], putting $r = r_1$,

$$H_1 = 2G\left(\frac{2C_1}{r_1^2} - \frac{m+1}{m-1}\frac{aT_0}{2\log_e K}\right) - P$$

Inserting value of C_1 from [22],

$$H_1 = 2G\left(\frac{2K^2}{K^2-1}\frac{m+1}{m-1}\frac{aT_0}{2} - \frac{m+1}{m-1}\frac{aT_0}{2\log_e K}\right) + \frac{2K^2P}{K^2-1} - P$$

$$= GaT_0\frac{m+1}{m-1}\left(\frac{2K^2}{K^2-1} - \frac{1}{\log_e K}\right) + P\frac{K^2+1}{K^2-1} \quad \ldots [24]$$

(the inside hoop stress.) The term in P is due to pressure only, and the other part to temperature only. We call these F_1 and f_1, respectively.

Axial Stresses. By the usual stress-strain equations, we can write the axial strain at the outside and inside surfaces. Consider temperature stresses only, or $P = 0$.

$$\left.\begin{aligned}\frac{dv}{dz} &= \frac{Z_0}{E} - \frac{H_0}{mE} + aT_0\\ \frac{dv}{dz} &= \frac{Z_1}{E} - \frac{H_1}{mE}\end{aligned}\right\} \quad \ldots [25]$$

Subtracting and transposing, since dv/dz is constant,

$$Z_1 - Z_0 = \frac{1}{m}(H_1 - H_0) + EaT_0 \quad \ldots [26]$$

From [23] and [24],

$$H_1 - H_0 = 2GaT_0\frac{m+1}{m-1} \quad \ldots [27]$$

Inserting this in [26], noting that $E = \frac{2G(m+1)}{m}$,

$$Z_1 - Z_0 = 2GaT_0\frac{m+1}{m(m-1)} + 2GaT_0\frac{m+1}{m}$$

$$= 2GaT_0\frac{m+1}{m}\left(\frac{1}{m-1} + 1\right)$$

$$= 2GaT_0\frac{m+1}{m+1}$$

$$Z_1 - Z_0 = H_1 - H_0 \text{ from [27].}$$

This last relation shows us by [26] that Z_1, Z_0, can be interchanged with H_1, H_0 in [25], or $Z_1 = H_1$ and $Z_0 = H_0$, or the temperature parts of the hoop and axial stresses are the same at the inside and outside surfaces.

The axial stress due to pressure in a tube closed at the ends is

$$\frac{P\pi r_1^2}{\pi(r_0^2 - r_1^2)} = \frac{P}{K^2-1}$$

Summary. Collecting from the above, we can write the equations for the stress in terms of E, Young's modulus, instead of G, the shearing modulus.

$$H_1 = \frac{maET_0}{2(m-1)}\left(\frac{2K^2}{K^2-1} - \frac{1}{\log_e K}\right) + P\frac{K^2+1}{K^2-1} \quad \text{(Inside hoop stress)}$$

$$H_0 = \frac{maET_0}{2(m-1)}\left(\frac{2}{K^2-1} - \frac{1}{\log_e K}\right) + \frac{2P}{K^2-1} \quad \text{(Outside hoop stress)}$$

$$Z_1 = \frac{maET_0}{2(m-1)}\left(\frac{2K^2}{K^2-1} - \frac{1}{\log_e K}\right) + \frac{P}{K^2-1} \quad \text{(Inside axial stress)}$$

$$Z_0 = \frac{maET_0}{2(m-1)}\left(\frac{2}{K^2-1} - \frac{1}{\log_e K}\right) + \frac{P}{K^2-1} \quad \text{(Outside axial stress)}$$

$$R_1 = -P \quad \text{(Inside radial stress)}$$

$$R_0 = 0 \quad \text{(Outside radial stress)}$$

In the above, the formulas for temperature and pressure stresses are those of Lorenz and Lamé, respectively.

Appendix No. 2—Derivation of Formulas for Calculating the Optimum Tube

THE heat input through a unit area is given by the temperature gradient multiplied by the coefficient of heat conductivity k.

Calling the heat input Q, and taking the temperature gradient from Equation [18], Appendix No. 1,

$$Q = \frac{kT_0}{\log_e K}\frac{1}{r}$$

or

$$T_0 = \frac{Qr \log_e K}{k}$$

For a given total-heat input to a circular tube, Qr is constant. We can then write, for the inside hoop stress (see Summary, Appendix No. 1),

$$H_1 = \frac{maEQr}{2(m-1)}\frac{\log_e K}{k}\left(\frac{2K^2}{K^2-1} - \frac{1}{\log_e K}\right) + P\frac{K^2+1}{K^2-1}$$

$$= \frac{maEQr}{2(m-1)k}\left(\frac{2K^2 \log_e K}{K^2-1} - 1\right) + P\frac{K^2+1}{K^2-1}$$

Differentiating with respect to K,

$$\frac{dH_1}{dK} = \frac{maEQr}{2(m-1)k}\left(\frac{(K^2-1)(4K\log_e K + 2K) - 4K^2\log_e K}{(K^2-1)^2}\right) - \frac{4PK}{(K^2-1)^2}$$

Equating to 0 and simplifying

$$\frac{maEQr}{2(m-1)k}(K^2 - 1 - 2\log_e K) - 2P = 0 \ldots\ldots [28]$$

Putting $\frac{maEQr}{2(m-1)k} = b$, we have the condition for minimum inside hoop stress,

$$\frac{2P}{b} = K^2 - 1 - 2\log_e K \ldots\ldots\ldots\ldots [29]$$

from which Table 4 has been calculated. The following values have been taken for cracking-coil conditions:

	m	a	E	k
	pure	$\frac{1}{\text{deg.}}$		$\frac{\text{B.t.u.} \times \text{in.}}{\text{sq. ft.} \times \text{hr.} \times \text{deg.}}$
Units...	number		lb. per sq. in.	
Steel...	4	10×10^{-6}	23×10^6	330
KA2S..	3½	11.5×10^{-6}	24.5×10	144

which give $b = 0.47\,Qr$ for steel
$b = 1.4\,Qr$ for KA2S.

Since $\frac{Qr}{k} = \frac{T_0}{\log_e K_1}$, we can write [28] as

$$\frac{maET_0}{2(m-1)}\left(\frac{K^2-1}{\log_e K} - 2\right) - 2P = 0$$

or

$$\frac{maET_0}{2(m-1)}\left(\frac{2}{K^2-1} - \frac{1}{\log_e K}\right) + \frac{2P}{K^2-1} = 0$$

or the outside hoop stress = 0 and the outside axial stress reduces to $P/(K^2-1)$, the axial stress due to pressure only. The inside axial stress is readily shown to be $H_1 - PK^2/(K^2-1)$ from the summary, Appendix No. 1. For the optimum tube we thus have:

Inside hoop stress:	$H_1 = bD\log_e K + PB$ (See Table 2)
Outside hoop stress:	$H_0 = 0$
Inside axial stress:	$Z_1 = H_1 - \frac{PK^2}{K^2-1}$
Outside axial stress:	$Z_0 = \frac{P}{K^2-1}$ or $\frac{PA}{2}$
Inside radial stress:	$R = -P$
Outside radial stress:	$R = 0$

or, when H_1 is a minimum, three of the other stresses are less than H_1 and the remaining two are zero.

FSP-53-13

High-Burned Kaolin Refractories

By F. H. NORTON,[1] CAMBRIDGE, MASS.

The development of high-fired kaolin refractories is briefly described. The various products made from kaolin such as brick, tank blocks, light-weight brick, and insulators, are discussed from the point of view of their physical characteristics. Also, the service results obtained with these products in typical installations for the past ten years are outlined. The results as a whole show that kaolin, when burned to a sufficiently high temperature, will make a refractory possessing very desirable qualities.



KAOLIN is the decomposition product of feldspar rock. It can be divided into two main classes, residual and sedimentary. The residual kaolin is found in the same position as the rock which it displaces, and contains, therefore, a considerable amount of undecomposed particles of quartz and other minerals which were present in the original rock. The sedimentary kaolin, however, has been washed out of the previous type of deposit and has settled to form beds in some ancient lake or sea. While this material was being transported, all of the larger particles were left behind, so that the kaolin itself is comparatively pure as found in the best beds and contains few non-clay minerals.

Probably the most important deposits of residual kaolin occur in the southern United States. While some of these deposits are unsatisfactory for making refractories because of a lowered melting point, the majority are remarkably pure and uniform. The deposits generally occur in layers of 10 to 40 ft. in thickness, and usually there is an overburden of 5 to 30 ft. Of course, the deposits with the least overburden are generally the most economical to work. The mining of this kaolin is comparatively simple as it is only necessary to strip off the overburden and take the clay out with steam shovels, the bed being so soft that blasting is unnecessary. It is also unnecessary to do any hand selecting as is the common practice in many fireclay mines, because here the kaolin is of great uniformity.

A typical analysis of a Southern sedimentary kaolin is given in the following table (it will be noted that the amount of fluxing elements is very low):

Silica	51.96
Alumina	45.38
Iron oxide	0.57
Titanium oxide	1.73
Calcium oxide	0.10
Magnesium oxide	0.22
Sulphur trioxide	0.04
Alkalies	0.00

The kaolin itself consists in large part of a clay mineral, but there is associated with it in very small amounts a number of minerals such as quartz and rutile.

The physical properties of this kaolin are also of considerable interest. The melting point is 3200 deg. fahr. in an oxidizing atmosphere. The shrinkage and porosity are given in Fig. 1. To show the comparatively large and continued shrinkage of the kaolin there are also plotted similar curves for a high-grade Pennsylvania fireclay.

The main advantage of the kaolin as a refractory material is its extreme purity and uniformity, which permit the manufacture of a high-grade refractory which will not vary an appreciable amount as received from month to month or year to year. Kaolin, however, has a high shrinkage on firing, which until recently had prevented the use of this material for a high-grade refractory. It was not until the development of practical methods of burning at temperatures around 3000 deg. fahr. that successful high-grade refractories could be made from this material. At present, however, kaolin refractories are being manufactured on a large scale.

Treatment of the Raw Material

The shrinkage of the kaolin is completed by burning it nearly to the melting point, which also reduces the porosity practically to zero. This hard-burned material is then crushed into grog of the proper fineness for the purpose, and is then bonded together with some of the raw kaolin and again burned at a high temperature. Years of experimenting have led to the belief that a good refractory must have a structure which is in equilibrium at all temperatures up to the highest that it would experience in use. This means that the brick to be stable must be made of pure homogeneous material and be fired at a temperature higher than the temperature of use.

Standard Refractory Bricks

The standard brick is made of a rather coarse kaolin grog bonded together to give a well-knitted and continuous structure, but at the same time to allow sufficient flexibility to give good resistance to spalling. The structure is well shown in the photomicrograph in Fig. 2. This brick has been used where temperatures or loads are high, such as in boiler furnaces, reheating and forge furnaces, enamel furnaces, oil stills, kilns, etc. This brick has the advantage over the usual fireclay brick in that furnaces can be run at higher temperatures, higher loads can be supported by the hot refractories, and more insulation can be safely used. Under conditions where the brickwork is subjected to a flowing slag, it has been found in many cases that the kaolin brick does not show as large a margin of superiority over fireclay brick as under other conditions.

As an example of service, the kaolin bricks were installed in the solid front wall of the boilers of a large central power station several years ago, and have since been giving good service. These boilers are fired by underfeed stokers. The life of the kaolin brick is from 15 to 18 months, while fireclay brick averages 2 months, and ventilated silicon-carbide walls average 3 to 4 months.

Similar results were obtained at another large central station

[1] Assistant Professor of Ceramics, Massachusetts Institute of Technology. Professor Norton was graduated from Massachusetts Institute of Technology in 1918. He was Assistant Physicist and Chief Physicist for the National Advisory Committee for Aeronautics until 1923. He then joined the Babcock & Wilcox Company for research and development work in ceramics. In 1929 he was appointed to his present position. Professor Norton is a member of the American Ceramic Society and a Fellow of the American Physical Society. He has published a number of papers dealing with refractories.

Presented at the French Lick Meeting, French Lick, Ind., October 13 to 15, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

with underfeed stokers, the kaolin brick being operated 18 months with no trouble, while other refractories have given trouble by the adhering slag tearing off the surface. The kaolin brick has found wide use in oil-fired stationary boilers, and gives on an average of three to five times the life of first-class fireclay brick. Fig. 3 shows the bridge wall of an oil-fired boiler made of kaolin bricks after 4600 hours of service.

A large majority of all the American ships with oil-fired water-tube boilers use the kaolin brick because of the comparatively long life and freedom from shutdowns.

The high load capacity of this brick is illustrated by its wide use in enameling furnace piers. A large proportion of the continuous enameling furnaces in this country use kaolin brick in the piers. In one periodic enameling furnace this brick has given good service in the piers for over seven years.

The kaolin bricks find considerable use in reheat and forge furnaces because they allow a higher furnace temperature, and therefore an increased capacity for a given furnace. They also permit considerable insulation of the walls and crown, which in turn means a considerable fuel saving. In the case of kilns for burning refractories, terra cotta, etc. the kaolin brick has been found to give good service in fireboxes, piers, floors, and lower walls. In general they give a life of three to four times the life of the average firebrick without distortion or spalling.

Tank Blocks

The structure of the tank blocks made from kaolin is somewhat different from that of the standard brick as shown in the photomicrograph of Fig. 4. Here the porosity is reduced and

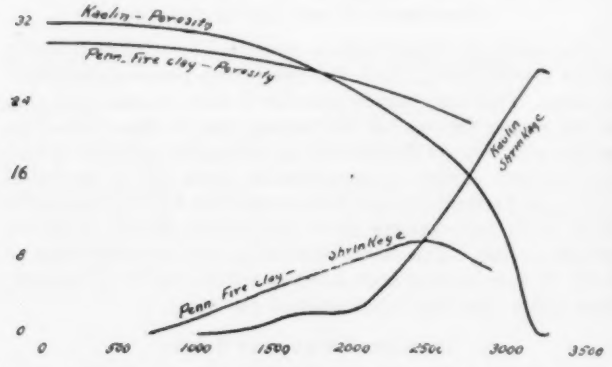


Fig. 1 Chart Showing Shrinkage and Porosity of Kaolins and Fireclays

the grog grains are completely knitted together to form a continuous structure. However, enough pores remain to give a slight amount of flexibility to the structure, so that there is no danger of cracking the blocks when heating up the tank. This structure is unique in being homogeneous and yet not too sensitive to spalling and cracking. The homogeneous structure has the great advantage of permitting the block to wear down with a smooth and even surface. No pits or grooves are formed, as is usually the case with most fireclay blocks, and because of this, seeds and cords are eliminated from the glass.

Some of these blocks were used in a tank making flint glass. The results showed that the kaolin blocks give a life of three to five times that of good fireclay blocks. Another set of kaolin blocks was used with boro-silicate glass in a small tank, and these are still giving good service. These blocks after one month's service are shown in Fig. 5; the freedom from cracks and grooves is noticeable.

Other blocks have been used with opal glass, and have shown a life of about one and one-half times the best of any of the other



Fig. 2 Photomicrograph of Thin Section of Kaolin Brick. ×60, Crossed Nicol Prisms

(The clear black areas are voids. The mottled appearance of the fired particles is due to the formation of very fine mullite crystals, which crystallize with the same orientation as the original kaolinite particles.)



Fig. 3 Boiler Bridge Wall of Kaolin Bricks After 4600 Steaming Hours

blocks. It was particularly interesting to note that they showed no cracks and no grooving, as indicated in Fig. 6, which shows a tank after a 33-week run.

Light-Weight Brick

There is a considerable demand for a brick that will withstand the severe conditions of oil-burning boilers, and yet be light in order to reduce the weight of the propulsive equipment of ships. A kaolin brick of this type has been developed which weighs not more than 5¼ lb., as compared with the 7 lb. for the average firebrick. This brick has a high porosity, but is strongly bonded and will give service equal to or better than the average fireclay brick. Some of these light-weight brick have been in service 18 months in destroyer boilers and are still in good condition. A kaolin-base cement is used to coat the surface of the wall in order to close the pores and prevent penetration of the oil slag into the brick.

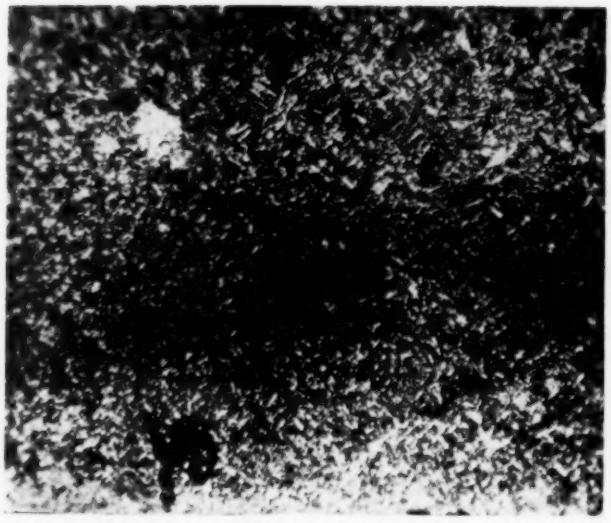


FIG. 4 PHOTOMICROGRAPH OF THIN SECTION OF KAOLIN GLASS TANK BLOCK. ×60, CROSSED NICOL PRISMS

(The structure is much denser than that of Fig. 2, and the mullite crystals are much larger. The light-colored particle is a small bleb of corundum.)



FIG. 5 EXPERIMENTAL TANK WITH THREE KAOLIN BLOCKS

(After one month's service with boro-silicate glass.)

INSULATING BRICK

An insulating brick has been developed using kaolin, which, though still somewhat in the experimental stage of manufacture, has been thoroughly tried out. When this brick is used in furnaces, it takes the place of the refractory and at the same time has a high degree of insulating value. This unique property limits the high-temperature zone close to the inner surface and permits unusual furnace efficiencies. Two small furnaces were made up exactly alike, one with fireclay brick and one with the kaolin insulator. With the same burner the first one reached the maximum temperature of 1900 deg. fahr., and the second a temperature of 2600 deg. fahr. Some of these bricks have been used in furnaces which were suddenly heated and cooled as much as 200 times, with maximum temperatures ranging from 2000 to 2900 deg. fahr., with no signs of cracking or deterioration. It should be clearly understood, however, that these bricks will not withstand slag. This condition would of course be expected from their high porosity.

CRUCIBLES

Carbon-free crucibles are being manufactured of kaolin for melting glass, enamels, and metals. A fine-grained, dense structure is used with about 8 per cent porosity, but the flexibility is kept high enough to prevent cracking, even on rapid heating. These crucibles will withstand reheating many times without cracking, and may be used for melting glass, enamels, or metals without danger of contaminating the melt. In melting alloy steel in high-frequency furnaces as many as 250 heats have been obtained from one crucible by lining the crucible with kaolin cement and relining at intervals whenever the cement became thin.

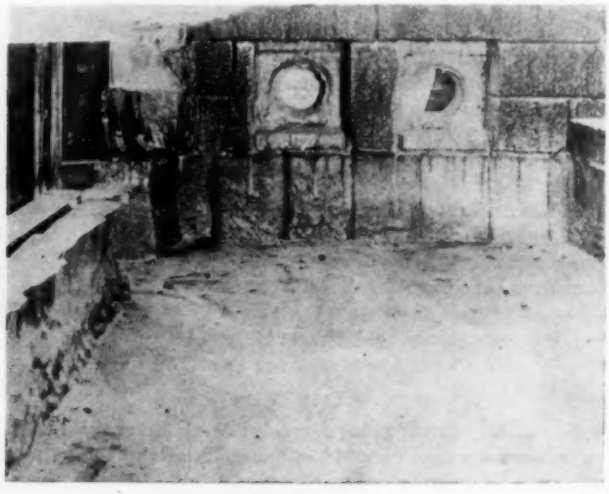


FIG. 6 A COMPLETE TANK OF KAOLIN

(After 33 weeks' run on opal glass.)

TABLE 1 PHYSICAL PROPERTIES OF VARIOUS TYPES OF KAOLIN REFRACTORY

Property	Kaolin tank block	Standard kaolin brick	Light weight kaolin brick	Kaolin insulating refractory	Average of three first-quality Pennsylvania bricks
Weight of standard brick, lb.	8.4	7.3	5.25	1.6	7.0
Density, lb. per cu. ft.	145	126	95	27.5	1.30
Porosity, per cent	10	25	40	85	28
Start of deformation under load of 25 lb. per sq. in., deg. fahr.	2825	2750	2450	....	2100
Deformation of 10 per cent under load of 25 lb. per sq. in., deg. fahr.	3000	2900	2620	....	2725
Start of permanent volume change without load, deg. fahr.	3000	2900	2900	2700	2300
Mean coefficient of expansion per deg. fahr.	0.0000024	0.0000024	0.0000024	0.0000024	0.0000028
Cycles in 2900 deg. fahr. air-spalling test	....	20	...	...	14
Melting point, deg. fahr.	3200	3200	3200	3200	3065
Corresponding cone	35+	35+	35+	35+	31
Thermal conductivity at 1000 deg. fahr.	16	12	...	1.9	9
Thermal conductivity at 2000 deg. fahr.	19	13	...	3.1	11

CEMENTS

A number of types of cement have been developed with a kaolin base. These cements include high-temperature cements with a kaolin bond, a low-vitrifying cement with a low-vitrifying clay bond, air-setting cement, and patching cement with a low fire shrinkage.

FSP-53-14

Comparative Physical Properties of Chromium-Nickel, Chromium-Manganese, and Manganese Steels

By C. L. CLARK[1] and A. E. WHITE,[2] ANN ARBOR, MICH.

This paper gives the results of short-time tensile tests at 75 and 1000 deg. fahr., and creep tests at 1000 deg. fahr., on certain selected steels. The steels investigated belong to either the chromium-nickel, the chromium-manganese, or the manganese series. Certain other alloying elements as silicon and tungsten were also present. The chromium-nickel steel specimens were taken from both bar and tube stock, and certain of them were tested in both the hot-rolled and water-quenched conditions.

The results indicate that the substitution of manganese for nickel in chromium-nickel steels is detrimental, at least in so far as the load-carrying ability of the alloy is concerned. Also that while the creep resistance of the manganese and manganese-tungsten steel at 1000 deg. fahr. is superior to that of many pearlitic steels, it is not equal to that of the Enduro KA2 type of alloy.

WITH the advancing trend in operating temperatures in many industries, the need for metals to successfully withstand not only the necessary stresses but also the oxidizing atmosphere is becoming more pronounced. For temperatures up to a certain maximum, 1000 or 1100 deg. fahr., the so-called pearlitic steels may be successfully used. Certain of them, when in the proper heat-treated condition, possess remarkable strength at the higher temperatures and are able to resist to the necessary extent the corrosive action of the gases with which they are in contact.



C. L. Clark

For temperatures much above 1000 deg. fahr. the effect of any beneficial heat treatment is largely lost, and the strength of all pearlitic steels decreases rapidly. In this temperature range the austenitic steels display superior characteristics both in regard to load-carrying ability and resistance to oxidation. The dividing line between these two zones is not definitely marked, and in certain applications pearlitic steels can be used above 1000 deg. fahr., and in many, austenitic steels are used below 1000 deg. fahr.

The types of austenitic steels commonly used for high-temperature service are not diversified, commercial practice confining itself almost completely to those chromium-nickel steels known as Enduro KA2. On the basis of our present knowledge, the confinement to this particular series of steels is not unwarranted, for they do possess the outstanding properties for the majority of applications. These steels, however, also possess certain undesirable characteristics, and substitute steels, free from these disadvantages and yet possessing the valuable properties of Enduro KA2, would be highly welcomed. Whether or not certain other austenitic steels would fulfil these conditions is unknown, at least from the information available in the literature.



A. E. White

The work reported in this paper was undertaken to determine the effect of certain changes in composition upon the physical properties of steels of the Enduro KA2 type. Also in order to ascertain the truth of the statement sometimes made that all austenitic steels should exhibit nearly the same properties at high temperatures, regardless of the alloying elements present, the investigation was extended to include a straight 14 per cent manganese steel, and a 14 per cent manganese, 1.0 per cent tungsten steel.

MATERIALS

The alloys investigated belong to either the chromium-nickel, the chromium-nickel-silicon, the chromium-manganese-silicon, the manganese, or the manganese-tungsten series. Their compositions and the condition in which they were tested are given in Table 1.

The first four steels in the table were secured from commercial sources, while the remaining four were made at the University of Michigan laboratories. Those designated as KA2 and RS were received in 1-in. and $^3/_4$-in. round bars, respectively, while KA2S-H and KA2S-R were obtained in the form of a 5-in.-diameter tube having a wall thickness of approximately $^3/_4$ in. As shown in the table, one section of this tube was water quenched and pickled, while the remaining was hot rolled.

The steels prepared in the U. of M. laboratories were made in a 35-lb. Ajax high-frequency induction furnace. The melts were cast into tapered ingots approximately $2^1/_2$ in. square at the top, and these were forged into $^3/_4$ in. round bars by the Atlas Drop Forge Company. In order to maintain the carbon content as low as possible, Armco iron was used as a base in melting.

PROCEDURE

The two temperatures chosen for the comparison of the ten-

---

[1] University of Michigan. Mr. Clark received the degrees of B.S. in Chemical Engineering in 1925, M.S. in 1926, and Ph.D. in 1928 from the University of Michigan. He has specialized in metallurgy and is now Research Engineer in the Department of Engineering Research at the University of Michigan.

[2] Director of the Department of Engineering Research and Professor of Metallurgical Engineering at the University of Michigan. Mem. A.S.M.E. Professor White received his B.A. from Brown University in 1907 and his Sc.D. in 1925. After graduation he served a few years with the Jones and Laughlin Steel Company, Pittsburgh, Pa. He has been at the University of Michigan since 1911, except for two years spent with Ordnance Department, U. S. A., during the World War.

Contributed by the Joint Research Committee on Effect of Temperatures on the Properties of Metals and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

TABLE 1 CHEMICAL COMPOSITION AND HEAT TREATMENT OF STEELS INVESTIGATED

Designation	Type of alloy	Heat treatment	Chemical composition, per cent C	Mn	Si	Cr	Vi	W
KA2	Ni-Cr	Water quenched	0.09	0.40	0.57	18.15	9.65	..
KA2S-H	Ni-Cr	Water quenched	0.05	0.37	0.68	18.28	9.35	..
KA2S-R	Ni-Cr	Hot-rolled	0.05	0.37	0.68	18.28	9.35	..
RS	Ni-Cr-Si	Water quenched	0.022	0.494	2.35	18.00	10.26	..
A1	Mn-Cr-Si	Water quenched	0.158	6.32	2.58	18.02	...	..
B1	Mn-Cr-Si	Water quenched	0.110	9.78	2.81	18.72	...	..
A4	Mn	Water quenched	0.072	15.40	..	...	...	..
A5	Mn-W	Water quenched	0.076	14.42	..	...	...	1.23

sile characteristics of these steels were room temperature (75 deg.) and 1000 deg. fahr. The experimental work consisted of short-time tensile tests at each of these temperatures, and long-time creep tests at 1000 deg. fahr.

The method used for the determination of the short-time tensile properties was the same as that described in previous papers by the authors.[3,4] The deformation produced by each increment of stress is measured by Ames dials reading to one-ten-thousandth (0.0001) of an inch.

The apparatus employed in the creep tests is shown in Fig. 1. This is fully described in a paper presented at the Semi-Annual Meeting of The American Society of Mechanical Engineers at Detroit[5] last June. The deformation produced by a fixed load is measured by the rotation of two mirrors mounted on rollers. Readings to 2.8 millionths (0.0000028) of an inch are possible.

In performing the creep tests, a fixed load is applied and maintained constant until either the flow comes to a complete stop or until it has reached a steady state. The load is then either increased or decreased, depending on the magnitude of the rate of deformation under the previous load.

RESULTS

Short-Time Tensile Tests

The short-time tensile properties of these steels were determined at 75 and 1000 deg. fahr. The results are given in Tables 2 and 3.

Tests at 75 Deg. Fahr. The results of the tests at 75 deg. fahr. are given in Table 2.

TABLE 2 TENSILE PROPERTIES OF SELECTED STEELS AT 75 DEG. FAHR.

Designation	Type composition	Tensile strength, lb. per sq. in.	Proportional limit, lb. per sq. in.	Elongation in 2 in., per cent	Reduction of area, per cent
KA2	0.05 C, 18 Cr, 8 Ni	100,000	25,000	60.0	63.0
KA2S-H	0.05 C, 18 Cr, 8 Ni	82,275	16,250	67.5	72.3
KA2S-R	0.05 C, 18 Cr, 8 Ni	82,300	17,500	43.5	68.0
A1	0.05 C, 18 Cr, 6 Mn, 3 Si	92,200	54,000	22.5	44.0
B1	0.05 C, 18 Cr, 10 Mn, 3 Si	121,450	35,000	35.0	37.3
A4	0.05 C, 14 Mn	84,850	9,000	[a]	[a]
A5	0.05 C, 14 Mn, 1 W	77,750	8,000	[a]	[a]

[a] Specimen broke outside of gage marks.

The KA2 bar material possesses higher strength properties than does the steel of similar composition, but with a lower carbon content, taken from tubes. With the ductility values the converse is true. These differences are partly due to the variation in carbon content and to the manner in which the metal was fabricated.

The effect of heat treatment on the properties of the chromium-nickel alloys is obtained from a comparison of specimens KA2S-H and KA2S-R. The hot-rolled material possesses slightly higher strength characteristics, but its ductility is considerably lower than that of the water-quenched material. Especially is this true of the percentage of elongation in 2 in.

Steels A1 and B1, which differ from KA2 mainly in that the nickel has been replaced by 6.32 and 9.78 per cent of manganese, respectively, show this substitution to have resulted in increased strength and decreased ductility, with the latter being the more pronounced. As is the case with lower-alloyed steels, increasing the manganese from 6.32 to 9.78 per cent resulted in increased tensile-strength values, but in a decrease in the proportional limit. No consistent variation was obtained in the ductility, the higher-manganese steel showing the maximum elongation and the minimum reduction of area values.

Steels A4 and A5 differ from the preceding ones in that their principal alloying constituent is manganese. These steels possess a tensile strength comparable to that of the low-carbon Enduro KA2, but their remaining physical properties are much lower. Their ductility was so low that in each case fracture was obtained in the fillet of the specimen.

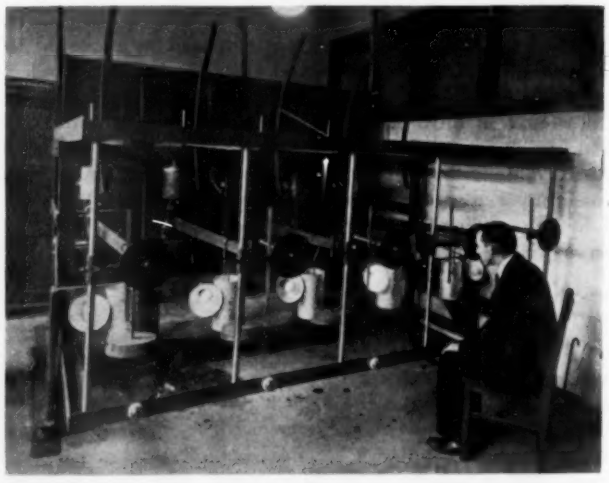


Fig. 1 Apparatus for Determining Rate of Creep of Metal

Tests at 1000 Deg. Fahr. The results of the tests at 1000 deg. fahr. are given in Table 3.

As at atmospheric temperature the higher-carbon bar stock KA2 has greater strength and lower ductility than the lower-carbon tube stock, these differences are the most marked in the case of the proportional-limit and percentage-of-elongation values. Also, as before, the quenched and hot-rolled materials differ from one another mainly in that the quenched material possesses the greater ductility, which is most noticeable in the percentage of elongation values.

TABLE 3 TENSILE PROPERTIES OF SELECTED STEELS AT 1000 DEG. FAHR.

Designation	Type composition	Tensile strength, lb. per sq. in.	Proportional limit, lb. per sq. in.	Elongation in 2 in., per cent	Reduction of area, per cent
KA2	0.05 C, 18 Cr, 8 Ni	62,050	14,000	40.5	68.6
KA2S-H	0.05 C, 18 Cr, 8 Ni	54,950	7,000	67.0	73.1
KA2S-R	0.05 C, 18 Cr, 8 Ni	54,600	8,000	43.5	67.0
RS	0.05 C, 18 Cr, 8 Ni, 3 Si	81,300	21,000	37.8	52.2
A1	0.05 C, 18 Cr, 6 Mn, 3 Si	43,100	13,000	29.0	62.3
B1	0.05 C, 18 Cr, 10 Mn, 3 Si	61,350	8,000	23.0	54.5
A4	0.05 C, 14 Mn	43,400	7,000	45.0	74.4
A5	0.05 C, 14 Mn, 1 W	50,200	8,000	36.0	72.4

Steel RS, which differs from the remaining chromium-nickel

[3] A. E. White and C. L. Clark, "Properties of Boiler Tubing at Elevated Temperatures Determined by Expansion Tests," Trans. A.S.M.E., vol. 48 (1926), p. 1075.

[4] A. E. White and C. L. Clark, "Stability of Metals at Elevated Temperatures," Trans. A.S.S.T., vol. xv (1929), p. 670.

[5] A. E. White, C. L. Clark, and L. Thomassen, "An Apparatus for the Determination of Creep at Elevated Temperatures," Preprint, Semi-Annual A.S.M.E. Meeting, Detroit, June 9–12, 1930.

steels in that it contains a higher amount of silicon, possesses the highest strength characteristics at this temperature of any of the steels considered. As would be expected, its ductility is somewhat lower.

The replacement of nickel of the chromium-nickel steels with 6.32 per cent of manganese resulted in decreased tensile strength and percentage of elongation values, but produced no marked effect in either the proportional limit or percentage reduction of area. Increasing the manganese content to 9.78 per cent, steel B1, raised the tensile strength but caused a marked lowering in the remaining physical properties.

The 14 per cent manganese steel, A4, has properties comparable to those of the low-carbon Enduro KA2 in the hot-rolled state. Its tensile strength is somewhat lower, but the remaining values are nearly identical. The addition of 1.23 per cent tungsten to this steel slightly increased the strength and lowered the ductility.

Creep Characteristics at 1000 Deg. Fahr.

Creep tests were conducted at 1000 deg. fahr. on all of these steels with the exception of the bar stock KA2 and the manganese-chromium-silicon steel, A1. The results are given in Figs. 2, 3, and 4, and in Tables 4 and 5.

Figs. 2 and 3 give the time-elongation curves of the chromium-nickel and manganese steels, respectively. In each of these tests a fixed load was applied and maintained constant until either the flow came to a stop or until it remained at a constant rate for at least 200 hours. Then the load was either increased or decreased, depending on the magnitude of the flow previously obtained.

Numerical results taken from these two figures are given in Table 4. The first three columns of this table are self-explanatory. The fourth gives the deformation occurring in what is commonly known as the first period of creep. This includes both elastic and plastic deformation, and also any elongation involved in the specimen's adjusting itself to the applied load. It does not include the relatively large amount of elastic deformation obtained immediately upon application of the load. The fifth column records the period of time required for this creep, while the sixth, which is obtained by dividing the figures in the fourth column by those in the fifth, gives the average rate of creep during this period. This value has no practical significance as the rate of creep is constantly diminishing. It is, however, a measure of the metal's strain-hardening ability.

The figures in the seventh column are of greatest importance, for they show at what rate the metal will deform when subjected to the given condition. These figures are obtained from the slope of the curves in Figs. 2 and 3 after a steady state of flow has been obtained.

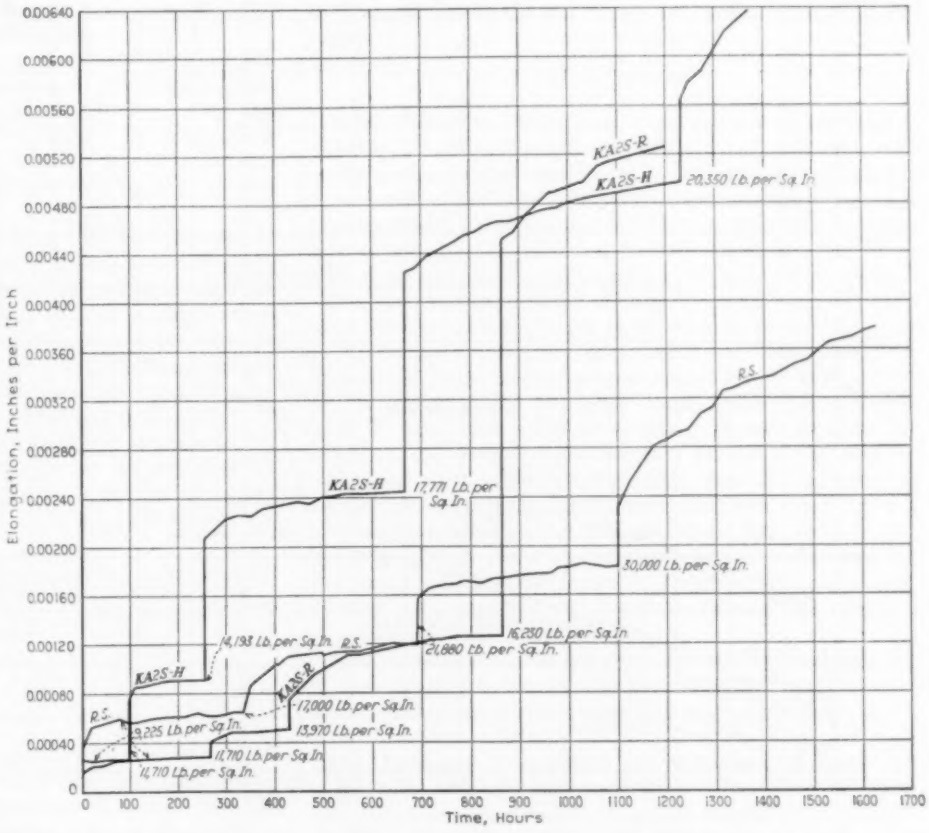


Fig. 2 Creep Characteristics of Chrome-Nickel Austenitic Steels at 1000 Deg. Fahr.

TABLE 4 FLOW CHARACTERISTICS AT 1000 DEG. FAHR. OF SELECTED STEELS

Designation	Type of alloy	Load per sq. in., lb.	Total deformation 1st period, in.	Duration of 1st period, hr.	Rate of deformation in 1st period, in. per hr.	Rate of deformation in 2nd period, per cent/1000 hr.
RS	0.05 C, 18 Cr, 8 Ni, 3 Si	9,225	0.000250	75.0	0.0000033	0.011
		17,000	0.000470	125.0	0.0000037	0.031
		21,800	0.000530	100.0	0.0000053	0.060
		30,000	0.001000	75.0	0.0000133	0.170
KA2S-H	0.05 C, 18 Cr, 8 Ni	9,225	0.000000	0.0	0.0000000	0.000
		11,710	0.000140	50.0	0.0000028	0.000
		14,193	0.000180	200.0	0.0000009	0.500
		17,771	0.000410	200.0	0.0000021	0.066
		20,350	.....	...	.....	Under 0.290
KA2S-R	0.05 C, 18 Cr, 8 Ni	9,225	0.000105	75.0	0.0000014	0.000
		11,710	0.000080	50.0	0.0000016	0.000
		13,970	0.000500	325.0	0.0000015	0.000
		16,230	0.000380	100.0	0.0000038	0.101
B1	0.05 C, 18 Cr, 10 Mn, 3 Si	11,900	0.000360	150.0	0.0000024	0.128
		13,300	0.000000	0.0	0.0000000	0.290
		15,850	0.000260	50.0	0.0000052	0.362
A4	0.05 C, 14 Mn	5,000	0.000040	25.0	0.0000016	0.000
		7,500	0.000060	25.0	0.0000024	0.000
		11,900	0.000650	175.0	0.0000037	0.092
		14,200	0.001640	225.0	0.0000073	0.272
		17,085	0.004400	175.0	0.0000251	0.992
A5	0.05 C, 14 Mn, 1 W	9,225	0.000360	100.0	0.0000036	0.000
		11,710	0.000400	100.0	0.0000040	0.000
		14,200	0.000560	175.0	0.0000032	0.056
		16,500	0.000760	175.0	0.0000043	0.165
		19,350	0.001420	150.0	0.0000094	0.360
		20,185	0.000880	100.0	0.0000088	0.496

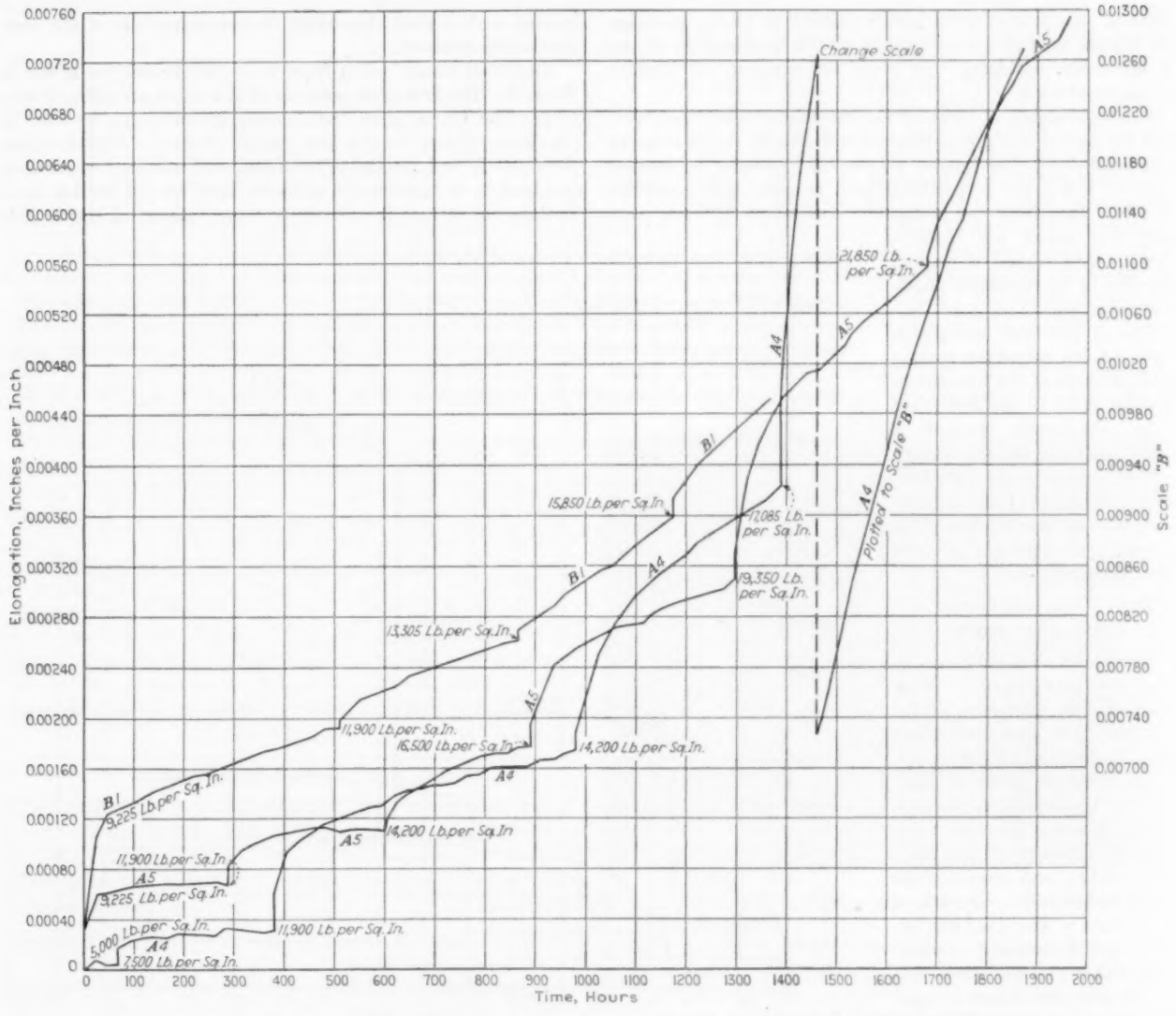


FIG. 3 CREEP CHARACTERISTICS OF MANGANESE AUSTENITIC STEELS AT 1000 DEG. FAHR.

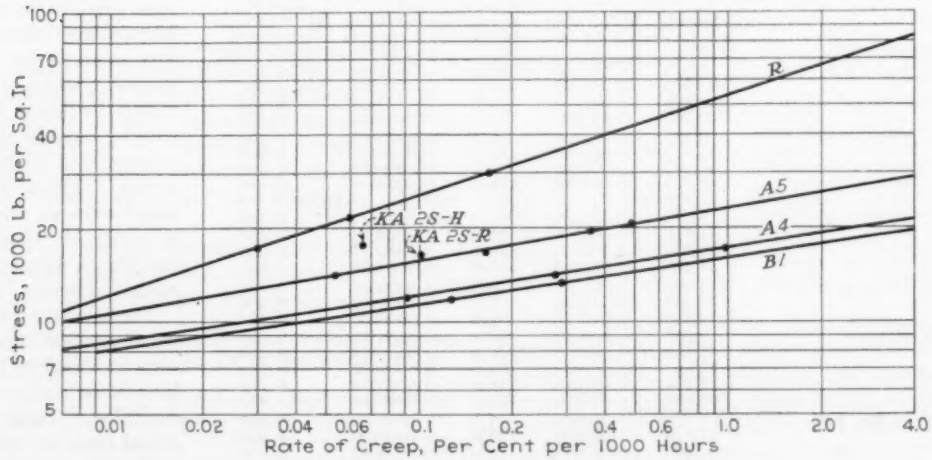


FIG. 4 LOGARITHM OF RATE OF CREEP VERSUS LOGARITHM OF STRESS

Previous work of several investigators has shown that if the logarithm of the rate of deformation be plotted against the logarithm of the stress producing the deformation, a straight-line relationship will be obtained under certain conditions. Fig. 4 shows the values plotted in this manner. It will be observed that practically straight lines are obtained in nearly all cases.

This type of chart is useful for two reasons: First, it allows various steels to be readily compared in so far as their load-carrying ability is concerned; and second, if the amount of allowable deformation is known, then the maximum stress to which a metal can be subjected is easily determined.

Considering the relative merits of the steels being investigated, the chromium-nickel alloys are seen to exhibit the greatest load-carrying ability at this temperature. The alloy of this type containing 2.35 per cent of silicon is superior to the specimens taken from tube stock. Results to date also indicate the water-quenched material to be somewhat superior to that in the hot-rolled conditions.

Replacing the nickel in Enduro KA2 with 9.78 per cent of manganese, steel B1, has resulted in a marked lowering of the metal's ability to withstand continuous loads at 1000 deg. fahr. The straight 14 per cent manganese steel, A4, is slightly superior to B1, while the addition of 1.23 per cent of tungsten has noticeably improved the load-carrying ability of the manganese steel. In fact, the properties of the manganese-tungsten alloy are comparable to those of the hot-rolled KA2 tube stock.

In Table 5 are given values taken from Fig. 4, showing the stresses required to produce deformation at the rate of 0.01, 0.10, and 1.0 per cent per 1000 hours. Values are not given for steels KA2S-H and KA2S-R as the tests on those are not as yet completed.

TABLE 5 STRESSES REQUIRED TO PRODUCE DESIGNATED RATES OF CREEP IN SELECTED STEELS AT 1000 DEG. FAHR.

Designation	Type	Per cent per 1000 hr. 0.01	0.10	1.0
		Lb. per sq. in.		
RS	0.05 C, 18 Cr, 8 Ni, 3 Si	12,000	25,000	56,000
B1	0.05 C, 18 Cr, 10 Mn, 3 Si	8,000	11,250	16,000
A4	0.05 C, 14 Mn	8,600	12,100	17,000
A5	0.05 C, 14 Mn, 1 W	10,600	15,750	23,000

The load of 12,000 lb. per sq. in. required to produce flow at the rate of 0.01 per cent per 1000 hours with the low-carbon high-silicon chromium-nickel alloy is comparable to the value of 15,000 lb. previously reported for Enduro KA2.

CONCLUSIONS

The results obtained from the short-time tensile and long-time creep tests allow certain conclusions to be drawn regarding the effect of composition, fabrication, and heat treatment on the high-temperature characteristics of the selected steels.

In so far as the short-time tensile properties at room temperature are concerned, the substitution of manganese for the nickel in the Enduro KA2 type of alloy results in a slightly increased strength and greatly decreased ductility. The tensile strength increases with the manganese content while the proportional limit decreases. Specimens of the chromium-nickel alloy taken from tube stock display less strength and greater ductility than those from the bar material. Whether or not this material is water-quenched or hot-rolled makes no appreciable difference in the strength properties, but does greatly affect its ductility as shown in the percentage of elongation, the quenched material possessing much higher values. The manganese steels proved to be extremely brittle.

The short-time tensile properties of the material at 1000 deg. fahr. arrange themselves in approximately the same order as at room temperature. The two straight manganese steels possess a good degree of ductility at this temperature, in fact, superior to that of the chromium-manganese alloys and comparable to that of the hot-rolled KA2.

The creep results at 1000 deg. fahr. indicate that the chromium-nickel alloy from the bar stock has greater load-carrying ability than those from tube stock, and that the water-quenched material is slightly superior to the hot-rolled material. Also that the substitution of the nickel with manganese has produced a marked lowering in the material's ability to withstand continuous stress. The 14 per cent manganese steels possess properties comparable to those of the chromium-manganese steel, while the addition of tungsten was found to increase its resistance to a marked degree.

Discussion

F. H. NORTON.[6] Professors Clark and White have given valuable and interesting data in their paper. On the whole their results verify the statement the writer made some time ago that all austenitic steels were in a separate class in regard to resistance to creep at high temperatures. At that time tests were available only on the nickel-chromium austenites, and these new data on manganese and manganese-tungsten steels are very welcome.

The authors show that all of these steels have a creep resistance of 8000 to 12,000 lb. per sq. in. for a 1 per cent elongation in 100,000 hours at 1000 deg. fahr., which may be compared with a resistance, under the same conditions, of only 3000 or 4000 lb. for carbon steel. At higher temperatures this difference would be even more marked. It would appear that the small differences found between the various austenitic steels tested for this paper might almost be within the experimental error, as the specimens were held at one load for only 300 hours.

The authors have plotted their results on logarithmic paper, and it is interesting to notice that their points, in all cases, fall on a straight line and that the average slope is 1 in 5, which is an excellent confirmation of the 5th power flow regime which the writer recently called attention to.

It would be most interesting if the authors continued the work on these steels up to higher temperatures, to determine whether the various austenitic steels showed the same similarity.

J. J. KANTER.[7] The authors report tension test values for 75 deg. fahr. and 1000 deg. fahr. These values afford an intercomparison of several austenitic alloys, several of the nickel-chromium alloys studied giving results comparable with those obtained at Crane Company laboratories. As is noted from the accompanying table (see Table 6), the results of the two laboratories are in fairly good agreement on similar grades and conditions of materials. There is a notable peculiarity brought out very clearly in the Crane Company results that bears emphasis. The proportional limits shown by the water-quenched material in general are considerably less than those given by the hot-rolled and hard-rolled conditions. That proportional limit at high temperatures cannot be taken as a criterion of long-time strength in this instance is demonstrated by the fact that creep stresses obtained on water-quenched KA2S are equal to those for the smaller-grained hot-rolled metal. Elsewhere[8] it has been shown that at higher temperatures than 1000 deg. fahr. creep stresses for water-quenched KA2S are considerably higher than those shown by the metal as it cools from rolling.

Even at 1000 deg. fahr., the extreme case of fine grain size inferiority is brought out by creep results on hard-rolled KA2 as compared with hot forged KA2. A very high elastic ratio is

[6] Massachusetts Institute of Technology, Cambridge, Mass.

[7] Crane Company, Chicago, Ill.

[8] J. J. Kanter and L. W. Spring, "Some Long-Time Tension Tests at Elevated Temperatures," A.S.T.M. Proceedings, 1930, and *Valve World*, Dec., 1930.

TABLE 6 TESTS OF SEVERAL AUSTENITIC ALLOYS, IN POUNDS PER SQUARE INCH

	Tested at 75° F.				Tested at 1000° F.									
	Tensile strength		Proportional limit		Tensile strength		Proportional limit		Creep stresses (per cent per 1000 hours) 0.01		0.10		1.0	
	U.M.[a]	C.Co.[a]	U.M.	C.Co.	U.M.	C.Co.	U.M.	C.Co.	U.M.	C.Co.	U.M.	C.Co.	U.M.	C.Co.
KA2, forged	....	90,000	....	26,000	....	56,500	....	21,500	....	17,000	....	22,000	....	33,000
KA2, hard-rolled	....	108,950	....	62,500	....	69,300	....	40,000	....	7,000	....	16,000	....	50,000
KA2, water-quenched	100,000	90,000	25,000	32,900	62,050	57,000	14,000	13,500	....	....	....	....	....	....
KA2S, hot-rolled	82,300	87,300	17,500	20,100	54,600	59,100	8,000	16,000	....	13,500	20,000[b]	18,500	....	23,000
KA2S, water-quenched	82,275	82,800	16,250	20,200	54,950	58,600	7,000	5,500	....	13,500	17,000[b]	18,000	....	23,000
Ni-Cr-Si, hot-rolled	....	128,900	....	45,000	....	79,100	....	35,500	....	10,000	....	19,000	....	53,000
Ni-Cr-Si, water-quenched	....	....	....	....	81,300	....	21,000	....	12,000	....	25,000	....	56,000	....

[a] U.M., University of Michigan; C.Co., Crane Company.
[b] Estimated from Fig. 4 (Clark and White).

exhibited by the hard-rolled material at both atmospheric and 1000 deg. fahr. This high elasticity in short-time tests is reflected by the creep stresses representative of high rates of deformation (1 per cent in 1000 hours). When minute creep is observed, however, the high-tensile KA2 is found to suffer a drastic reduction in its ability to withstand slight deformation.

The tensile properties of austenitic steels, in the various conditions and treatments to which they are responsive, apparently obey the same rules which have been found to apply to other types of steel. Fine grain size at the lower temperatures is conducive to high strength and elasticity. At extremely elevated temperatures large grain promotes creep resistance.

N. B. Pilling.[9] Manganese and nickel have a number of similarities in the way they behave in iron alloys, but they also have some points of dissimilarity, and it is not very safe to venture predictions regarding the qualities of one series based upon the nature of the other. The authors have pointed out a difference in strength properties at high temperatures between the chrome-manganese and chrome-nickel steels. While this is significant, the question is potentially one of so much importance to users and producers of metals for high-temperature service that it may be urged, without the slightest reflection upon the authors' research, as one deserving careful and independent verification.

While the paper is of course limited to a discussion of strength properties, it has doubtless come under the observation of the authors and would be interesting to know what comparison can be made between the chrome-manganese and chrome-nickel steels with respect to their corrosion by oxidation or other agencies at high temperatures.

[9] Research Laboratory, International Nickel Company, Inc., Bayonne, N. J.

Authors' Closure

The authors appreciate the comments of the people who have discussed this paper and are pleased to know that the results reported by them are in accord with those obtained by other investigators.

In regard to the comments of Professor Norton, the authors agree that austenitic steels possess the greatest creep resistance at the more elevated temperatures, but do not believe that such is necessarily the case at 1000 deg. fahr. They are aware of steels of the pearlitic type which possess, at 1000 deg. fahr., creep characteristics above the austenitic steels herein discussed or those of the chromium-nickel type.

The authors also agree that for each metal there is perhaps a temperature range in which the creep results, when plotted on logarithmic paper, fall on straight lines and that the average slope is 1 in 5. They feel, however, that at temperatures below this range, if the results fall on straight lines, the lines have decidedly different slopes.

They are also aware of many cases, in the lower temperature ranges, in which this type of plotting does not yield straight lines.

Concerning Mr. Pilling's remarks, the authors did not intend to convey the impression that the chromium-manganese steels were not worthy of additional study, and they are looking forward to the publication of additional data on these steels.

The authors are also in agreement with Mr. Kanter in that fine-grained materials possess the maximum creep resistance at temperatures below the "equicohesive" temperature range, while coarse-grained materials possess the maximum values at temperatures above.

They also wish to thank Mr. Kanter for the confirmatory data he has presented.

FSP-53-15

Properties of Non-Ferrous Alloys at Elevated Temperatures

By C. L. CLARK[1] AND A. E. WHITE,[2] ANN ARBOR, MICH.

This paper gives the results of an investigation of the properties of certain non-ferrous alloys at elevated temperatures as determined by short-time tensile and long-time creep tests. The alloys considered are of four groups: the nickel-copper, nickel-cobalt-titanium, copper-zinc, and copper-zinc-tin series. The alloys are tested at temperatures both above and below their lowest temperature of recrystallization.

Results indicate a superiority of the nickel-base alloys over the entire temperature range. The outstanding member of this group is Konel metal, which has short-time tensile properties at 1000 deg. fahr. comparable to Enduro KA2, and the greatest creep resistance at this temperature of any alloy of which the authors are aware. Monel metal retains its properties well up to 800 deg. fahr.

Of the copper-zinc alloys, those containing 70 per cent or more copper are superior to those with 60 per cent of this element. None of these alloys can be used at temperatures much above 400 deg. fahr., and for those containing lower amounts of copper, 300 deg. fahr. is believed to be the maximum working temperature if stresses of appreciable magnitude must be considered.

THE question of the behavior of metals when subjected to sustained loads at elevated temperatures is one that is receiving a great deal of attention at the present time. To date, however, a large share of the published work has been concerned with the behavior of ferrous alloys, or of non-ferrous alloys of the chromium-nickel type. This present paper records the results obtained from non-ferrous alloys of the copper-zinc, the copper-zinc-tin, the nickel-copper, and the nickel-cobalt-titanium series.



C. L. CLARK



A. E. WHITE

The work on these alloys was undertaken as it was felt a great need existed for this type of information. Many of these alloys find wide use at slightly elevated temperatures in such applications as valve fittings, while others are used in tubular form at even higher temperatures. Each will render satisfactory service at temperatures up to a certain maximum, provided the proper balance is maintained between temperature and stress. The information herein given is hoped to be of great value in properly defining these conditions.

MATERIALS

The alloys investigated belong to either the copper-zinc, the copper-zinc-tin, the nickel-copper, or the nickel-cobalt-titanium series. Their composition, the condition in which they were tested, and the source from which they were obtained are given in Table 1.

Ambrac metal differs chemically from Monel mainly in that the relative percentages of copper and nickel are interchanged and an appreciable amount of zinc added. The metal designated as K is the material known as Konel metal, which was developed at the Westinghouse Company and is claimed to have outstanding high-temperature properties. It differs from Monel metal in that the amount of nickel has been increased, the copper replaced with cobalt, and titanium added.

The amount of copper in the copper-zinc alloys varies from 85.0 to 69.5 per cent, while in the copper-zinc-tin alloys it ranges from 77.3 to 58.8 per cent.

The materials were received in $^3/_4$-in. round bars and were all in the hot-rolled condition except three, designated as C, A, and B in the table. In each of these cases the bars were given a final $^1/_8$-in. cold-draw.

PROCEDURE

The experimental work in this investigation consisted of short-time tensile tests at normal and elevated temperatures, determination of lowest temperature of recrystallization by means of hardness tests, and metallographic examination and long-time creep tests at selected elevated temperatures.

The method used for the determination of the short-time tensile properties was the same as that described in previous papers by the authors.[3,4] The specimen is maintained at the desired temperature by means of an electric furnace inserted in a screw type of tensile machine. The furnace is calibrated so that the temperature distribution over its entire length is known. The load is applied in increments up to the proportional limit, and the amount of deformation produced by each is determined by two Ames dials reading to one ten-thousandth (0.0001) of an inch.

The apparatus and procedure used in the long-time creep or flow tests was described at the 1930 Semi-Annual Meeting

---

[1] University of Michigan. Mr. Clark received the degrees of B.S. in Chemical Engineering in 1925, M.S. in 1926, and Ph.D. in 1928, from the University of Michigan. He has specialized in metallurgy, and is now research engineer in metallurgy in the Department of Engineering Research at the university.

[2] Director of the Department of Engineering Research and Professor of Metallurgical Engineering at the University of Michigan. Mem. A.S.M.E. Professor White received his B.A. from Brown University in 1907 and his Sc.D. in 1925. After graduation he served a few years with the Jones and Laughlin Steel Company, Pittsburgh, Pa. He has been at the University of Michigan since 1911, except for two years spent with the Ordnance Department, U. S. A., during the World War.

Contributed by the Joint Research Committee on Effect of Temperatures on the Properties of Metals and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[3] A. E. White and C. L. Clark, "Properties of Boiler Tubing at Elevated Temperatures Determined by Expansion Tests," Trans. A.S.M.E., vol. 48 (1926), p. 1075.

[4] A. E. White and C. L. Clark, "Stability of Metals at Elevated Temperatures," Trans. A.S.S.T., vol. xv (1929), p. 670.

TABLE 1 CHEMICAL COMPOSITION, MECHANICAL CONDITION, AND COMMERCIAL SOURCE OF NON-FERROUS ALLOYS INVESTIGATED

Designation	Types of material	Chemical composition								Condition	Source
		Cu	Zn	Sn	Ni	Fe	Mn	Co	Ti		
M	Monel	29.70	..	..	67.70	1.77	1.28	...	..	Hot rolled	Int. Nickel Co.
C	Ambrac	64.08	5.67	..	29.38	..	0.42	...	..	Final 1/8-in. cold drawn	Am. Brass Co.
K	Konel	...	(0.36 Al)	(0.54 Si)	73.02	6.47	0.15	17.20	2.38	Unknown	Westinghouse Mfg. Co.
A	Admiralty	71.05	27.95	0.97	...	..	..	..	..	Final 1/8-in. cold drawn	Am. Brass Co.
B	Brass	69.48	30.44	..	...	..	..	..	..	Final 1/8-in. cold drawn	Am. Brass Co.
E-1	Brass	60.21	39.72	0.16	...	..	..	..	..	0.020 grain size	Bridgeport Brass Co.
R-85	Brass	85.00	14.92	..	...	..	..	..	..	0.030 grain size	Bridgeport Brass Co.
E-2	Admiralty	77.26	21.61	1.18	...	..	..	..	..	0.020 grain size	Bridgeport Brass Co.
E-3	Tobin bronze	58.79	40.43	0.88	...	..	..	..	..	0.025 grain size	Bridgeport Brass Co.

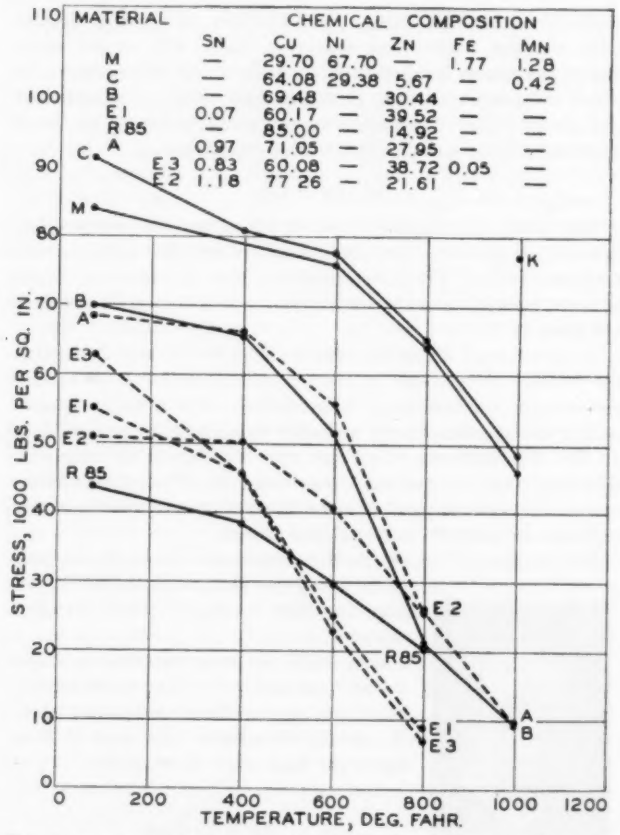


FIG. 1 TENSILE STRENGTH OF NON-FERROUS ALLOYS AT ELEVATED TEMPERATURES

TABLE 2 TEMPERATURE OF RECRYSTALLIZATION OF NON-FERROUS ALLOYS OF THE COPPER-ZINC, COPPER-ZINC-TIN, NICKEL-COPPER, AND NICKEL-COBALT SERIES

Designation	Type	Temp. of recrystallization, deg. fahr.	
		Hardness tests	Metallographic examination
R-85	85-15	Over 400, under 600	Over 400, under 500
E-2	77-22-1	Over 400, under 600	Over 400, under 500
E-1	60-40	Over 300, under 400	Over 300, under 400
E-3	59-40-1	Over 300, under 400	Over 300, under 400
B	70-30	Over 400, under 600	Over 400, under 500
A	71-28-1	Over 400, under 600	Over 400, under 500
M	Monel	Over 800, under 1000	.....
K	Konel	Over 1000, under 1200	.....

of The American Society of Mechanical Engineers in Detroit.[5] The apparatus is of the single-lever type, having a lever ratio of 24 to 1. A fixed load is applied and the resulting deformation measured by two mirrors fastened to rollers which are rotated by any change in length of the specimen. The sensitivity of the measuring system is such that readings accurate to 2.8 millionths (0.000028) of an inch are obtainable.

In performing this test a fixed load is applied and maintained constant until either the flow comes to a complete stop, at least in so far as it is measurably within the limits of sensitivity of the apparatus employed, or until it has assumed a steady rate for at least 200 hours. Then the load is either increased or decreased, depending on the magnitude of the deformation previously obtained.

For the determination of the lowest temperature of recrystallization, specimens of each alloy were severely cold worked, after which they were subjected to tempering at various temperatures. They were maintained at temperature five hours, hardness readings being taken before and after the heating operations. The temperature at which a sharp decrease in hardness was obtained was designated as the lowest temperature of recrystallization. The location of this recrystallization temperature was also checked by metallographic means.

RESULTS

Lowest Temperature of Recrystallization

The temperature of recrystallization of each of these alloys was determined by hardness tests and metallographic means. In this particular case an accurate determination of this temperature was not felt necessary and a recrystallization range rather than definite temperature was determined. The results are given in Table 2.

On the basis of the recrystallization temperature the alloys may be divided into three groups. In one group, this temperature lies above 800 deg. fahr., in another, between 400 and 600 deg. fahr., and in the third, below 400 deg. fahr.

The alloys containing large amounts of nickel fall in the first group. The Konel metal retained at temperatures as high as 1000 deg. fahr. all of the hardness given it through cold working, while the Monel metal did not undergo a loss in hardness at temperatures up to 800 deg. fahr. This indicates their respective temperatures of recrystallization to be over 1000 and 800 deg. fahr. Since these alloys differ mainly in that the copper in the Monel metal has been replaced by cobalt, it is evident that this replacement has resulted in the recrystallization range being raised 200 deg. fahr. Cobalt of course melts at a much higher temperature than does copper, and this may account for the increase.

The copper-zinc and copper-zinc-tin alloys containing 70 per cent or more of copper fall in the second group, that is, their lowest temperature of recrystallization lies over 400 and under 600 deg. fahr. The alloys of these two series containing 60 per cent copper recrystallize at temperatures below 400 deg. fahr. Thus they belong to the third group. The lowering of this tem-

TABLE 3 TENSILE STRENGTH AT INDICATED TEMPERATURES OF NON-FERROUS ALLOYS OF THE NICKEL-COBALT-TITANIUM, COPPER-NICKEL, COPPER-ZINC, AND COPPER-ZINC-TIN SERIES

Designation	Material			Temperature of test, deg. fahr.				
	Type	Composition	Condition	75	400	600	800	1000
K	Ni-Cu-Ti	73-17-2	Unknown	...	...	...	...	76,700
M	Ni-Cu	67-30	Hot rolled	84,000	...	76,000	64,000	46,000
C	Ni-Cu-Zn	29-64-5	1/8-in. cold drawn	91,500	80,950	77,500	65,250	48,500
R-85	Cu-Zn	85-15	Hot rolled	43,800	38,525	29,950	20,250	...
B	Cu-Zn	70-30	1/8-in. cold drawn	70,000	65,550	51,500	21,000	9,400
E-1	Cu-Zn	60-40	Hot rolled	55,350	45,750	25,150	8,700	...
E-2	Cu-Zn-Sn	77-22-1	Hot rolled	50,600	50,175	40,625	25,600	...
A	Cu-Zn-Sn	71-28-1	1/8-in. cold drawn	68,500	65,950	55,800	26,400	9,250
E-3	Cu-Zn-Sn	59-40-1	Hot rolled	62,750	45,875	23,050	6,950	...

[5] A. E. White, C. L. Clark, and L. Thomassen, "An Apparatus for the Determination of Creep at Elevated Temperatures," Preprint, A.S.M.E. Semi-Annual Meeting, Detroit, June 9–12, 1930.

perature with increasing amounts of zinc should be expected, as zinc itself possesses a much lower recrystallization temperature than does copper, being 70 deg. fahr. as compared to 392 deg. fahr.

Short-Time Tensile Tests

The short-time tensile properties of each of these alloys were determined at room and selected elevated temperatures. The results are given in Tables 3 to 6 and Figs. 1 to 4, inclusive.

Fig. 1 and Table 3 show the effect of composition, mechanical working, and temperature on the tensile strength of these alloys. Three of the alloys, designated as A, B, and C, were in the cold-worked condition and the remainder were hot-rolled.

Again the high-nickel alloys display the maximum properties throughout the entire temperature range investigated. Konel

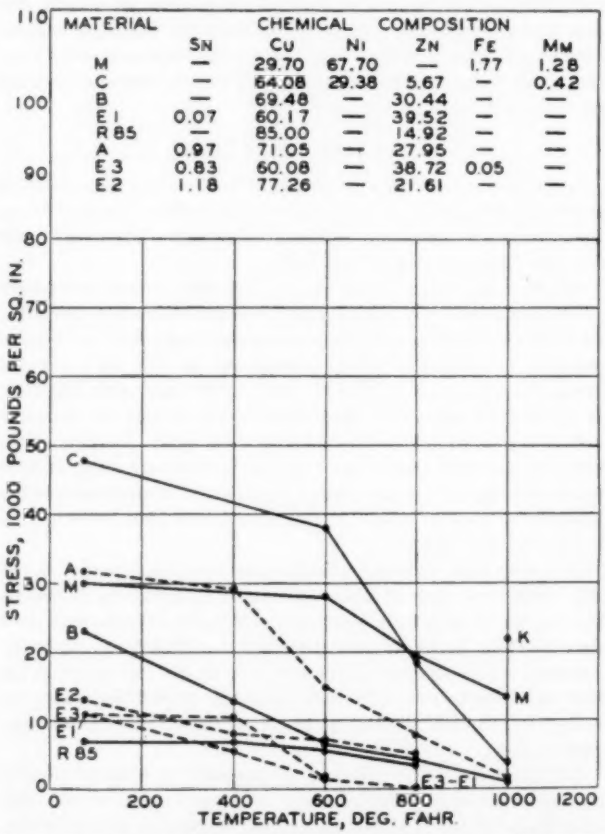


Fig. 2 Proportional Limit of Non-Ferrous Alloys at Elevated Temperatures

TABLE 4 PROPORTIONAL LIMIT AT INDICATED TEMPERATURES OF NON-FERROUS ALLOYS OF THE NICKEL-COBALT-TITANIUM, COPPER-NICKEL-COPPER-ZINC, AND COPPER-ZINC-TIN SERIES

Designation	Material: Type	Composition	Condition	Temperature of test, deg. fahr.: 75	400	600	800	1000
K	Ni-Cu-Ti	73-17-2	Unknown	...	...	...	...	22,000
M	Ni-Cu	67-30	Hot rolled	30,000	...	28,000	19,500	13,500
C	Ni-Cu-Zn	29-64-5	1/8-in. cold drawn	47,800	...	38,000	18,500	4,000
R-85	Cu-Zn	85-15	Hot rolled	7,000	7,000	5,500	3,500	...
B	Cu-Zn	70-30	1/8-in. cold drawn	23,000	13,000	6,750	4,000	1,000
E-1	Cu-Zn	60-40	Hot rolled	11,000	5,500	1,500	500	...
E-2	Cu-Zn-Sn	77-22-1	Hot rolled	13,000	8,000	7,000	5,000	...
A	Cu-Zn-Sn	71-28-1	1/8-in. cold drawn	31,750	29,000	15,000	8,000	1,500
E-3	Cu-Zn-Sn	59-40-1	Hot rolled	11,000	10,500	1,750	250	...

metal was tested only at 1000 deg. fahr., but at this temperature it was much superior to the remainder. Its strength is comparable to that of Enduro-KA2, a chromium-nickel steel widely used at elevated temperatures. The Ambrac alloy displayed

TABLE 5 PER CENT ELONGATION IN TWO INCHES AT INDICATED TEMPERATURES OF NON-FERROUS ALLOYS OF THE NICKEL-COBALT-TITANIUM, COPPER-NICKEL, COPPER-ZINC, AND COPPER-ZINC-TIN SERIES

Designation	Material: Type	Composition	Condition	Temperature of test, deg. fahr.: 75	400	600	800	1000
K	Ni-Cu-Ti	73-17-2	Unknown	...	...	...	...	42.5
M	Ni-Cu	67-30	Hot rolled	45.0	...	42.0	33.0	15.5
C	Ni-Cu-Zn	29-64-5	1/8-in. cold drawn	17.5	15.0	9.5	15.5	12.0
R-85	Cu-Zn	85-15	Hot rolled	47.0	35.5	...	4.2	...
B	Cu-Zn	70-30	1/8-in. cold drawn	24.3	12.0	4.5	26.0	24.5
E-1	Cu-Zn	60-40	Hot rolled	51.0	50.5	40.0	18.0	...
E-2	Cu-Zn-Sn	77-22-1	Hot rolled	71.0	60.5	54.5	23.0	...
A	Cu-Zn-Sn	71-28-1	1/8-in. cold drawn	35.0	24.0	14.5	57.5	33.0
E-3	Cu-Zn-Sn	59-40-1	Hot rolled	46.0	54.0	79.5	39.0	...

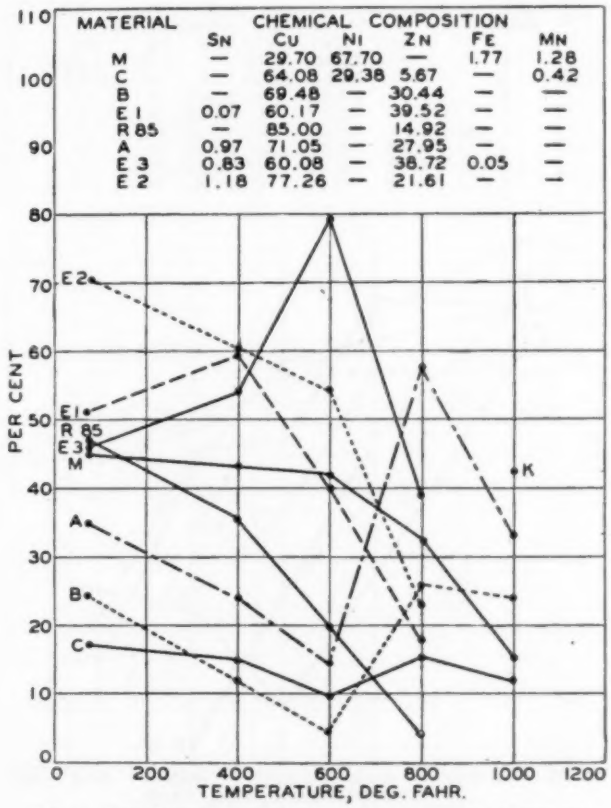


Fig. 3 Per Cent Elongation of Non-Ferrous Alloys at Elevated Temperatures

slightly higher values than did Monel, but it is to be remembered that the Ambrac was cold-rolled while the Monel metal was hot-rolled. Initial treatment greatly affects the short-time tensile properties, and if the Monel had been cold worked it would have displayed a much higher strength.

With the copper-zinc-base alloys, the two cold-worked materials possess the greatest strength. With increasing temperatures their superiority diminishes, and at the lowest temperature at which marked recrystallization occurs under the condition of the test, their strength is nearly identical to that of the other alloys containing 70 per cent or more of copper. The two alloys containing 60 per cent of copper lose their strength rapidly at temperatures above their lowest temperature of recrystallization, about 400 deg. fahr.

These figures emphasize the need of knowing the initial condition of the metal. From Fig. 1 it appears as if the 70-30 brass has a strength at the lower temperatures much superior to that of the 85-15 alloy. This is without doubt due to the lower-copper alloy being in the cold-worked and the 85-15 brass in the hot-rolled condition. If

TABLE 6 PER CENT REDUCTION OF AREA AT INDICATED TEMPERATURES OF NON-FERROUS ALLOYS OF THE NICKEL-COBALT-TITANIUM, COPPER-NICKEL, COPPER-ZINC, AND COPPER-ZINC-TIN SERIES

Designation	Material: Type	Composition	Condition	Temperature of test, deg. fahr. 75	400	600	800	1000
K	Ni-Cu-Ti	73-17-2	Unknown	..	..	..	..	44.8
M	Ni-Cu	67-30	Hot rolled	67.0	..	59.2	44.1	15.7
C	Ni-Cu-Zn	29-64-5	1/8-in. cold drawn	67.0	68.2	47.5	46.0	22.3
R-85	Cu-Zn	85-15	Hot rolled	69.0	48.2	..	3.4	..
B	Cu-Zn	70-30	1/8-in. cold drawn	61.5	28.5	9.0	23.0	26.0
E-1	Cu-Zn	60-40	Hot rolled	57.2	57.9	37.5	25.6	..
E-2	Cu-Zn-Sn	77-22-1	Hot rolled	76.4	65.7	54.6	35.0	..
A	Cu-Zn-Sn	71-28-1	1/8-in. cold drawn	64.0	40.0	25.5	56.5	39.4
E-3	Cu-Zn-Sn	59-40-1	Hot rolled	51.8	57.6	56.6	49.2	..

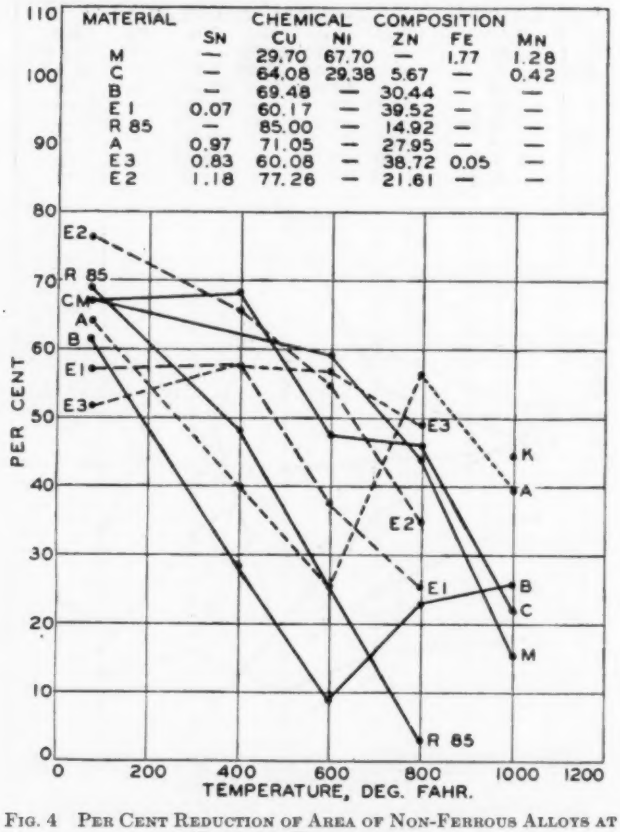


FIG. 4 PER CENT REDUCTION OF AREA OF NON-FERROUS ALLOYS AT ELEVATED TEMPERATURES

the two were placed in the same initial condition, it is probable that the 85-15 brass would have the greatest strength.

Fig. 2 and Table 4 show the variations in proportional limits under the same conditions. The alloys high in nickel have the maximum values at practically all temperatures. At 70 deg. and 400 deg. fahr. the cold-worked Admiralty metal has greater strength than the hot-rolled Monel, but this is because of the initial condition of the two materials. The Konel metal is much superior at the only temperature it was tested, 1000 deg. fahr., again having a value comparable to that of Enduro KA2. The cold-worked Ambrac has greater values at the lower temperatures than the Monel, but above 800 deg. fahr. the case is reversed. This is again due to the initial condition of the alloys.

The two cold-worked alloys of the copper-base series are superior to the remaining alloys at the lower temperatures, but at 800 deg. fahr. their strength is comparable to that of the materials containing 70 per cent or more of copper. Those containing 60 per cent of copper lose their strength rapidly at temperatures above 400 deg. fahr., which is approximately their lowest temperature of recrystallization.

Again the importance of knowing the initial condition of the metal should be emphasized, for while the 70-30 brass possessed values superior to those of the 85-15 alloy, this is believed to be due to their initial treatment and not to the composition.

The variation in the percentage of elongation values are given in Fig. 3 and Table 5, while those of the percentage of reduction of area are given in Fig. 4 and Table 6. These all indicate that the ductility of these alloys tends to decrease with increasing temperature. In the case of the cold-worked materials, however, this decrease is interrupted at the temperature at which recrystallization can occur, and a sharp increase in ductility is obtained. With increasing temperatures, the ductility of these cold-worked alloys again decreases.

The only exceptions to the above statements appear to be the materials containing 60 per cent of copper. The ductility of these alloys is increased with the first rise in temperature, the decrease not beginning until temperatures of 400 to 600 deg. fahr are attained. These alloys differ for the other copper-base alloys studied in that they are not homogeneous solid solutions, and the presence of the additional constituent may account for their difference in behavior.

LONG-TIME CREEP TESTS

Long-time creep tests were conducted on the majority of these non-ferrous alloys at temperatures both above and below their lowest temperature of recrystallization. The results are given in Figs. 5 through 8 and Table 7.

While a large number of time-elongation curves have been collected showing the behavior of these materials when subjected to continuous stresses for long periods of time, only one typical example is included. That, designated as Fig. 5, gives the creep characteristics of Monel metal at 600 and 800 deg. fahr. In obtaining these data, a fixed load is applied and the deformation of the specimen recorded at daily intervals. As soon as it is evident that flow has either come to a complete stop, at least within the limits of sensitivity of the apparatus employed, or has assumed a steady state for 200 hours, the load is either increased or decreased.

It is seen that in this particular case the Monel metal at 600 deg. fahr. was able to withstand loads as great as 22,000 lb. per sq. in. without undergoing continuous deformation with time. When the load was increased to 27,500 lb., however, continuous flow did take place but only at the rate of 0.028 per cent per 1000 hours. This same material at 800 deg. fahr. can withstand stresses at least as great as 15,850 lb. without continuous flow.

Attention should be called to the apparent downward slope of the line representing the behavior of Monel metal at 600 deg. fahr. under a load of 17,000 lb. Closer inspection will show that the line does not have a continuous slope, but is composed of two practically horizontal lines, with one displaced downward with respect to the other. This displacement is without doubt due to an external factor, such as a sudden jar on the apparatus, and should be considered as such.

When continuous flow is obtained, its magnitude is determined mathematically from the slope of the curves shown in Fig. 5. It may, of course, be expressed in any variety of unit desired, but the authors have found that of percentage per 1000 hours to be convenient.

Previous work has shown that, if the logarithm of the rate of creep be plotted against the logarithm of the stress producing that deformation, a straight-line relationship will be obtained under certain conditions. This has been done and the results are shown in Figs. 6 to 8, inclusive. It is seen that not only are straight lines obtained in nearly all cases, but also that a majority of the lines are nearly parallel to one another.

Charts of the above type are valuable, for they not only allow the relative merits of the various alloys to be compared but they also furnish a rapid means of calculating allowable working stresses, provided the amount of permissible deformation is known. Certain values of this type have been taken from the charts and are given in Table 7.

Table 7 briefly summarizes the results obtained from the creep tests, and places them in a form so they may be readily used. It also gives certain other information such as the short-time proportional limit at the various temperatures, and the recrystallization range.

From Table 7 it is seen that the Konel metal possesses the greatest creep resistance of any of the alloys. At 1000 deg. fahr. it was able to withstand a stress of 43,920 lb. per sq. in. without suffering continuous deformation. Although its short-time tensile properties were comparable to those of Enduro KA2, its creep resistance is vastly superior.

Of the remaining materials, the Monel metal is the superior and retains its creep resistance well at temperatures up to 800 deg. fahr. At 1000 deg. fahr. a sharp break occurs, a load of 1650 lb. being sufficient to produce continuous deformation at the rate of 0.01 per cent per 1000 hours.

The copper-base alloys possess good creep resistance at 400 deg. fahr., but have little load-carrying ability at 600 deg. fahr. The cold-worked brass and Admiralty metal have the greatest resistance of any alloys of these series at 400 deg. fahr., and their values are nearly equal. At 600 deg. fahr. the Admiralty is slightly superior to brass, but

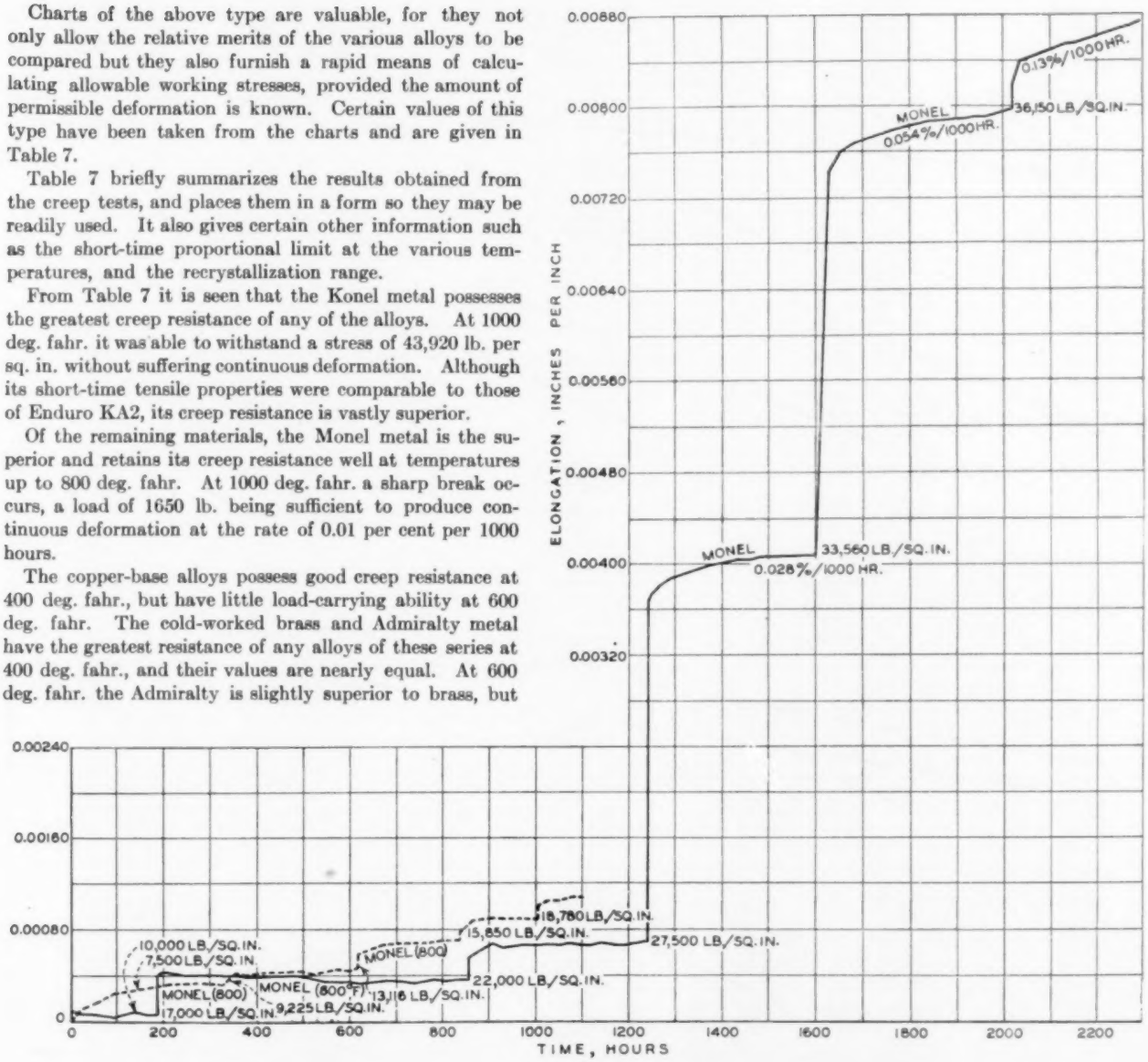


FIG. 5 CREEP CHARACTERISTICS OF MONEL METAL AT INDICATED TEMPERATURES

TABLE 7 SHORT-TIME TENSILE AND LONG-TIME CREEP CHARACTERISTICS OF NON-FERROUS ALLOY AT ELEVATED TEMPERATURES

Material	Temperature of recrystallization, deg. fahr.	Temperature, deg. fahr.	Prop. limit, lb. per sq. in.	Creep characteristics, per cent per 1000 hr. No flow	0.01	0.10	1.00	10.0
70-30	Over 400	400	13,000	10,000	12,700	18,000	27,000	40,000
	Under 600	600	6,750	...	...	850	...	...
		800	4,000	...	...	...	...	...
70-29-1	Over 400	400	21,000	10,000	13,000	19,000	27,000	40,000
	Under 600	600	15,000	...	...	1,950	3,800	7,200
		800	8,000	...	...	...	1,100	1,900
Monel	Over 800	600	28,000	22,000	26,000	36,000	46,000	...
	Under 1000	800	20,000	18,780	19,000	23,500	29,000	...
		1000	13,500	...	1,650	4,300	11,500	30,000
		1200	...	...	...	...	1,700	4,650
Konel	Over 1000 Under 1200	1000	22,000	43,920	...	...	...	...
85-15	Over 400 Under 600	400	7,000	7,500	8,800	12,000	17,000	...
77-22-1	Over 400 Under 500	400	8,000	...	10,500	13,000	16,500	...
60-40	Over 300 Under 400	400	8,000	...	...	...	...	...
59-40-1	Over 300 Under 400	400	16,000	...	3,500	5,700	9,400	..

neither has sufficient strength to allow its use commercially at this temperature. Both were tested at 800 deg. fahr. and while values are included for the Admiralty metal, the brass deformed so rapidly that the rate of creep was not constant but continuously increasing. Consequently they cannot be included in a chart of this type.

The 85-15 brass is inferior to the 70-30 alloy in its ability to resist creep at 400 deg. fahr., but this may be due to the initial condition of the two materials. The 77-22-1 alloy was much superior to the 59-40-1 material, but inferior to the 70-29-1 Admiralty metal.

The values in the table show the creep characteristics of the alloys to be dependent on their lowest temperatures of recrystallization. In every case at the temperatures below this critical one the alloy was able to withstand loads of considerable magnitude without continuous deformation, while, at temperatures above this critical point, continuous

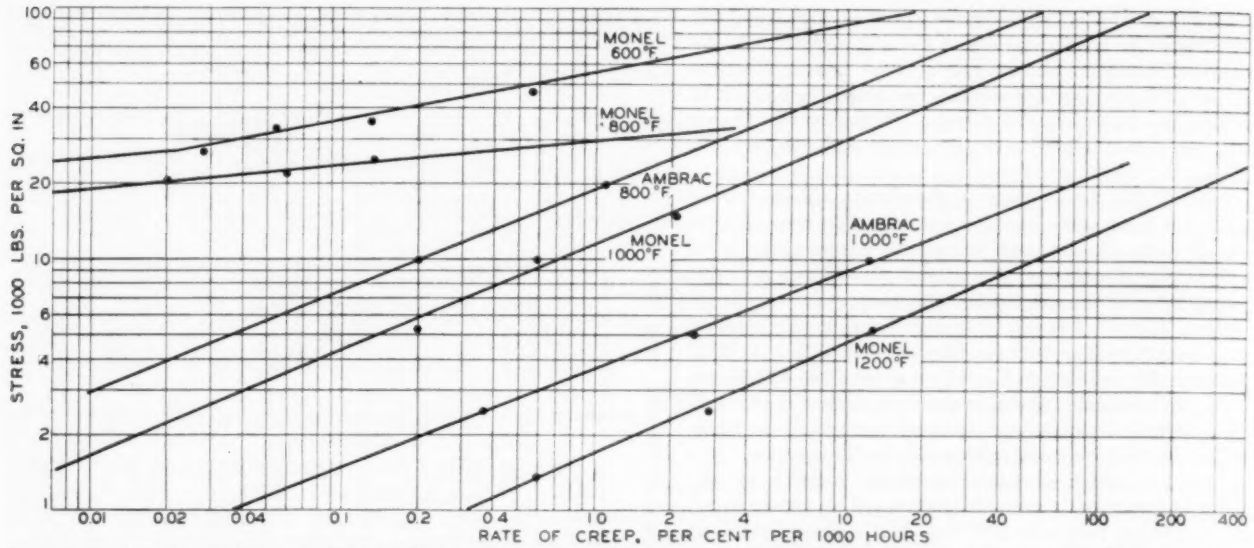


FIG. 6 LOGARITHM OF RATE OF CREEP VERSUS LOGARITHM OF STRESS FOR NICKEL-COPPER ALLOYS AT INDICATED TEMPERATURES

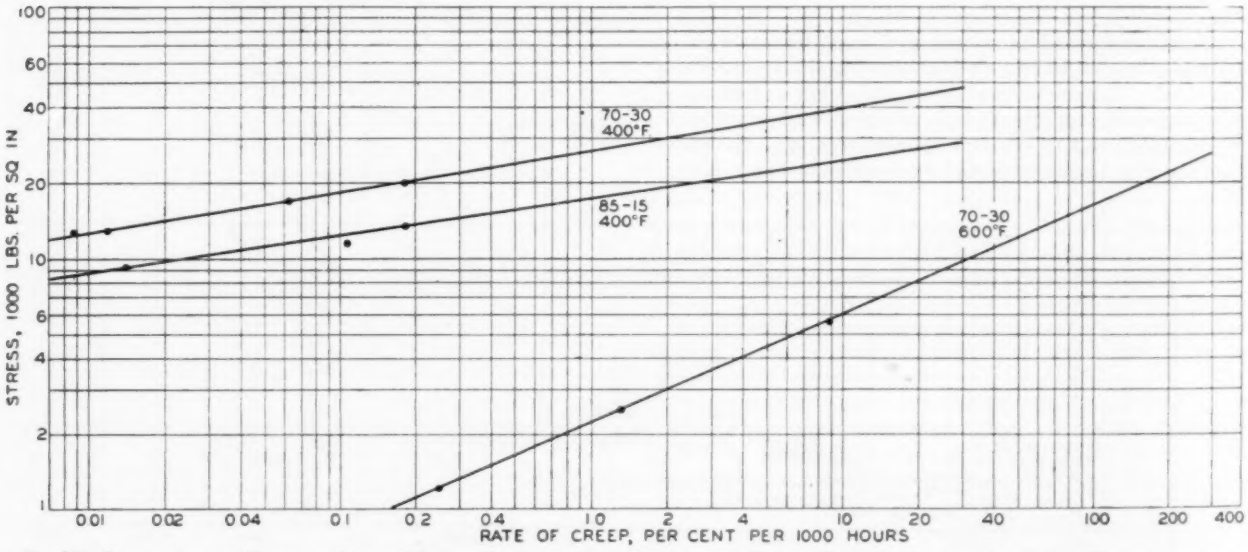


FIG. 7 LOGARITHM OF RATE OF CREEP VERSUS LOGARITHM OF STRESS FOR COPPER-ZINC ALLOYS AT INDICATED TEMPERATURES

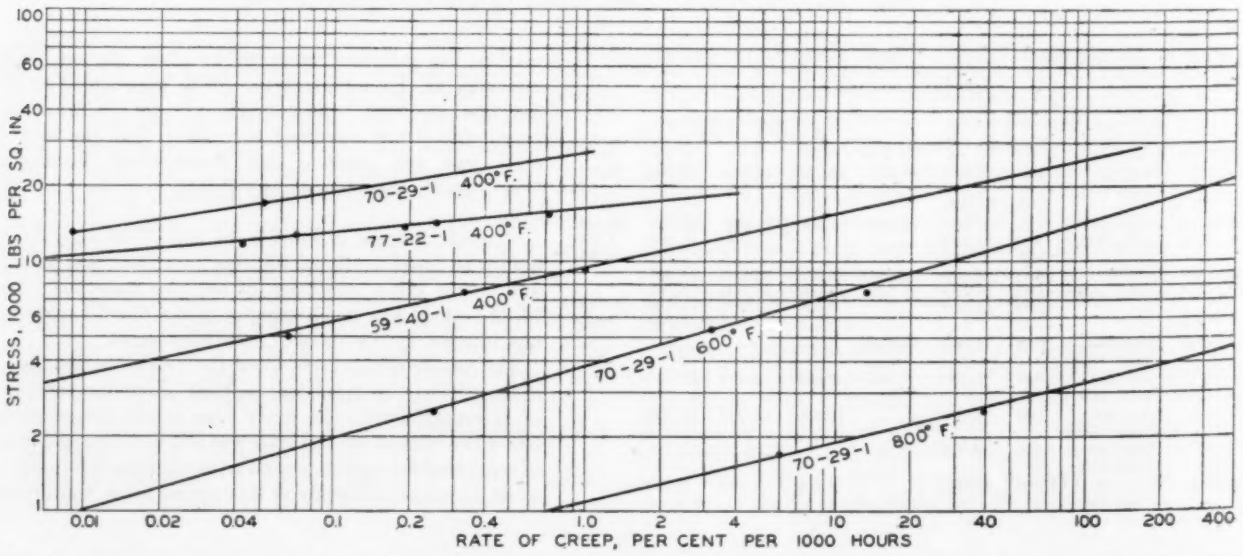


FIG. 8 LOGARITHM OF RATE OF CREEP VERSUS LOGARITHM OF STRESS FOR COPPER-ZINC-TIN ALLOYS AT INDICATED TEMPERATURES

creep was obtained with any load, no matter how small.

It is regretted that at the present time there are no results on similar alloys available in the literature so that a comparison can be made. More confidence is placed in results which may be checked by those from other sources. Until additional information does appear, these figures must be taken at face value.

CONCLUSIONS

The results obtained allow certain conclusions to be drawn regarding the effect of temperature, mechanical working, and chemical composition upon the short-time tensile and creep characteristics of certain non-ferrous alloys. The effect of chemical composition and the temperature of recrystallization was also determined.

In so far as the temperature of recrystallization is concerned, the alloy series arrange themselves in the order of decreasing temperature as nickel-cobalt-titanium, nickel-copper, and copper-zinc alloys. Of the copper-zinc alloys, those containing 70 per cent or more of copper have a higher range than those containing 60 per cent.

The short-time tensile tests show the high-nickel alloys to possess the maximum strength over the entire temperature range, with the Konel metal being the outstanding member of this group. With the remaining alloys, the cold-worked materials possess the greatest strength at the lower temperatures, but their superiority decreases with increasing temperature, and at temperatures at which marked recrystallization can occur, their properties are comparable to those of the hot-rolled alloys. The strength of the alloys containing 60 per cent of copper falls off at a more rapid rate with increasing temperature than do those containing 70 per cent or more of copper.

The ductility of each of the alloys tends to decrease with increasing temperature. In the case of the cold-worked materials, however, this decrease is interrupted at the temperature of recrystallization. At this point a sharp increase occurs, which is again followed by a decrease with rising temperature.

The creep results again show the high-nickel alloys to possess the maximum load-carrying ability at all temperatures and to be capable of use at higher temperatures than the copper-base alloys. The Konel metal possesses the greatest load-carrying ability at 1000 deg. fahr. of any material of which the authors are aware.

On the basis of these results the maximum safe working temperature for certain of the alloys when appreciable stresses must be withstood are believed to be as follows: Konel, at least 1000 deg. fahr.; Monel, 800 deg. fahr.; copper-zinc-base alloys with 70 per cent of copper, 400 deg. fahr.; copper-zinc alloys with 60 per cent of copper, 300 deg. fahr.

Concerning the relationship between the lowest temperature of recrystallization and creep characteristics, it is observed that at temperatures below the lowest temperature of recrystallization loads of appreciable magnitude do not produce continuous flow, while at temperatures above, continuous flow is obtained with any load regardless of its magnitude.

The authors wish to express their appreciation to The Detroit Edison Company for the financial aid given to this work, and to C. F. Hirshfeld, Chief of Research for The Detroit Edison Company, for his kindly interest and advice.

TABLE 8 TESTS OF MONEL METAL AND NICKEL

Source of tests	Material	Temperature of tests, deg. fahr. 75	400	600	800	1000	1200
Tensile Strength (Lb. per Sq. In.)							
Clark and White	Hot-rolled Monel	84,000	.....	76,000	64,000	46,000	....
Crane Company	Cold-drawn Monel	95,300	87,000	83,000	75,700	58,350	37,900
Crane Company	Cold-drawn Monel	120,800	107,000	100,000	86,250	59,900	34,400
Crane Company	Hot-rolled nickel	70,000	70,000	65,000	53,400	38,700	27,100
Proportional Limit (Lb. per Sq. In.)							
Clark and White	Hot-rolled Monel	30,000	.....	28,000	19,500	13,500	....
Crane Company	Cold-drawn Monel	47,500	57,000	55,000	44,800	....	....
Crane Company	Cold-drawn Monel	92,500	70,000	77,000	52,500	....	....
Crane Company	Hot-rolled nickel	24,000	28,000	26,500	17,500	14,000	6,000
Elongation (Per Cent in 2 In.)							
Clark and White	Hot-rolled Monel	45.0	..	42.0	33.0	15.5	..
Crane Company	Cold-drawn Monel	28.5	26.5	28.0	24.5	13.0	43.2
Crane Company	Cold-drawn Monel	19.5	17.5	22.0	21.0	3.5	23.5
Crane Company	Hot-rolled nickel	48.0	46.0	46.0	52.0	29.5	32.0
Reduction of Area (Per Cent)							
Clark and White	Hot-rolled Monel	67.0	..	59.2	44.1	15.7	..
Crane Company	Cold-drawn Monel	53.3	53.0	53.0	45.2	23.5	43.2
Crane Company	Cold-drawn Monel	57.4	57.5	52.0	47.8	4.6	35.1
Crane Company	Hot-rolled nickel	77.5	74.0	72.0	70.5	33.6	34.2

Discussion

J. J. KANTER.[6] The field of long-time tests on non-ferrous metals is one which as yet has been but little touched. Contributions in this direction fulfil an urgent need.

In regard to rolled Monel metal, it should be of interest to note that much higher strength and elasticity at elevated temperatures than the authors report can be obtained on cold-drawn material. If the object of using Monel metal under temperature is high strength, these results should recommend it. (See Table 8.)

The strength of cold-drawn Monel, nearly 60,000 lb. per sq. in. at 1000 deg. fahr., is second only to the Monel result reported by the authors. The reduced elongation and reduction of area which characterize these nickel alloys are shown by the Crane Company tests to occur also in pure nickel. This tendency toward brittleness, however, is greatly alleviated at 1200 deg. fahr., indicating that minimum points on the ductility-temperature curves occur in the vicinity of 1000 deg. fahr.

An important aspect of tensile properties of wrought brasses and bronzes is their change during fabrication. When hot-rolled and extruded stock is forged or hot-stamped, it invariably is found to have undergone drastic softening. The difference in hardness between the "as received" stock and the formed parts corresponds approximately to the loss sustained through a dead anneal. In order to properly evaluate high-temperature tensile properties representative of hot-worked articles, it is imperative that the test specimen be prepared from either hot-forged or annealed stock. A number of non-ferrous alloys have been tested under temperature in both the "received" and annealed conditions at Crane Company laboratories. These tests show that annealed materials are inferior to the hot-rolled and extruded in proportional limit at temperatures up to 500 deg. fahr., to the order of 50 to 75 per cent. Tensile-strength values likewise undergo an appreciable reduction of from 15 to 30 per cent.

The proportional limits of dead-annealed brasses, while small in value at temperatures from 75 to 400 deg. fahr., persist in measurable values at temperatures above those for which this constant has practically vanished in the case of hot-rolled and extruded material. At 600 deg. fahr., a temperature for which the authors report 1500 and 1750 lb. per sq. in. proportional limit for hot-rolled yellow brass, it can be shown that the same material, when annealed at 1500 deg. fahr., retains a proportional limit of 7000 to 12,000 lb. per sq. in. Creep tests on the annealed brass show a rapid decline of load-carrying ability in the vicinity of 400 deg. fahr., as has been reported by the authors for hot-rolled brasses. The red bronzes (not covered in the Clark and White report) are found

[6] Crane Company, Chicago, Ill.

in Crane Company tests to resist creep up to about 600 deg. fahr., a margin of 200 deg. fahr. superiority to yellow brass.

It is of interest to note the types of fracture which are obtained on high-temperature tensile-test specimens, particularly nonferrous metals and alloys. Normally, at atmospheric temperature, wrought brasses and bronzes rupture with cup-like fractures of a silky contour. Fractures of this kind persist up 300 or 400 deg. fahr., where they commence to take on a more irregular or heterogeneous appearance. At 500 deg. fahr. and above, the breaks are characterized by jagged contours and complete disappearance of cup. In the case of Monel metal and other nickel-base alloys, similar transitions occur between 800 and 1000 deg. fahr. For carbon and low-alloy steels a similar change is usually found between 800 and 900 deg. fahr. Highly tempered alloy steels that give a star-like or radially cracked fracture when cold, usually pass through a temperature range giving cup fractures before reaching irregular rupture. Austenitic irons and steels do not completely transform from silky to irregular rupture until they reach 1200 deg. fahr.

These transformations in fracture type are usually interpreted as being due to change from rupture through the grains to rupture by separation of grains along their boundaries. This temperature or range of temperature has come to be called the "equicohesive" on the assumption that it represents a state in which the grain boundary strength is equivalent to the crystalline cohesive force. Long-time test results indicate that it is in this critical range that metals start to creep at small loads. The temperature of recrystallization determined by the authors for the various alloys they considered are all well within the range where the fracture type usually changes. May it be suggested that the value of Table 2 would be greatly enhanced if it were correlated with descriptions of the fractures obtained in the tensile investigations.

W. B. Price.[7] In support of the authors' statement that "Results indicate a superiority of the nickel-base alloys over the entire temperature range," the writer would like to record the results of some short-time tensile tests carried out some six years ago on Adnic. Adnic is a copper-nickel-tin alloy of the following composition: copper, 70 per cent; nickel, 29 per cent; and tin, 1 per cent. Its recrystallization temperature would be over 1100 deg. fahr. and under 1200 deg. fahr. Tests conducted on $^{3}/_{16}$-in. cold-drawn material gave the following results:

Temperature of tests, deg. fahr.	Tensile strength, lb. per sq. in.
70	94,000
387	82,900
557	81,000
769	70,600
957	59,300

If these results are plotted on Fig. 1 of the paper, it will be noted that the curve is above the curve for Ambrac and Monel metal.

The proportional-limit results over the same temperature range were as follows:

Temperature of tests, deg. fahr.	Tensile strength, lb. per sq. in.
70	60,000
387	58,500
557	60,000
769	46,300
957	31,300

These results plotted on Fig. 2 of the paper show them to be very much superior to any of the other alloys, even Konel.

In the November, 1927, issue of *Mining and Metallurgy* there was an article by the writer on "The Physical Properties of Adnic." In that paper, in Fig. 1, a comparison of the physical properties of Monel metal and Admiralty nickel (Adnic) are shown in which the tests were carried up to 1500 deg. which is 500 deg. higher than the temperature used by the authors. Note the high proportional limit and tensile-strength values at elevated temperatures exhibited by Adnic when compared with Monel.

P. G. McVetty.[8] It is indeed interesting to see the results of independent tests of one of the Konel alloys which have been developed in the Westinghouse Research Laboratories. The first knowledge that the authors were investigating this material came when the paper was received.

In Tables 2, 4, 5, and 6, the condition of the test specimens is given as "unknown." The company is not acquainted with the history of this particular bar, but has obtained similar test results after heating at 900 deg. cent. (1652 deg. fahr.) for two hours and cooling in the furnace.

To supplement the data given in the paper, Table 9 gives physical properties at 24, 400, 500, and 600 deg. cent., corresponding, respectively, to 75, 752, 932, and 1112 deg. fahr. These values were all obtained after annealing two hours at 900 deg. cent. (1652 deg. fahr.).

TABLE 9 PHYSICAL PROPERTIES OF KONEL

Density, 8.59
Average coefficient of expansion: 0–100 deg. cent. 10.75×10^{-6}; 0–700 deg. cent. 14.30×10^{-6}

Temperature, deg. cent.	24	400	500	600
Proportional limit, lb. per sq. in.	32,500	37,000	24,000	23,000
	30,000	30,000	....	....
Johnson's limit, lb. per sq. in.	42,500	44,700	29,500	31,500
	35,000	33,300	....	....
Yield point (0.2 % plastic strain), lb. per sq. in.	55,500	55,400	39,800	41,600
	46,500	42,750	....	....
Yield point (0.5% total strain), lb. per sq. in.	57,000	56,900	41,000	43,200
	48,500	44,500	....	....
Ultimate strength, lb. per sq. in.	103,450	104,000	82,500	66,000
	96,900	93,750	....	....
Young's modulus, millions of lb. per sq. in.	29.4	26.3	25.9	24.3
	29.6	28.5	....	....
Elongation in 3 in., per cent	33.3	29.0	18.3	15.6
	35.0	33.0	..	..
Elongation in 2 in., per cent	37.0	34.0	33.0	18.5
	40.0	36.5	..	..
Reduction of area, per cent	59.8	51.6	44.5	25.5
	51.6	45.4	..	..

The analysis referred to in the paper is one of a series of alloys. By variations in the constituents, it has been found possible to improve the alloy to meet various service conditions. A comprehensive investigation of the system has been in process for about three years, and the writer expects to present a paper on the subject before the A.I.M.E. in February.

Arthur J. Herschmann.[9] This is an exceedngly interesting paper which has given much information and many ideas. In view of the statement that one of the main objectives of these investigations is the application of results to benefit central-station practice and construction for highest pressure in conjunction with highest temperature, it may be pertinent to tell of one instance of highest pressure and temperature where carbon steel has thus far been satisfactory.

The Loeffler boiler, which the writer represents, operates on 1900 lb. and 950 deg. total temperature. Drums and tubes have been in constant operation on slightly lower pressure and temperature for over three years, and no creep or change of

[7] Chief Chemist and Metallurgist, Scovill Manufacturing Company, Waterbury, Conn.

[8] Mechanical Engineer, Research Engineering Department, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

[9] Agent for U. S. A. for Vitrovice Steel Works (Czechoslovakia), New York, N. Y. Mem. A.S.M.E.

physical properties has been observed with testing equipment at disposal. It may be stated that these high-pressure drums, which are now about 44 in. diameter each and about 26 ft. long, are out of the fire. They are $2^{5}/_{8}$ in. thick. The superheater tubes are of 45 carbon steel, there being no rolled joints; everything is welded. There is now a slight addition of molybdenum, but only because it is relatively cheap and its increased cost is offset by the obvious advantages of consequently thinner tubes. The reason why this company can use 950 deg. total temperature is that its system precludes an accidental rise of such total temperature in the superheaters during operation, owing to the pump of the Loeffler system which invariably cools these tubes with ample steam. Loeffler boilers are built for 50 and 75 tons of steam per hour, on steel as stated.

The 2500-hp. Loeffler locomotive of the German Reichsbahn has a high-pressure drum of nickel-chromium steel owing to the desirability of obtaining the lowest possible weight.

Answering the statement that ordinary carbon steel could not very well serve a much higher combination of pressure and temperature than the foregoing, the writer would repeat that the circulating pump of the Loeffler design enables the working in these limits with considerable safety. Professor Loeffler could see no advantage in going above such pressure and temperature.

Authors' Closure

The authors appreciate the discussion which has been offered to this paper and are pleased to find their work in agreement with that of other investigators. They wish to thank Mr. Kanter for his comments and additional data showing the affect of heat treatment and methods of fabrication on the resulting physical properties of these non-ferrous alloys. They are well aware that the initial condition of these alloys greatly affects their properties, but time has not as yet allowed the long-time creep tests to be conducted on each of these alloys in all possible conditions of fabrication and heat treatment.

In regard to Mr. Price's comment, it should be understood, of course, that when the authors made the statement, "Results indicate a superiority of the nickel-base alloys over the entire temperature range," they were referring only to the alloys considered in this paper.

They have not as yet conducted any tests on Adnic, but are hoping to be able to investigate this interesting alloy at a later date.

The authors are also gratified for Mr. McVetty's comments and additional data concerning Konel metal and to Mr. Herschmann for his comments as to operating temperatures and pressures in the Loeffler boiler.

FSP-53-16

Fractures in Boiler Metal

By A. E. WHITE[1] and R. SCHNEIDEWIND,[2] ANN ARBOR, MICH.

During the past several years many specimens of failed boiler materials have been brought to the authors' attention with questions as to the mechanism and probable cause of the failures. In some cases the pieces consisted of fractured portions of tubes, blow-off pads, saddles, plates, rivets, and similar boiler parts. The examinations, which were to a large extent metallographic in nature, have led to some generalizations and classification of cracks and fractures as related to the causes of failure. This paper classifies the types of failure and gives their metallographic characteristics. The material presented consists of a short review of the literature on boiler-metal failures, a detailed description of fractures in boiler metal due to known causes, and a description of actual failures in service.

FROM a study of the literature the causes for failures in boiler materials may be divided roughly under three general headings: defects due to the composition of steel or to steel-mill practice, defects due to methods of boiler construction, and defects developing as a result of conditions of operation of the equipment. These divisions are not entirely clear-cut due to the close interrelationship of these factors.

High sulphur and especially high phosphorus contents in steel have been mentioned as contributing causes of brittleness by Fowler[3] and by Law.[4] Houghton[5] believed that the combined contents of phosphorus and arsenic determined to a large extent the satisfactoriness of boiler steel.

Segregation of constituents[6,7,8] and other non-homogeneities such as are revealed by macroscopic etching, a Rosenhain etch,[9] or by sulphur prints[10] are commonly found near fractures. In many cases, however, segregations do not constitute a serious defect.

Pipes, blow holes, seams, and inclusions constitute defects of a serious nature. In order to improve the quality of boiler plate, Houghton[5] suggested that not more than 45 per cent of the ingot be used for this purpose. It is general practice, however, according to McKnight and Humpton,[11] with rimming carbon steel to utilize about 70 per cent for boiler plate, and with nickel steels about 60 per cent. In order to improve the physical properties much of the boiler steel is being supplied in the normalized condition.

Many failures have been traced to serious strains induced by careless riveting[12,13,14,15] in the process of constructing the boiler. Sharply bent flanges[16] if not properly heat-treated are quite apt to suffer from aging and become brittle or they are liable to fail by corrosion. Boiler-making methods have been improved greatly by drilling instead of punching the rivet holes, by more careful calking, and in general by improving the fit of the various parts of the boiler so that undue stresses are not set up in the seams.[17] Many of the defects enumerated thus far can be detected by good inspection methods.

A large share of failures occur from defects which although they may have their inception in the streases or flaws present when the boiler went into service do not, however, make their appearance until after the boiler has been in service for some time. Cracking sometimes occurs in stressed parts due to a phenomenon known as blue brittleness. If a piece of low carbon steel is deformed and heated to from 300 to 450 deg. cent., the material becomes brittle, and the fractured steel has a bluish tinge. Nickel and especially chromium raise the temperature at which such embrittling takes place. Brittleness[18] is also developed on aging. It was observed that some boiler plate which was known to be ductile when the boiler was built became markedly less ductile after several years' use.

Age-hardening after cold drawing is due to the carbon according to Pfeil.[19] Carbon-free iron does not age, but as little as 0.0025

---

[1] Professor of Metallurgy, Director, Department of Engineering Research, University of Michigan. Has been a member of the faculty of the U. of M. since 1911. Previous to that he was with the Jones and Laughlin Steel Co. in charge of certain phases of research work. He is the author of many papers on metallurgy.

[2] Associate Investigator in the Department of Engineering Research, University of Michigan. Born December 16, 1900. Attended Detroit Junior College, the University of Chicago, and received degree of Bachelor of Science, Chemical Engineering, at the University of Michigan in 1923; received degree of M.S.E. at same institution in 1928. Was associated with the Studebaker Corporation for three years as analytical and plant chemist. Associated with the University of Michigan continuously since 1924. Member of honorary societies Sigma Xi and Iota Alpha and of technical societies American Society for Steel Treating and American Electrochemical Society. Published one paper on metal cleaning and six on chromium plating.

[3] H. Fowler, "Notes on Fractures in Locomotive Boiler Tubes," Trans. Faraday Soc., 1921, pp. 82–90.

[4] E. F. Law, "Brittleness and Blisters in Thin Steel Sheet," Jl. Iron and Steel Institute, vol. 69, pp. 134–144, 1906.

[5] S. A. Houghton, "Failures of Heavy Boiler Shell Plates," Jl. Iron and Steel Institute, vol. 89, pp. 266–292, 1914.

[6] R. S. Williams and V. O. Homerberg, "The Macroscopic Examination of Steel," Trans A.S.S.T., vol. 6, pp. 295–314, 1924.

[7] W. Rosenhain and D. Hanson, "The Failure of Boiler Plates in Service and Investigations of the Stresses That Occur in Riveted Joints," Jl. Iron and Steel Institute, vol. 97, pp. 313–324, 1918.

[8] Anon., "Explosion of an Autogenously Welded Boiler," *Genie Civil*, vol. 90, pp. 245–246.

Contributed by the Joint Research Committee on Boiler Feedwater Studies and presented at the Semi-Annual Meeting, Detroit, Mich., June 9 to 12, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[9] W. Rosenhain and J. L. Houghton, "A New Reagent for Etching Mild Steel," Jl. Iron and Steel Institute, vol. 89, pp. 515–524, 1914.

[10] F. Rogers, "Note on the Investigation of Fractures," Jl. Iron and Steel Institute, vol. 85, pp. 379–386, 1912.

[11] C. McKnight and W. G. Humpton, "The Manufacture of Nickel Steel Plate," International Nickel Co., Bulletin No. 14, 1929.

[12] E. Schulz, "Boiler Sheet Failure," *Elektro-Journal*, vol. 5, pp. 1–10, 1925.

[13] P. Ludwik, "Elastic Limits, Blue Brittleness, and Caustic Embrittlement," Zeit. Verein deut. Ing., vol. 70, 385–386, 1926.

[14] F. Müller, "Explosion of a Cellulose Cooker," Zeit. Bayerischen Revisionsvereins, vol. 31, pp. 195–197, 209–213, 1927.

[15] Anon., "Embrittlement of Boiler Plates by Chemical Agency," *Iron and Coal Trades Review*, vol. 117, p. 367, 1928.

[16] W. Quack, "Boiler Explosion Due to Fatigue Fracture," Wärme, vol. 48, pp. 369–372, 1925.

[17] S. W. Parr and F. G. Straub, "Embrittlement of Boiler Plate," University of Illinois Experiment Station, Bulletin No. 177, 1928.

[18] C. E. Stromeyer, "The Aging of Mild Steel," Jl. Iron and Steel Institute, vol. 73, pp. 200–221, 1907, also "Further Experiments on the Aging of Mild Steel," Jl. Iron and Steel Institute, vol. 75, pp. 86–104, 1908, also "The Aging of Mild Steel and the Influence of Nitrogen," Jl. Iron and Steel Institute, vol. 79, pp. 404–420, 1909.

[19] L. B. Pfeil, "The Change in Tensile Strength Due to Aging of Cold-Drawn Iron and Steel," Jl. Iron and Steel Institute, vol. 118, pp. 167–181, 1928.

per cent of carbon will permit aging. Iron oxide acts similarly.

Cracking has been ascribed by some writers to pure physical stresses.[3,20,21] Wolff[22] describes some transcrystalline cracks and suggests that fatigue is an important factor contributing to the failure. Similarly Rosenhain and Hanson[23,24] believed that cracking might occur in mild carbon steels through prolonged tensile stressing. However, upon loading steel containing 0.10 per cent carbon with stresses up to $^2/_3$ of the normal ultimate strength for over five years and maintaining a temperature of 300 deg. cent., no fracture took place.

Cracking in boiler metals has also been considered a corrosion phenomenon. Körber and Pomp[25] demonstrated by means of etching agents containing copper that locally strained areas were present and that corrosion and cracking took place along these lines or directions of high stress.

Cracking due to brittleness and strain is in general through the grain. A large number of boiler cracks are intercrystalline and start at points of high localized stresses. This phenomenon has been termed by some caustic embrittlement. Parr and Straub[17,26] in a series of carefully controlled tests showed that intercrystalline cracking such as is often found in boiler plate could be reproduced in the laboratory at operating boiler temperatures by maintaining a piece of steel in tension under a stress near the elastic limit in contact with a solution containing no less than 100 grams per liter of sodium hydroxide. They point out that high stresses may occur in the vicinity of riveted joints, and that in small cracks or fissures acting as capillaries a high concentration of salts can be formed. Careful control of the boiler-water composition and the addition of sodium sulphate will materially decrease chances of embrittlement. Suggestions of other possible mechanisms of intercrystalline cracking have been made, notably that of corrosion-fatigue by MacAdam.[27] Pfeil[28] has showed that occluded hydrogen always weakens the grain boundaries so that intercrystalline cracks are formed. In many cases intercrystalline cracks have developed in the presence of stress and a corroding medium. For example, low carbon-steel pans used in evaporating sodium-nitrate solutions cracked where the metal was bent sharply. Solutions of other nitrates and alkalies acted in the same manner on stressed steels.

The water in the boilers if not properly controlled[29] as to its pH, oxygen, and carbon-dioxide content may cause corrosion and pitting of the steel. Woodvine and Roberts[30] showed that segregation of constituents in the metal also affected the pitting and corrosion in boiler tubes. If the process of water softening is not adequate, scale will form in the tubes and permit overheating which may lead to bulges and even ruptures of the tubes.

Characteristic Fractures Due to Known Causes

In order to present more or less ideal type failures, boiler-tube material was caused to fail by bending, compression, tension, impact, fatigue, and by caustic embrittlement. The characteristics of each type of failure were noted and were later compared with those of failures actually met with in boiler materials.

The material used in this study was ordinary boiler tubing having the following analysis:

Carbon	0.14 per cent
Manganese	0.48 per cent
Sulphur	0.031 per cent
Phosphorus	0.010 per cent
Silicon	0.14 per cent

It was annealed at 1800 deg. fahr. for four hours and cooled slowly in the furnace in order to produce equiaxed, unstrained crystals of large size. Photomicrographs in Figs. 1 and 2 show the structure of the metal at 100 and 1000 diameters, respectively. At the lower magnification a tendency can be seen for the carbides to be distributed along lines in the direction of working. In general the material is fairly clean, homogeneous, with distinct, straight grain boundaries.

The failures produced in the manners described were in some cases accompanied by deformation of the crystals and in others the crystals were not deformed. The results of the tests are therefore taken up according to that classification.

Failures Accompanied by Deformation of the Crystals

Slip Lines. Upon subjecting a polished and etched section of this steel to a stress by squeezing in a vise, lines corresponding to the crystallographic planes of slip are formed. Typical slip lines are shown in photomicrographs in Figs. 3 and 4 at 100 and 1000 diameters, respectively. Lines of slip are shadows due to differences in level caused by the sliding of different portions of a crystal along gliding planes. In the case of α-iron, this represents the trace of the [111] plane. The stress must be, locally at least, beyond the elastic limit of that crystallographic constituent so that failure or disorganization of a series of unit cells takes place. After repolishing and re-etching the specimen it will be found that the slip lines (or slip bands) have disappeared. This is shown in photomicrographs in Figs. 5 and 6. Repolishing effaces the differences in level and thus eliminates the evidence of slip.

X Lines. A strip of this steel was bent through an angle of 180 deg. and then bent flat again. Photomicrograph in Fig. 7 presents a longitudinal section at 100 diameters through the axis of bending. Bending took place along an axis perpendicular to the edge shown in photomicrograph Fig. 7 and roughly through the middle of the picture. A number of lines can be seen existing in some of the crystals. Since polishing and etching do not completely efface these lines, they must be considered a severer type of slip line or, to use the nomenclature of Howe,[31] they are X lines or X bands. They are found in this steel only in the ferrite; a typical system is shown in photomicrograph Fig. 8 at 1000 diameters. It is to be noted that X lines have formed only in a portion of the crystals, undoubtedly in those crystals only whose orientation favored deformation by stresses applied in the direction in which the force was applied in this case. It is to be noted further that all of the X lines make angles of 45 deg. or less with the axis of bending. It can be inferred that the orientation of the undeformed crystals was such that they resisted deformation sufficiently so that no traces of X lines resulted.

[20] B. Straus and A. Fry, "Crack Formation in Boiler Plates," *Stahl u. Eisen*, vol. 41, pp. 1133–1137, 1921.

[21] G. Ness and D. A. MacCallum, "Fractured Boiler Plates," Jl. West Scotland Iron and Steel Institute, vol. 35, pp. 101–109, 1928.

[22] E. B. Wolff, "The Failure of Boiler Plates in Service and Investigations of the Stresses That Occur in Riveted Joints," *The Engineer*, vol. 124, p. 456, 1917.

[23] W. Rosenhain and D. Hanson, "Intercrystalline Fracture in Mild Steel," Jl. Iron and Steel Institute, vol. 101, pp. 2–24, 1920.

[24] W. Rosenhain and D. Hanson, "The Behavior of Mild Steel Under Prolonged Stress at 300 Deg. Cent.," Jl. Iron and Steel Institute, vol. 116, pp. 117–125, 1927.

[25] F. Körber and A. Pomp, "Cracks and Corrosion in Boiler Parts," Zeit. Bayerischen Revisionsvereins, vol. 30, pp. 279–285, 295–301, 1926.

[26] S. W. Parr and F. G. Straub, "The Cause and Prevention of Embrittlement of Boiler Plate," University of Illinois Experiment Station Bulletin No. 155, 1926.

[27] D. J. MacAdam, Proc. A.S.T.M., vol. 26, p. 88, 1926, discussion; Proc. A.S.T.M., vol. 27, p. 70, 1927, discussion.

[28] L. B. Pfeil, "The Effect of Occluded Hydrogen," Proc. Royal Soc., London, vol. 112, p. 182, 1926.

[29] J. Jicinsky, "Corrosion and Cracking of Boiler Plates and Tubes," *Glückauf*, vol. 63, pp. 1373–1380, 1927.

[30] G. R. Woodvine and A. L. Roberts, "The Influence of Segregation on Corrosion of Boiler Tubes and Superheaters," Jl. Iron and Steel Institute, 1926, vol. 113.

[31] H. M. Howe, "The Metallography of Steel and Cast Iron," p. 452, edition of 1916, McGraw-Hill Book Co.

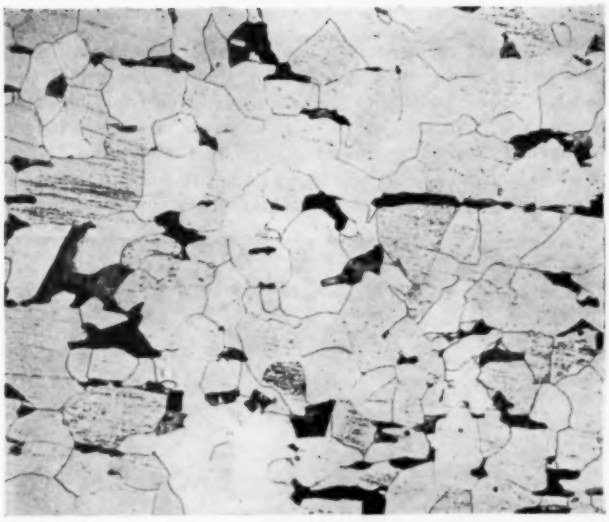


FIG. 1 TYPICAL SECTION OF BOILER-TUBE STEEL USED IN THIS INVESTIGATION AT 100 DIAMETERS ETCHED IN 4 PER CENT NITAL

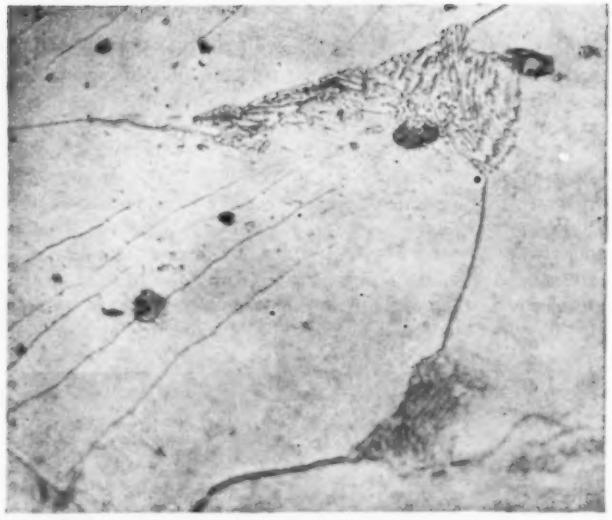


FIG. 4 SAME AS FIG. 3 BUT AT 1000 DIAMETERS

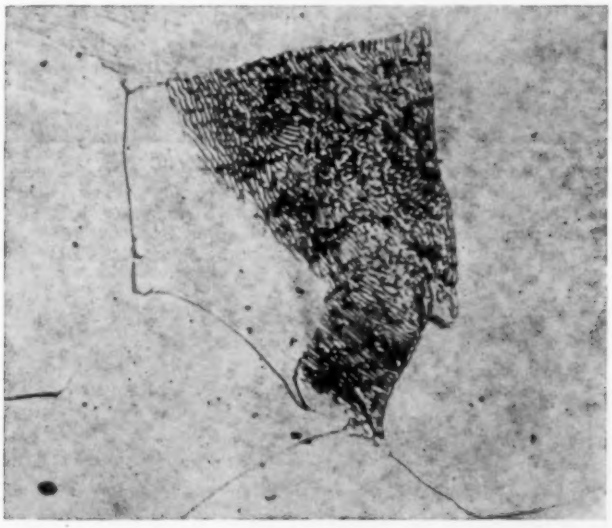


FIG. 2 SAME AS FIG. 1 BUT AT 1000 DIAMETERS

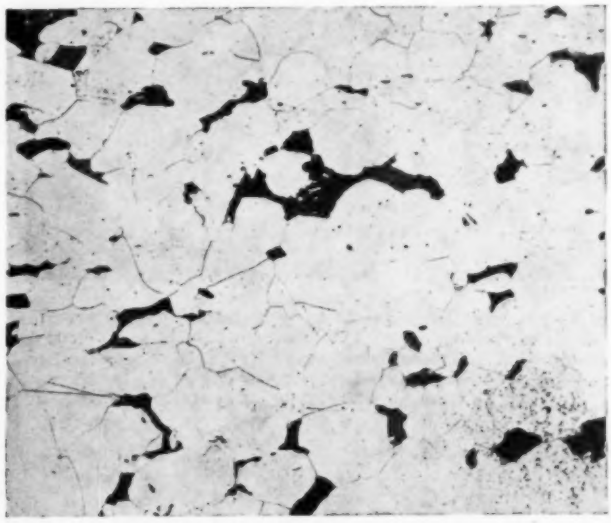


FIG. 5 THE SLIP LINES IN FIG. 3 HAVE BEEN EFFACED BY RE-POLISHING AND RE-ETCHING—PHOTOMICROGRAPH AT 100 DIAMETERS

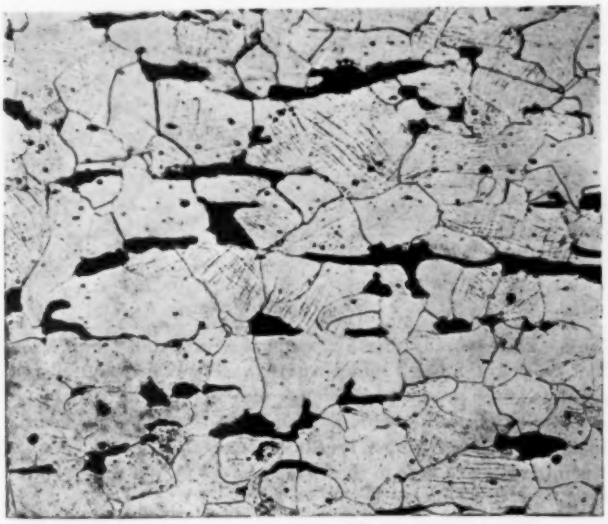


FIG. 3 SLIP LINES PRODUCED BY SQUEEZING SPECIMEN—AT 100 DIAMETERS



FIG. 6 SAME AS FIG. 5 BUT AT 1000 DIAMETERS

Whereas a slip line may be a phenomenon affecting material in a section having a thickness equal to a few unit lengths of the space lattice, several Angstroms wide, an X line embraces a section of metal at least 0.0001 in. wide.

It is generally accepted that etching effects in crystal aggregates are due to differences in the solubility of the various constituents. In pure substances or in solid solutions where the chemical composition is uniform, differences in appearance noted under the microscope are due to the differences in solubility as a result of orientation. X lines can probably be explained on the basis of one or more of the following hypotheses:

1 They may represent sections of a crystal containing greater

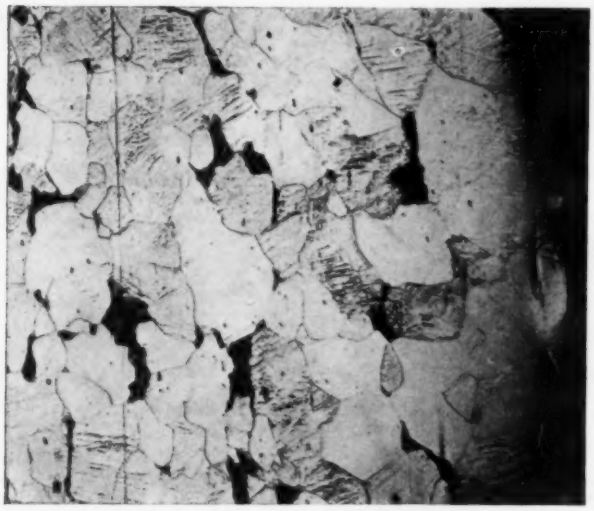


FIG. 7 SECTION THROUGH AXIS OF BENDING AFTER SPECIMEN WAS BENT THROUGH 180 DEGREES AND BACK—THE X LINES FORMED WILL STILL BE THERE AFTER REPOLISHING AND RE-ETCHING—NOTE X LINES MAKE ANGLES LESS THAN PLUS OR MINUS 45 DEGREES WITH AXIS OF BENDING—AT 100 DIAMETERS

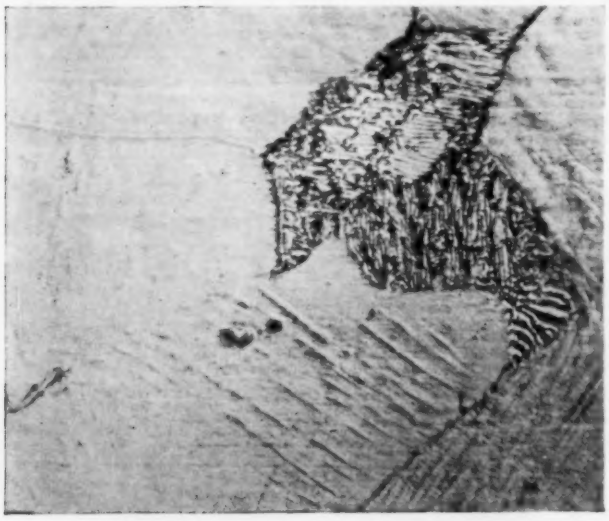


FIG. 8 SAME AS FIG. 7 BUT AT 1000 DIAMETERS

strain and therefore, being at a different electrolytic potential, dissolve at a different rate in the etching medium.

2 They may represent sections of metal having distorted, partially disorganized, or otherwise imperfect unit cells and thus exert a greater solution pressure.

3 They may represent sections of metal, the orientation of whose lattice is slightly different from that of the rest of the crystal. Howe[32] inclines to the last explanation when he says: "Hence these X bands may be twinned layers, or at least layers of different even if not symmetrically different orientation."

Failure by Bending. It was found necessary to bend this steel through 180 deg. and back three times before failure was induced.

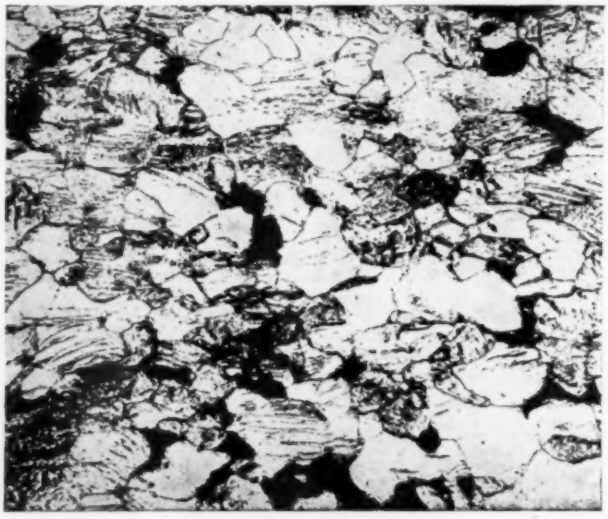


FIG. 9 SPECIMEN WAS BROKEN BY BENDING BACK AND FORTH THROUGH 180 DEGREES THREE TIMES—PHOTOMICROGRAPH AT 100 DIAMETERS SHOWS EDGE OF FRACTURE—NOTE THAT X LINES PARALLEL THE AXIS OF BENDING MUCH MORE CLOSELY THAN IN FIG. 7—IN MOST CASES THE X LINES MAKE ANGLES LESS THAN 10 DEGREES WITH AXIS OF BENDING



FIG. 10 SECTION ONE-HALF INCH BACK FROM BINDING FRACTURE—NOTE BURST GRAIN BOUNDARIES AND CRACKED CRYSTAL—AT 100 DIAMETERS

The stresses naturally were a mixture of tension and compression. Photomicrograph Fig. 9 shows at 100 diameters the fractured edge which had been preserved by copper plating for 36 hr. Although not readily apparent, careful examination will show that the fracture is transcrystalline, that is, it passes through the crystals. It is interesting to note in the vicinity of the failure that, (1) the

[32] H. M. Howe, "The Metallography of Steel and Cast Iron," p. 455, edition of 1916, McGraw-Hill Book Co.

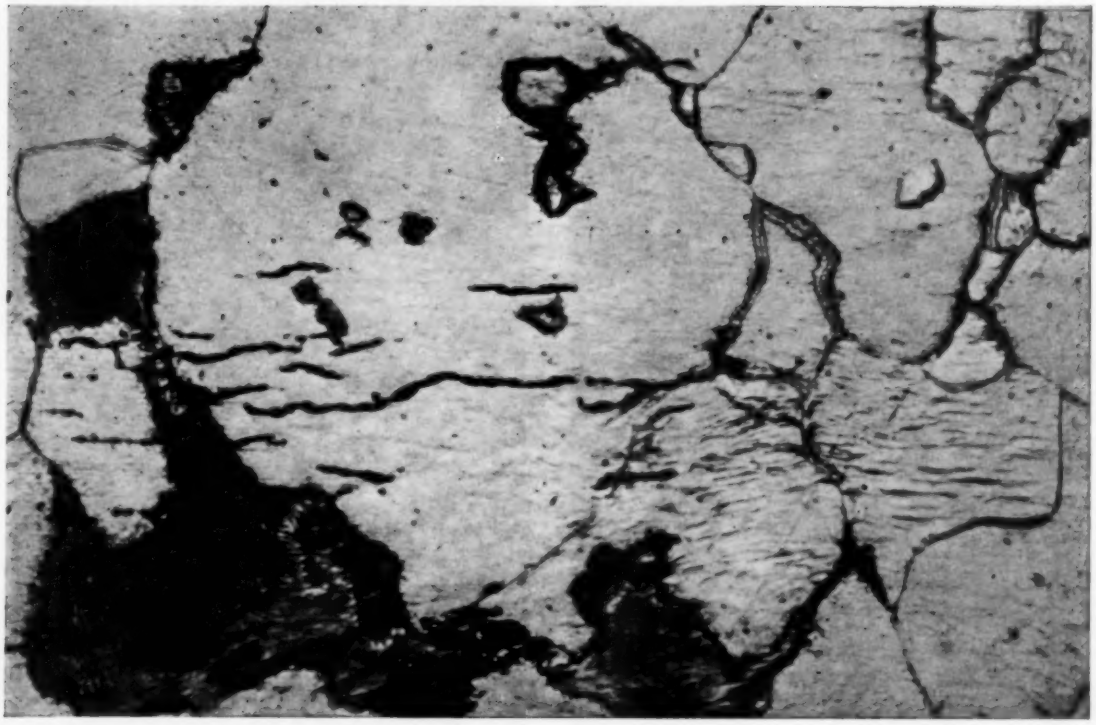


FIG. 11 SAME SECTION AS IN FIG. 10 BUT AT 500 DIAMETERS—NOTE THAT BENDING FRACTURES RUN THROUGH THE GRAINS

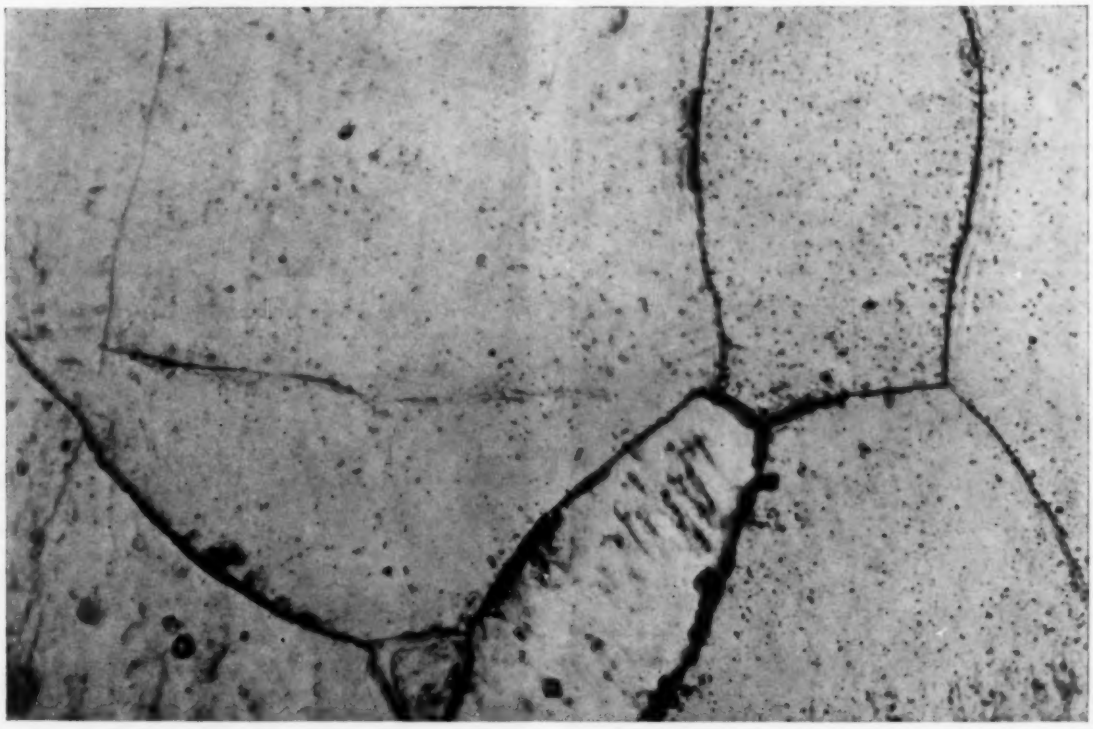


FIG. 12 BENDING FAILURE AT 500 DIAMETERS—NOTE BURST GRAIN BOUNDARIES IN LOWER PORTION AND THE CRACKS IN THE LARGEST CRYSTAL

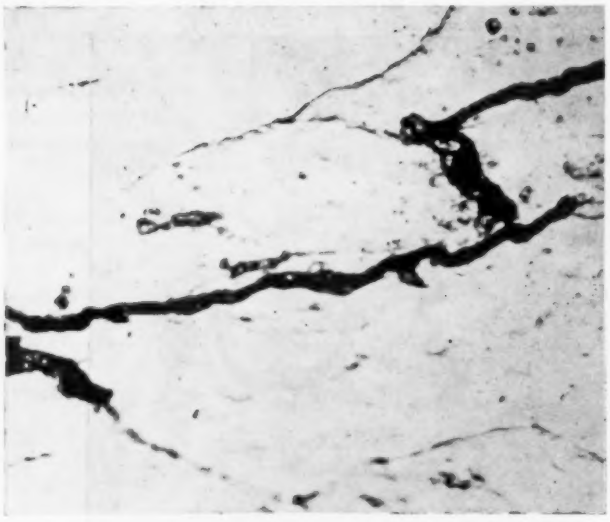


FIG. 13 A LONG TRANSCRYSTALLINE CRACK DUE TO BENDING—AT 1000 DIAMETERS

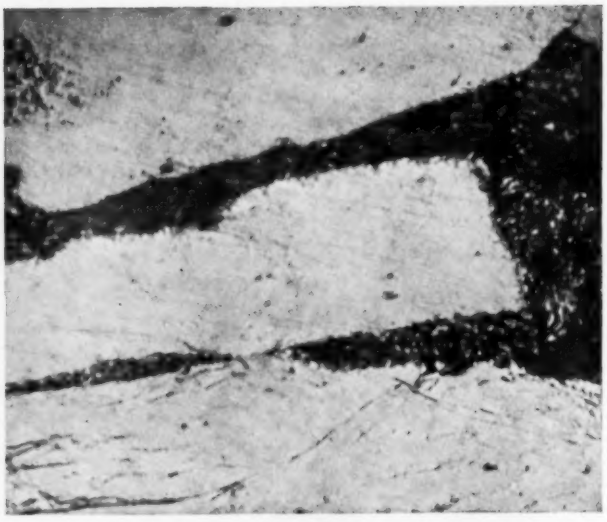


FIG. 16 SAME AS FIG. 14 BUT AT 1000 DIAMETERS—PROCESS OF ELONGATING THE PEARLITE CONSTITUENT AND THE FLOW OF FERRITE ARE SHOWN



FIG. 14 EDGE OF SPECIMEN SUBJECTED TO COMPRESSIVE STRESS—AT 100 DIAMETERS

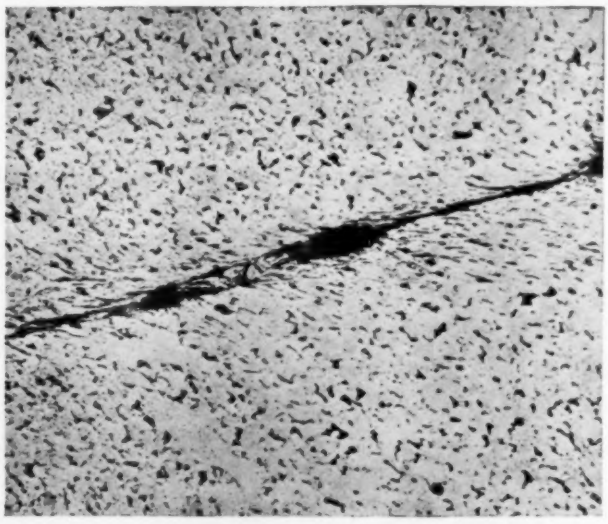


FIG. 17 COMPRESSION FAILURE AT 100 DIAMETERS—DISTORTION OF GRAIN TOOK PLACE ONLY ALONG LÜDER LINES

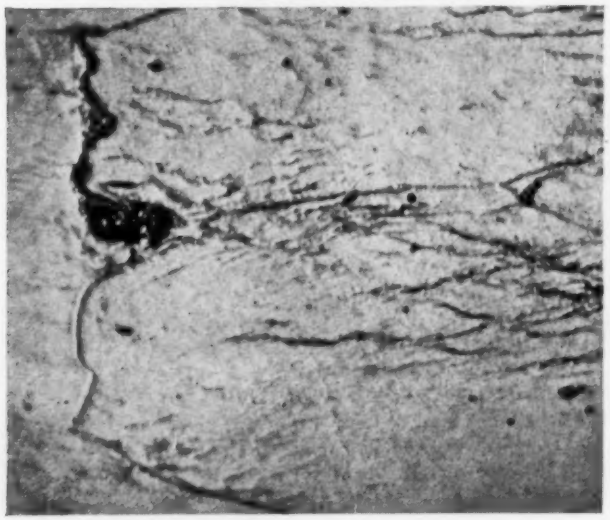


FIG. 15 SAME AS FIG. 14 BUT AT 1000 DIAMETERS—NOTE INCIPIENT FAILURES IN THE FERRITE

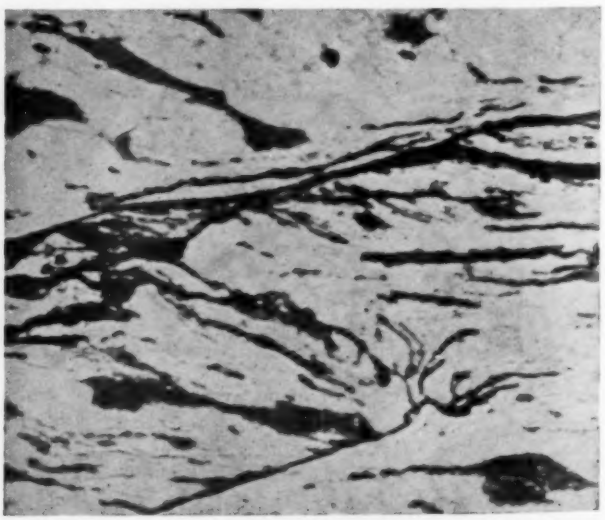


FIG. 18 PHOTOMICROGRAPH AT 1000 DIAMETERS OF COMPRESSION FAILURE

crystals are deformed as would be expected, and that (2) a larger portion of the grains exhibit strain lines or X lines than were visible in photomicrograph Fig. 7. Some of the lines are double so that it would not be incorrect to term them X bands. Few

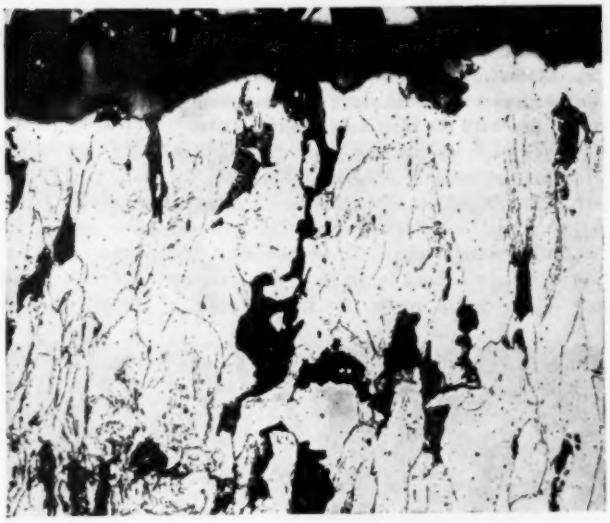


FIG. 19 EDGE OF TENSION FRACTURE—FAILURE IS ACROSS THE GRAIN—AT 100 DIAMETERS



FIG. 21 CRACKING DUE TO TENSION SHOWN AT 1000 DIAMETERS —THIS SECTION WAS ABOUT ONE-HALF INCH BACK FROM THE FRACTURE—CRACKING TOOK PLACE IN THE PEARLITE ALONG A LINE MAKING AN ANGLE OF 45 DEGREES TO THE APPLIED STRESS

of the X lines and X bands, the directions of which are assumed to be indicative of the orientation of the crystals, make angles greater in magnitude than about plus or minus 10 deg. with the axis of bending. Such a condition may well be explained on the basis of movement of the grains in the aggregate approaching as limit a condition of identical orientation.

Photomicrograph Fig. 10, also at 100 diameters, shows an area more remote from the edge of the fracture. Less severe X lines

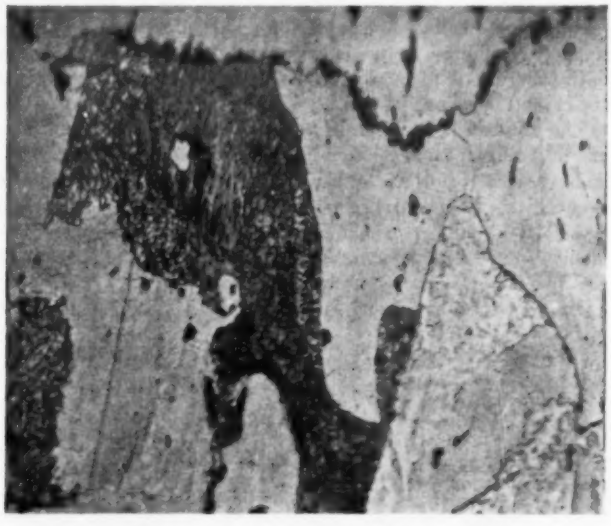


FIG. 20 TENSION FRACTURE AT 500 DIAMETERS—NOTE FRAGMENTATION OF CARBIDES AND ABSENCE OF STRAIN LINES IN ROW OF CRYSTALS ADJOINING FRACTURED EDGE



FIG. 22 PHOTOMICROGRAPH AT 1000 DIAMETERS SHOWING 45 DEGREE SHEAR FAILURES IN PEARLITE DUE TO TENSILE PULL

and a larger portion of undeformed crystals can be seen. Two important points are brought out in this picture. First, some of the crystals seem to have abnormally wide grain boundaries and, second, failure can be seen taking place in some of the grains. Upon examination of the wide boundaries at higher magnification, shown at 500 diameters in photomicrograph Fig. 11, the

boundary around a small grain possessing X lines is seen to be ruptured. The crystals are actually torn apart. This same condition is shown in photomicrograph Fig. 12 also at 500 diameters. In the former the formation of a series of cracks can be seen running across in a direction parallel to the axis of bending and parallel to the X lines in two smaller adjacent crystals. It is not unlikely to suppose that this large crystal also possessed X lines until failure took place which relieved the strains and permitted the crystal to recover elastically.

The presence of burst grain boundaries shows that technically one may think that a bending fracture is sometimes intercrystalline. Actually, however, a bending failure is transcrystalline as shown by the fractured edge in photomicrograph Fig. 9, in cracked crystals in photomicrographs Figs. 10 and 11, and even more convincingly at 1000 diameters in photomicrograph Fig. 13, which clearly shows a long transcrystalline crack.

Failure by Compression. Another strip of this same boiler tube steel was subjected to a compressive stress of about 125,000 lb. per sq. in., which flattened the specimen considerably. After being copper plated for 36 hr. it was polished and etched. Photomicrograph Fig. 14 at 100 diameters shows a condition of flattening of the crystals. The X lines parallel the edge of the specimen fairly well. Failure takes place in two different manners. The softer constituent, the ferrite, distorts along its gliding planes in those crystals which are favorably oriented. Two such crystals can be seen in photomicrograph Fig. 15 at 1000 diameters. Distortion has taken place up to the point of incipient cracking of the crystals. Again it is to be noted that the failures make angles of less than plus or minus 30 deg. with an axis normal to the compressive stress. Such deformation undoubtedly permits less favorably oriented crystals to move by manner of plastic flow until they too are oriented in a direction favorable to distortion by the existing stress.

On the other hand no definite cracking can be observed in the pearlite. The pearlite is elongated in the direction normal to the compressive force as is shown at 1000 diameters in photomicrograph Fig. 16. The pearlite does probably crack, but it cannot be observed readily at this magnification due to the darkness of the constituent. The portions cracked off are then carried along much as logs are floated along on the surface of a stream, and the voids filled with the relatively plastic ferrite.

Since no serious cracking or fracturing occurred, a less ductile steel was then used. A $^3/_4$-in. round bar, 3 in. long of 0.20 per cent carbon steel was subjected to a compressive load of 100,000 lb. Cracking took place along the Lüders lines, planes making 45-deg. angles with the axis of compression. Microscopic examination revealed the fact that no appreciable distortion of the crystals took place except along the Lüders lines. This situation is clearly shown at 100 diameters in photomicrograph Fig. 17, which was taken along a crack. Photomicrograph Fig. 18 at 1000 diameters demonstrates the stringing out of the ferrite and pearlite until the axes of each grain paralleled the axis of failure. Whether the failure goes through or around the grains cannot be unequivocally demonstrated although it is most likely of a transcrystalline nature due to slip.

Failure by Tension. A flat strip of the boiler-tube steel was next caused to fail by a tensile stress. After copper plating to preserve the fracture, the specimen was polished and etched for microscopic examination. Photomicrograph Fig. 19 of the edge of the fracture at 100 diameters shows the elongation of the grains in the direction of the tensile stress. Fracture has taken place across the crystals as is demonstrated even more clearly in

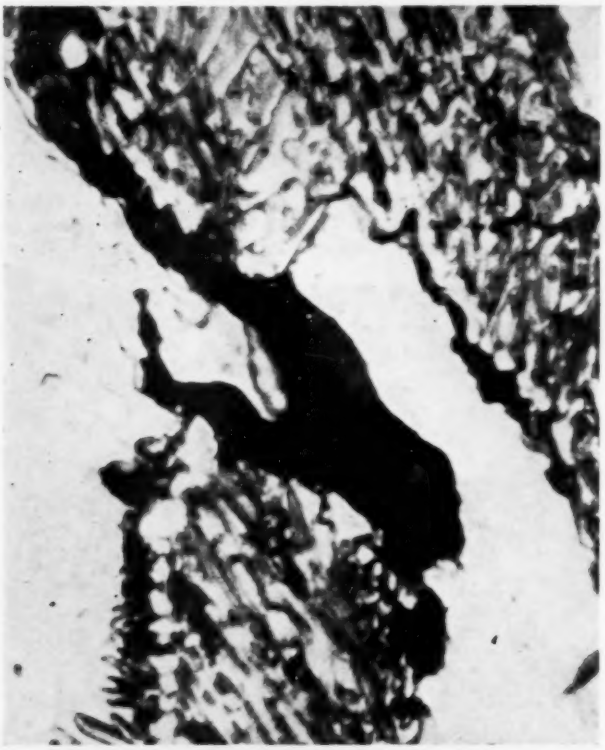


FIG. 23 PHOTOMICROGRAPH AT 3000 DIAMETERS OF A 45-DEGREE FAILURE DUE TO TENSION—THE FLOW OF FERRITE HAS CAUSED ROTATION OF THE LOWER FRAGMENT OF THE PEARLITE GRAIN—NOTE THE FERRITE FLOWING INTO THE VOID AT THE RIGHT SIDE



FIG. 24 TENSION SPECIMEN AT 1000 DIAMETERS—THE SLENDER PEARLITE GRAIN AT THE RIGHT HAS SEPARATED ALONG 45 DEGREE LINES AND FERRITE FROM THE SINGLE CRYSTAL AT ITS LEFT HAS FLOWED IN, COMPLETELY FILLING THE VOIDS

photomicrograph Fig. 20 at 500 diameters. The latter photomicrograph was taken with vertical lighting and hence does not bring out in full detail the X lines in the crystals. Close examination, however, will reveal the surprising fact that, although evidences of strain other than elongation of the crystals are visible in some grains, those in which fracture took place do not reveal any strain lines whatever. The small particles of dark material in the large ferrite grains are undoubtedly fragments of carbide particles. Those in the copper layer are presumably particles of pearlite which were torn loose in polishing and became embedded in the soft copper.

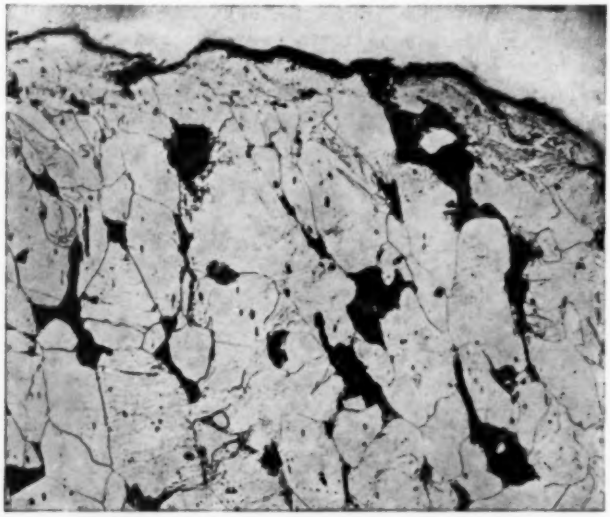


FIG. 25 PHOTOMICROGRAPH AT 100 DIAMETERS OF THE EDGE OF IMPACT FAILURE

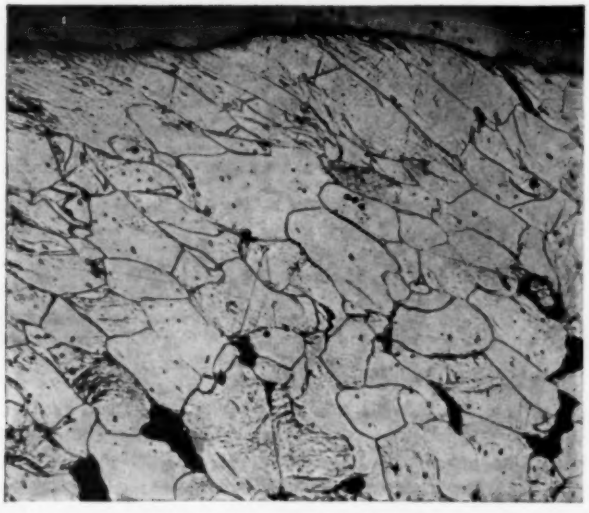


FIG. 26 EDGE OF IMPACT FAILURE AT 100 DIAMETERS—IN THIS SECTION SHEAR WAS AN IMPORTANT FACTOR

Naturally a fracture does not take place suddenly but occurs in stages along a plane which has been weakened by an accumulation of failures in individual crystals. Failure in mild carbon steels through tension seems to start in the relatively strong but brittle pearlite constituent. This was first noted by Rosenhain.[33]

As a tensile stress is applied the ductile ferrite, by elongating and cold working, constantly increases the stress existent in the pearlite which undergoes no appreciable elongation. Evidently the cohesive force between grains is greater than the strength of the pearlite. The pearlite fails not necessarily along any slip plane, but along a shear plane at about 45 deg. to the direction of the tensile stress. Photomicrograph Fig. 21 at 1000 diameters shows a typical failure occurring in a pearlite grain. The fracture cuts across the laminations in the eutectoid; the dark portion between the two halves of the crystal represents a void. The two portions were clearly parts of a single pearlite particle; after shearing along a 45 deg. plane, due to the movement of the ferrite, the two portions appear to be separated about 0.0005 in. Photomicrograph Fig. 22 also at 1000 diameters shows a similar condition except that fracture took place not along an unidirectional line but along several lines the direction of each being plus or minus 45 deg. to the line of stressing.

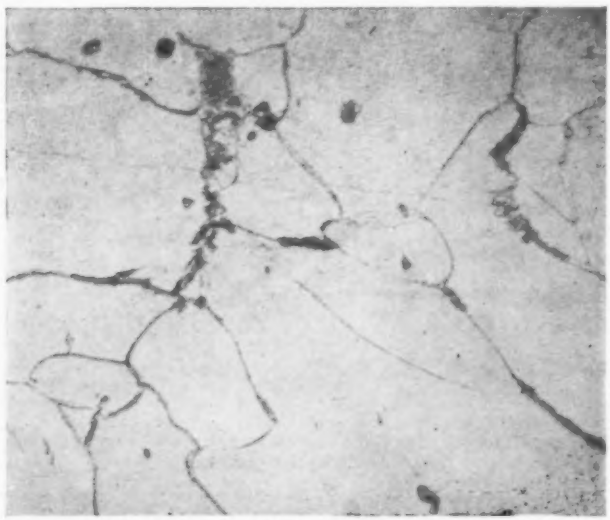


FIG. 27 BURST GRAIN BOUNDARIES DUE TO IMPACT—AT 500 DIAMETERS

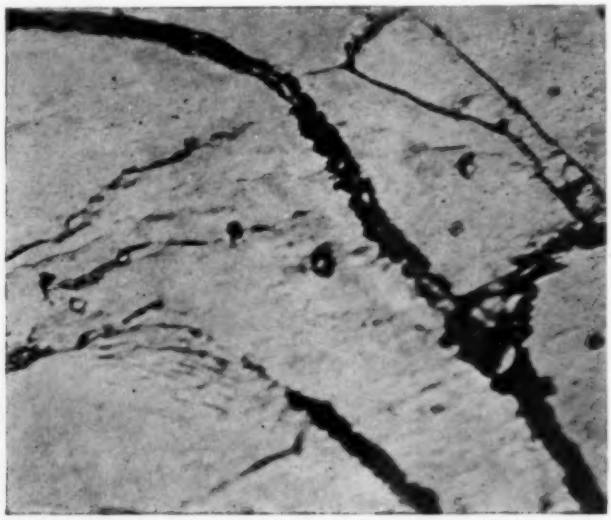


FIG. 28 BURST GRAIN BOUNDARIES AT 1000 DIAMETERS

The next stage in failure by tension is shown at 3000 diameters in photomicrograph Fig. 23. Rotation of the lower fractured portion of the pearlite can be seen to have occurred probably be-

[33] W. Rosenhain, "Deformation and Fracture in Iron and Steel," Jl. Iron and Steel Institute, vol. 70, pp. 189–228, 1906.

cause the long arm at the left of the picture extended outward into the moving stream of ferrite. Rotation was accomplished much as a log floating crosswise in a river and fixed at one end soon tends to assume a position paralleling the direction of flow. A second interesting feature brought out in this picture, tending to illustrate



FIG. 29 BURST GRAIN BOUNDARIES AND STRAIN LINES DUE TO IMPACT—AT 1000 DIAMETERS



FIG. 30 STRAINED AND UNSTRAINED CRYSTALS IN IMPACT SPECIMEN AT 1000 DIAMETERS—NOTE UNEVEN CONTOUR OF METAL—THESE FIGURES ARE PROBABLY THORNS, A TYPE OF NEUMANN BAND

the plastic nature of ferrite under stress is the fact that ferrite is flowing into the right-hand side of the void. It is generally known that if a specimen is left under a condition of stress in excess of the elastic limit for a period of time and then ruptured, the tensile strength proves to be greater than if a tensile-strength measurement had been made in the usual manner. It is not improbable that incipient local failures in the pearlite are healed by the plastic flow of the softer constituent, and that even new atomic bonding may be effected. The completion of this flowing-in process is shown at 1000 diameters in photomicrograph Fig. 24. The slender carbide particle at the right has cracked along 45 deg. shear lines, and the voids created in this manner have been apparently filled by material from the single-ferrite crystal adjoining it on the left. X lines can be seen in some of the ferrite grains in the vicinity. But again, no X lines can be seen in crystals which have been in contact with a crack or a void. Tension failure is undoubtedly a pure shear phenomenon.

Failure by Impact. A specimen of this boiler-tube material was notched and fractured by means of a blow of a hammer. In



FIG. 31 FATIGUE FRACTURE AT 100 DIAMETERS—THE FRACTURE GOES THROUGH THE GRAIN—THERE IS NO DISTORTION OF THE CRYSTALS

so ductile a metal as this, a failure by pure impact can hardly be expected. Although one would expect some of the grains to fail through impact, a large number tend to bend and, hence, to fail by shear. Photomicrograph Fig. 25 shows at 100 diameters a portion of the fracture in which impact played a major role. Strain lines and distorted Neumann bands of the wedge type can be seen. The grains in general are not very severely deformed. Photomicrograph Fig. 26 also at 100 diameters shows that portion of the edge of the fracture which was probably due to shear. The grains seem inclined in the direction of the applied stress. The edge of both of these sections is so indistinct that it is not possible to state definitely whether fracture occurred along the boundaries or through the grains. Photomicrograph Fig. 27 at 500 diameters shows a number of burst grain boundaries and some strain lines roughly paralleling the fracture. The same things are shown at 1000 diameters in photomicrographs Figs. 28 and 29. In these latter two pictures are shown strain lines in the

same crystals which are in contact with voids, a combination less common in the failures due to bending, compression, or tension. Because of the burst grain boundaries, failure by impact must be in part at least an intercrystalline phenomenon.

Some of the grains of ferrite are not distorted at all whereas others are very badly strained. Photomicrograph Fig. 30 shows at 1000 diameters such a case occurring in two adjacent crystals. The short parallel ridges (whose directions make angles of less than 45 deg. with the axis of failure) have been described by Osmond[34] as thorns, a special type of Neumann band.

A Neumann band by definition is a section of a crystal the orientation of whose crystallographic planes is the mirror image of the original crystallographic planes. This reversal of orientation is brought about in ferrite by sudden severe shock. Perhaps if the shock had been more severe than was the case in this sample, rupture would have taken place in shorter time and changes in orientation would have been effected without further deformation of the grain. Thus the usual straight Neumann band might have been produced.



FIG. 32 SAME AS FIG. 31 BUT AT 500 DIAMETERS

Obviously, since a thorn is a structure which is revealed only upon etching, the same possible explanations for their appearance on etching must be true as in the case of X bands. Thorns undoubtedly represent a severe disturbance of the orientation of the crystallographic axis in a portion of a grain. Since it is not possible by X-ray analysis or any other manner to measure the degree of change in orientation, it cannot be determined whether the axes are symmetrically reversed, and hence whether a thorn is a true Neumann band. Thorns might even represent a structure intermediate between a Neumann band and the original crystal.

It has been demonstrated that failures by bending, by compression, by tension, and by impact all occur with deformation of the crystals. The fractures are through the crystals. Impact failures show the presence of burst grain boundaries, but even so one could not accurately classify an impact failure as intercrystalline.

FAILURES NOT ACCOMPANIED BY DEFORMATION OF THE CRYSTALS

Failure in mild steels not accompanied by deformation of the crystals makes up the second general type. In this classification

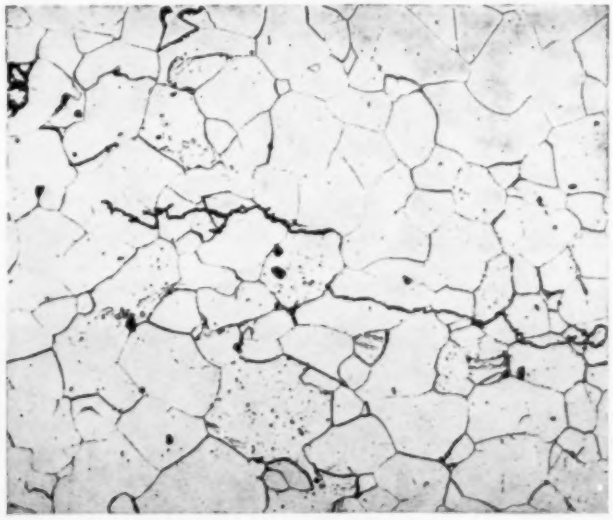


FIG. 33 FATIGUE CRACK IN FERRITE ONE-HALF INCH FROM FRACTURE—AT 100 DIAMETERS

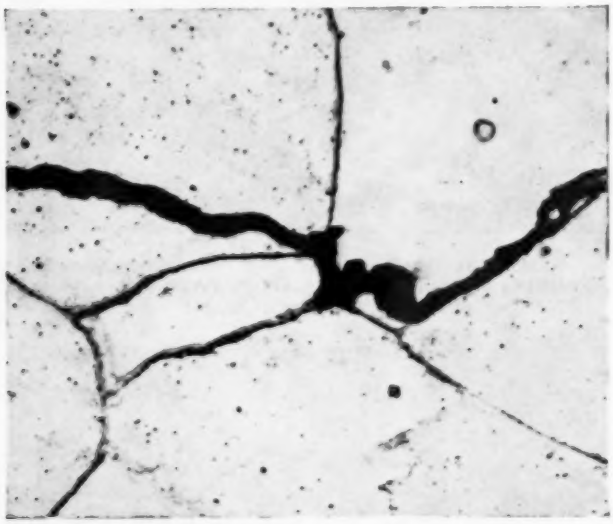


FIG. 34 FATIGUE CRACK AT 1000 DIAMETERS ILLUSTRATING TRANSCRYSTALLINE NATURE OF FAILURE

also the fractures or cracks may be divided into those which are transcrystalline and those which are intercrystalline.

Failure by Fatigue. Fatigue cracks illustrate the first type. A specimen of the boiler-tube material was caused to fail in an Upton Lewis type fatigue machine. After copper plating for 36 hr. photomicrographs Fig. 31 and 32 were taken to show at 100 and 500 diameters respectively the appearance of the fracture which is clearly across the crystals. Photomicrographs Fig. 33 and 34 at 100 and 1000 diameters respectively show a transcrystalline fatigue crack which occurred about $\frac{1}{2}$ in. from the

[34] Osmond, Fremont, and Cartaud. Rev. de Metallurgie, Mem., vol. 1, p. 28ff, 1904.

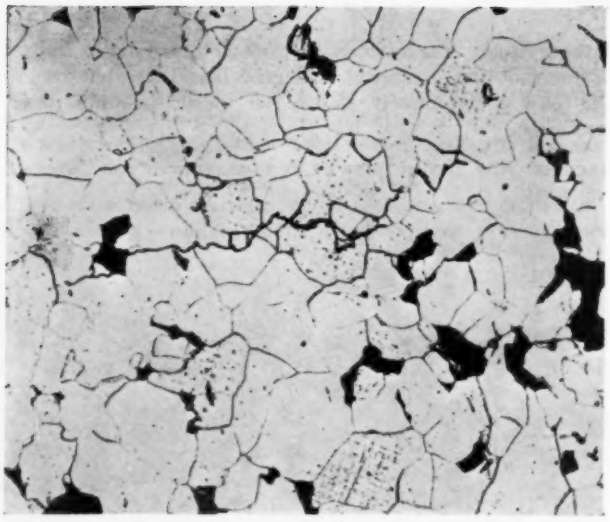


FIG. 35 FATIGUE CRACK AT 100 DIAMETERS

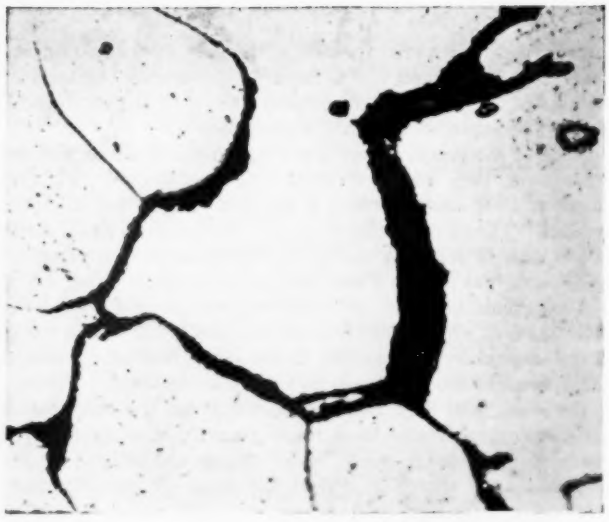


FIG. 38 SAME AS FIG. 37 BUT AT 1000 DIAMETERS

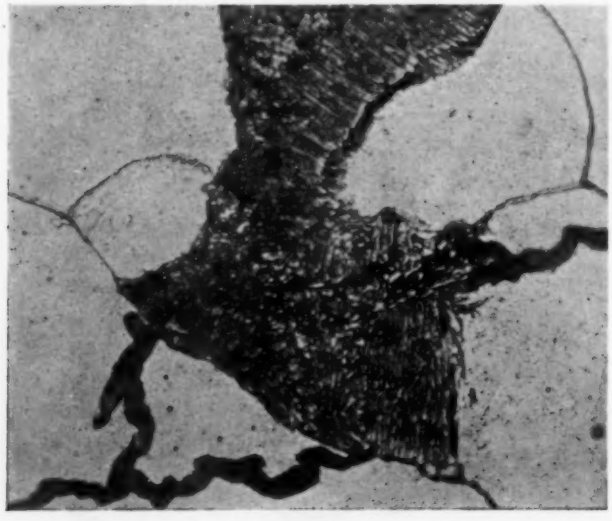


FIG. 36 SAME AS FIG. 35 BUT AT 1000 DIAMETERS—NOTE THAT CRACKING IS STOPPED BY PEARLITE GRAIN BUT PROCEEDS AGAIN IN THE NEXT FERRITE CRYSTAL



FIG. 39 INTERCRYSTALLINE CRACKING IN WROUGHT IRON DUE TO EMBRITTLEMENT BY CONTACT WITH PHOSPHATE LIQUORS—SHOWN AT 100 DIAMETERS

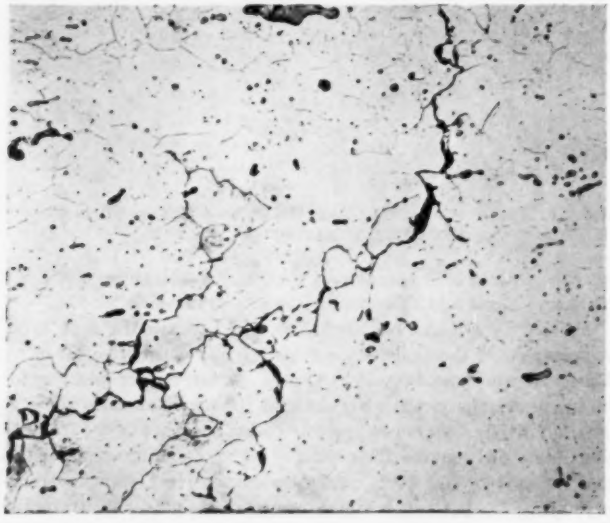


FIG. 37 INTERCRYSTALLINE CRACK DUE TO EMBRITTLEMENT BY CONTACT WITH CAUSTIC SODA LIQUORS—SHOWN AT 100 DIAMETERS

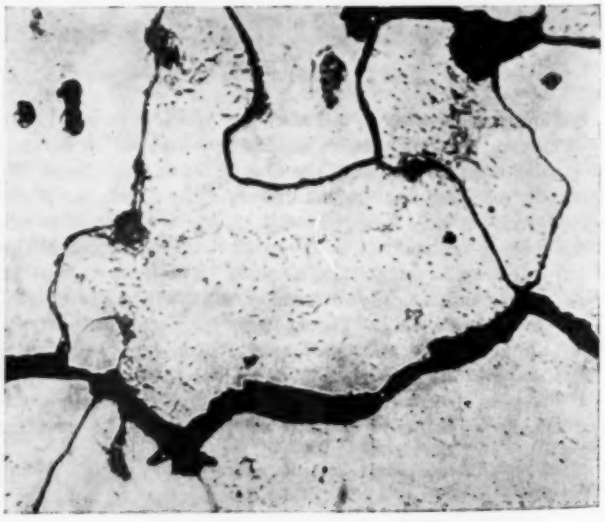


FIG. 40 SAME AS FIG. 39 BUT AT 1000 DIAMETERS

break. The probable mechanism of fatigue cracking is shown at 100 and 1000 diameters in photomicrographs Fig. 35 and 36. Cracking takes place in the ferrite in stages in a manner such that

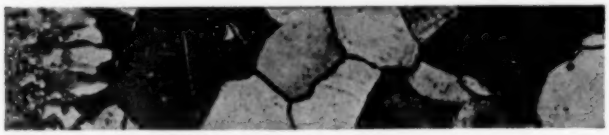


FIG. 41 INTERCRYSTALLINE CRACK IN IRON DUE TO OCCLUSION OF HYDROGEN (PHOTOMICROGRAPH BY PFEIL, PROCEEDINGS OF ROYAL SOCIETY, LONDON)

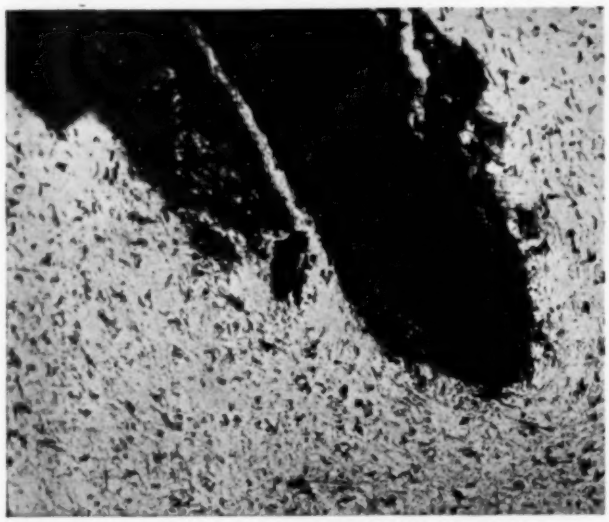


FIG. 42 CRACK IN BOILER TUBE DUE TO BENDING THE TOP PORTION IN REMOVING IT FROM TUBE SHEET—AT 100 DIAMETERS

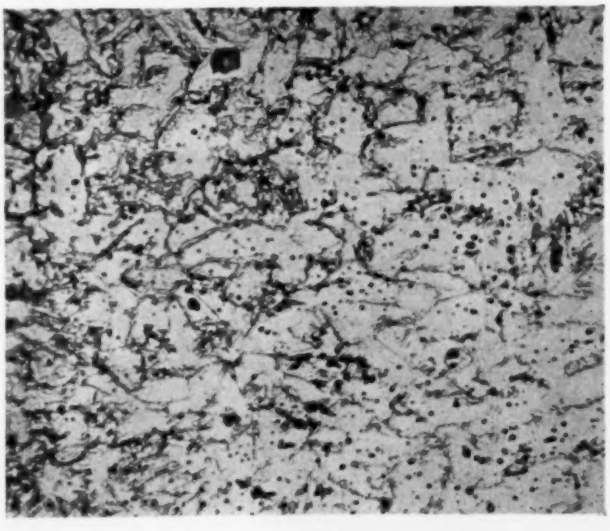


FIG. 43 PHOTOMICROGRAPH AT 100 DIAMETERS OF TYPICAL SECTION OF BRITTLE BOILER TUBE WHICH HAD BEEN IN SERVICE OVER TWENTY YEARS—BRITTLENESS WAS PROBABLY DUE TO AGING —NOTE PECULIAR GRAIN BOUNDARIES

the crack proceeds by short breaks varying in direction. Cracking does not proceed through the pearlite grains. The pearlite undoubtedly transmits the stresses to the next ferrite grain which will then crack in the usual manner. When actual fracture occurs the pearlite fails because it alone is bearing all the load.

Failure by Caustic Embrittlement. Intercrystalline cracking without deformation of the grains is sometimes brought on through the agency of sodium hydroxide. Photomicrographs Figs. 37 and 38 show at 100 and 1000 diameters an intercrystalline crack in a low-carbon steel tube. This tube had been in service in a vertical-tube caustic-soda evaporator for about one year.

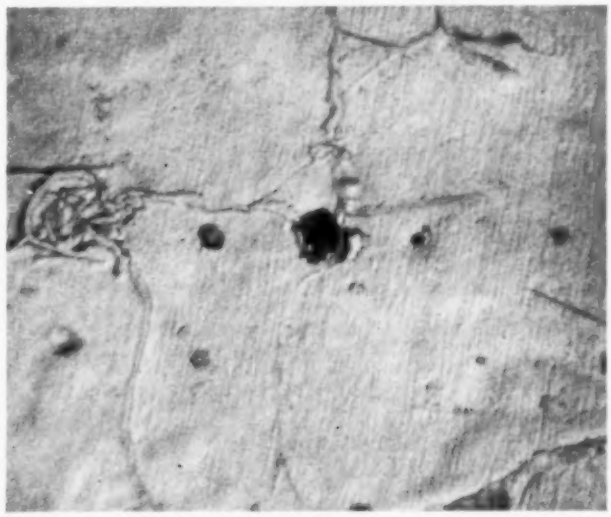


FIG. 44 INCLUSIONS IN BRITTLE BOILER TUBE—SHOWN AT 2000 DIAMETERS—THIS TUBE IS PROBABLY OF BESSEMER STEEL

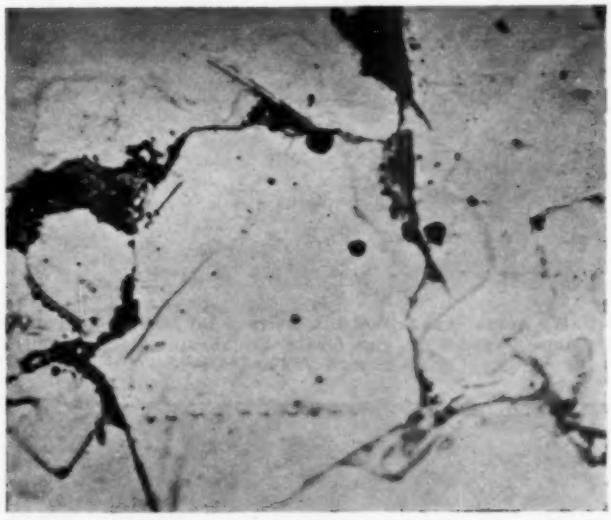


FIG. 45 TYPICAL STRUCTURE OF AGED BOILER TUBE AT 1000 DIAMETERS—NOTE THE DIRTY GRAIN BOUNDARIES AND THE UNEVEN SURFACE OF THE METAL

No stress other than the weight of the tube had been applied. The temperature of operation was 190 deg. fahr., and the crack shown occurred $1^1/_2$ in. below the tube sheet. At the time of removal the tube was exceedingly brittle. Its ductility returned upon being exposed to air for some time. This example is given because it represents a true case of caustic cracking.

Intercrystalline cracking and brittleness can also take place in phosphate solutions. Photomicrographs Figs. 39 and 40 show cracked portions of a wrought-iron hanger strap for a steam coil



FIG. 46 SECTION AT 1000 DIAMETERS OF BRITTLE BOILER TUBE SHOWING THORNS, A TYPE OF NEUMANN BAND—THESE WERE PROBABLY PRODUCED BY HAMMERING WHICH WAS DONE TO REMOVE THE TUBE FROM THE TUBE SHEET



FIG. 47 AFTER ANNEALING THE BRITTLE BOILER TUBE AT 1600 DEGREES FAHRENHEIT THE GRAIN BOUNDARIES BECAME QUITE CLEAN—SHOWN AT 100 DIAMETERS



FIG. 51 SAME AS FIG. 50 BUT AT 4000 DIAMETERS

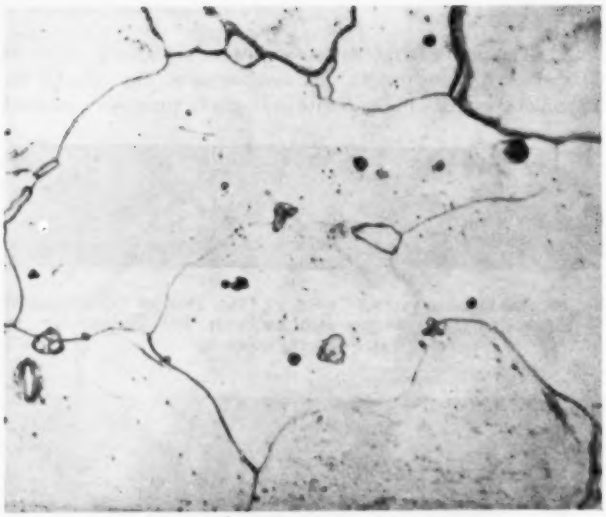


FIG. 48 SAME SPECIMEN AS SHOWN IN FIG. 47 AT 1000 DIAMETERS —NOTE THAT THE SURFACE OF THE METAL IS SMOOTH AND NORMAL IN ALL RESPECTS



FIG. 49 A SINGLE CRYSTAL IN THE AGE-EMBRITTLED BOILER TUBE AT 850 DIAMETERS—NOTE LINES OR CRACKS WITHIN CRYSTAL

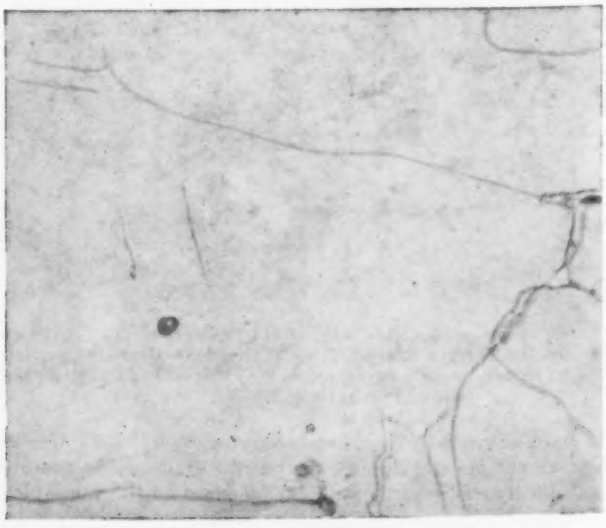


FIG. 50 SAME AS FIG. 49 BUT AT 2000 DIAMETERS—ONE CRACK EITHER IS STOPPING IN OR STARTING FROM AN INCLUSION—TRACES OF OLD GRAIN BOUNDARIES CAN BE SEEN

from a tank used for dissolving sodium phosphate. This strap did not support any appreciable load. It failed after eight weeks operation at 140 deg. fahr. in a solution containing 55 per cent of commercial trisodium phosphate calculated on the basis of the anhydrous salt. After exposure to the air the original ductility returned to the uncracked portion of the metal.

Pfeil[28] reports similar cracking while cathodically pickling coarse-grained iron in sulphuric acid. Photomicrograph Fig. 41 is a reproduction of an intercrystalline crack produced by him in this manner. He concluded that occluded hydrogen weakens the grain boundaries.

Actual Boiler Metal Failures

Up to this point failures have been discussed which were due to known causes. Failed boiler materials from service are usually not quite so characteristically clean cut as to type and easy to identify as were these more or less ideal cases. A few actual cases falling under the above types will be given.

Methods of Investigation. A few purely routine considerations must be briefly explained because it is only in the light of the results of these that approach can be made to an interpretation of the metallographic study. Before attempting any serious analysis of a failed part, data should be secured regarding the duration and temperature of service, the probable stresses to which the piece was subjected, the manner in which failure manifested itself such as by leaking, cracking, or by complete rupture, and any other facts that might have any bearing on the case.

A chemical examination of boiler steels will show them to fall roughly within the following limits of composition:

Carbon.....	0.10 to 0.30 per cent
Manganese.....	0.30 to 0.60 per cent
Sulphur.....	under 0.050 per cent
Phosphorus.....	under 0.050 per cent
Silicon.....	0.01 to 0.10 per cent

Some boiler tubing has been analyzed and found to contain even less than 0.01 per cent of silicon.

A metallographic examination of sections etched with 4 per cent Nital will reveal the relative grain size, the distribution of the carbides, whether or not the metal has been annealed, condition of working, and the presence of inclusions. Boiler tubing and plate often contain inclusions which if well dispersed rarely cause failure. A Rosenhain etch will reveal the presence of segregations and other non-homogeneities. One must not be over-hasty in ascribing cracks or ruptures solely to the presence of non-homogeneities as revealed by this etch because a large number of boiler tubes which rendered many years of satisfactory service have been examined, and under the Rosenhain etch showed severe ghost lines, phosphorus segregations, and other non-homogeneities. Wrought iron and Bessemer steel have been used widely in the past with fair success, and these as is well known contain many inclusions.

A metallographic examination of the failed area itself will yield the most valuable information. Failure usually manifests itself first through the formation of a crack; as the cracked area increases in size complete rupture of the piece may ultimately take place. The appearance of the fracture and of the metal in its vicinity will vary with the cause.

A study of a large number of boiler-material failures has led to a grouping of fractures under two main headings determined by the metallographic appearance. These are failures accompanied by deformation of the crystals, and failures not accompanied by deformation of the crystals. The principal subclassifications which can be made will distinguish between transcrystalline and intercrystalline fractures.

There is another distinctive type of failure observed when a boiler tube explodes due to overheating the metal. Because of the temperatures involved and the drastic thermal changes to which the metal is subjected, the crystal structure is far from normal, and no data can be obtained regarding the trans- or intercrystalline nature of the failure.

Failure by Bending. In removing a boiler tube a tool was used so that one end of the tube was badly crimped. A cracked surface was noted. It was then desired to know whether these cracks were due to caustic embrittlement, a matter of great concern to the plant from which the tube came, or to other causes. Photomicrograph Fig. 42 shows at 100 diameters a section near one of the cracks. This represents more or less of a tension failure. The metal was bent in one direction, the fiber in the side shown in the picture was in tension. Elongated crystals and the absence of fine secondary cracking can be seen. Undoubtedly this tube was somewhat brittle.

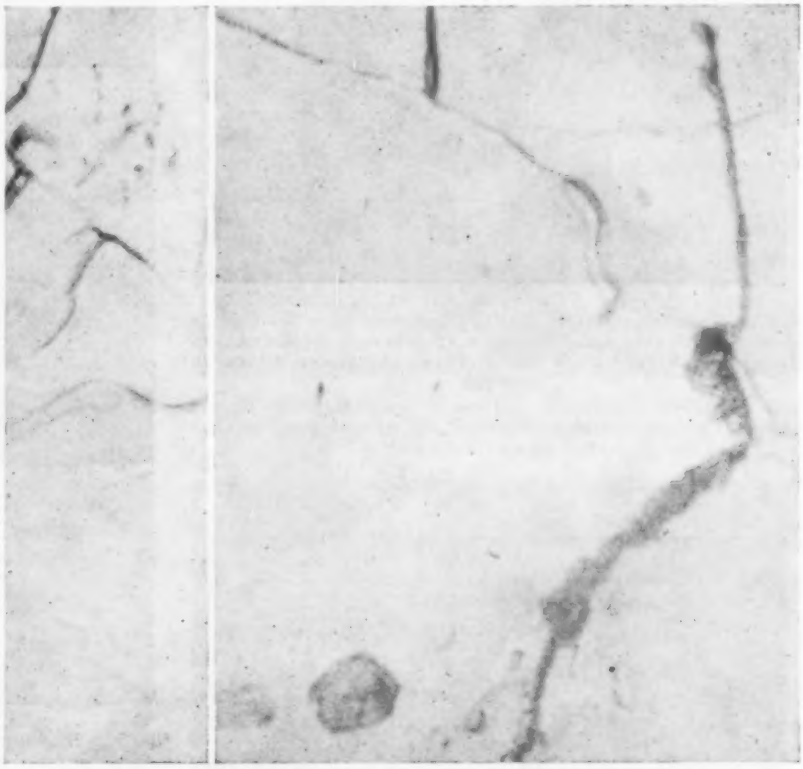


Fig. 52 Transcrystalline Cracks in Age-Embrittled Boiler Tube at 4000 Diameters

Failure by Impact and Aging. In striking, with a small hammer, the end of a boiler tube which had been in service from 1903 until 1928, pieces broke off which were so brittle that they fractured upon being dropped to the floor. Chemical analysis showed the presence of 0.087 per cent sulphur, indicating that it probably was of Bessemer steel. The presence of an exceedingly large number of inclusions is shown in photomicrograph Fig. 43 at 100 diameters and in photomicrograph Fig. 44 at 2000 diameters.



FIG. 53 PHOTOMICROGRAPH AT 100 DIAMETERS SHOWING CRACKS FORMED BY STRESS AND CORROSION AT JUNCTION OF SHANK AND HEAD OF A RIVET—NOTE SMALL TRANSCRYSTALLINE CRACK IN CENTER



FIG. 54 TRANSCRYSTALLINE CRACKS FORMED BY STRESS AND CORROSION AT JUNCTION OF SHANK AND HEAD OF RIVET SHOWN AT 100 DIAMETERS

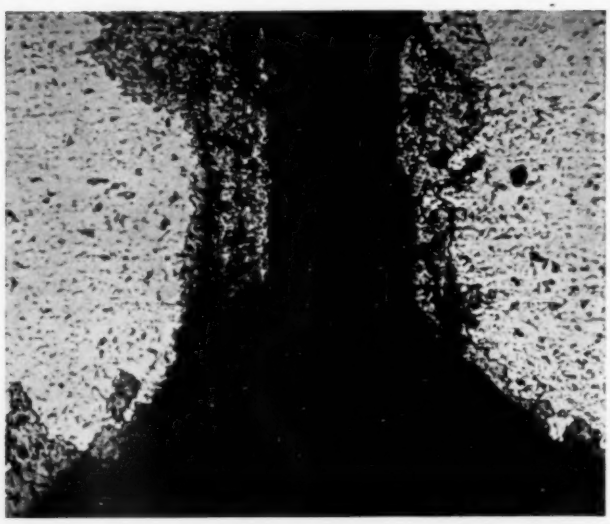


FIG. 55 FAILURE BY STRESS AND CORROSION IN A BOILER TUBE SHOWN AT 100 DIAMETERS—IN ROLLING IN THE TUBE A GOUGE WAS MADE ALONG ONE SIDE—CORROSION PROCEEDED UNTIL THE TUBE FAILED

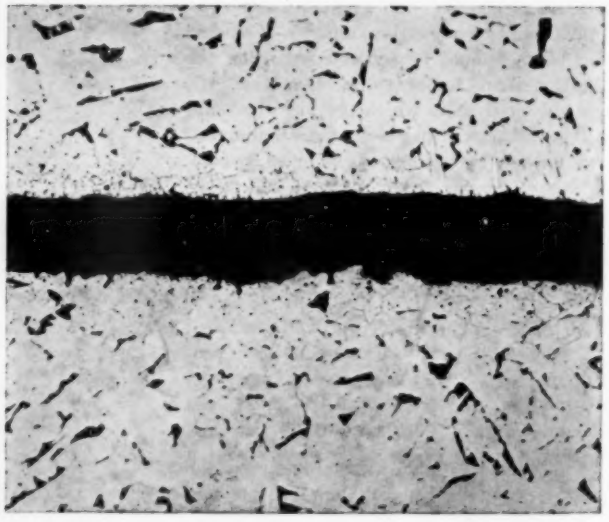


FIG. 56 FORGING LAP IN A BLOW-OFF PAD SHOWN AT 100 DIAMETERS

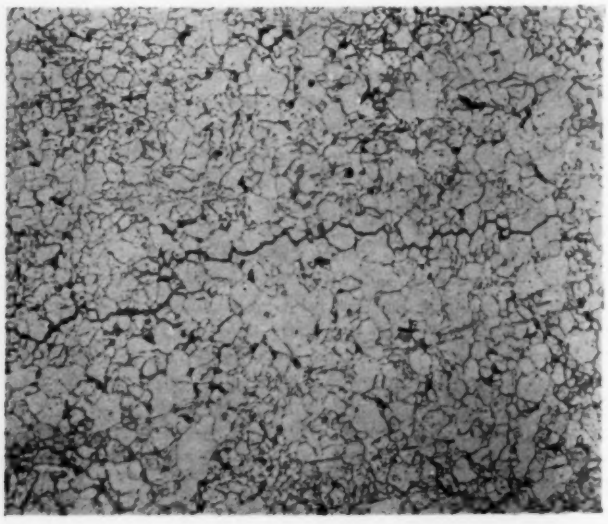


FIG. 57 CRACK IN TOP END OF BOILER TUBE AT 100 DIAMETERS

A study was made to determine the reason for the behavior of this metal. The metallographic appearance of this specimen was unusual. Referring back to photomicrograph Fig. 43 the grain boundaries can be seen to be wavy, indistinct, actually double in some cases. Photomicrograph Fig. 45 at 1000 diameters shows segregation of material at the boundaries. The first clue to the nature of the failure was gained by examination of the section shown at 1000 diameters in photomicrograph Fig. 46. In addition to all the peculiarities shown in the other illustrations, this

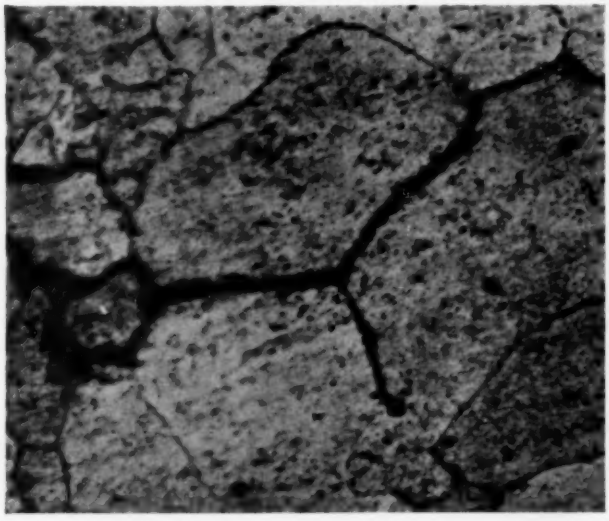


FIG. 58 SAME SECTION AS FIG. 57 BUT AT 1000 DIAMETERS SHOWING INTERCRYSTALLINE NATURE OF CRACK—THIS FAILURE WAS PROBABLY DUE TO CAUSTIC EMBRITTLEMENT

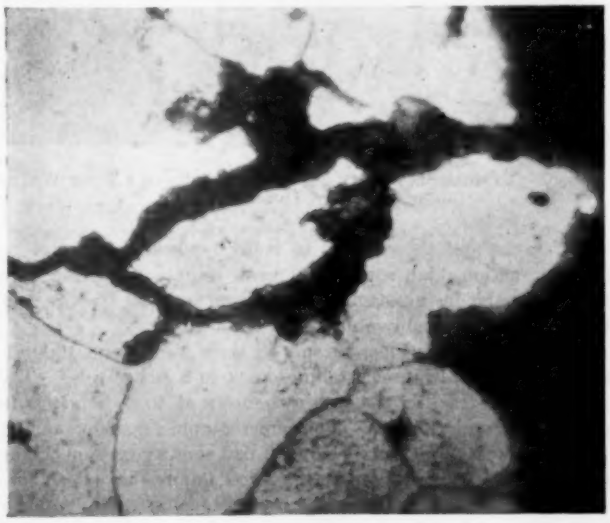


FIG. 59 INTERCRYSTALLINE CRACK DUE TO CAUSTIC EMBRITTLEMENT SHOWN AT 2000 DIAMETERS

section shows a series of fringes or thorns at the grain boundaries and presents an uneven surface giving the appearance of stamped impressions. This structure is identical with that of an impact failure shown in photomicrograph Fig. 30. This steel has undergone what is termed aging and has become embrittled through repeated heating and cooling. The original properties of an aged low-carbon steel can be restored by heating above the A_3 temperature. Photomicrographs Figs. 47 and 48 show a more normal structure after annealing at 1600 deg. fahr.

That this steel was brittle is shown in its manner of fracture. A ductile material will distort by gliding along slip planes; a brittle metal will crack along a plane of stress. Photomicrograph



FIG. 60 PHOTOMICROGRAPH AT 1000 DIAMETERS OF INTERCRYSTALLINE CORROSION IN THE PEARLITE CONSTITUENT IN A CRACKED BOILER BLOW-OFF PAD



FIG. 61 CRACKS IN A BOILER BLOW-OFF PAD AT 100 DIAMETERS—THIS TYPE OF CRACK IS FREQUENTLY MET—THE CRYSTAL STRUCTURE OF THE METAL AND WIDTH OF THE CRACK MAKE IT DIFFICULT TO DETERMINE WHETHER OR NOT IT IS DUE TO CAUSTIC EMBRITTLEMENT

Fig. 49 shows a single crystal which contains within its boundaries short cracks running in various directions. Higher magnifica-

tions reveal faint boundary lines as though this crystal had once been made up of several small ones and was therefore the result of a recrystallization process. Photomicrographs Figs. 50 and 51 at 2000 and 4000 diameters respectively show a crack running into or from a sulphide inclusion. It is very doubtful whether the presence of even this large number of inclusions contributed to the failure of the metal. Cracks appearing in the steel are transcrystalline and run through inclusions only if the inclusion happens to be in its path. This is shown at 4000 diameters in photomicrograph Fig. 52.

Two Failures by Stress and Corrosion. Some rivets taken from service were examined for possible cracking. Photomicrograph Fig. 53 shows at 100 diameters two cracks due to corrosion which occurred at the junction of the rivet head and shank. In the area



FIG. 62 ANOTHER CRACK FROM A BOILER BLOW-OFF PAD AT 100 DIAMETERS

of metal between the two large fissures a small transcrystalline crack proceeding through undeformed crystals can be seen. A similar appearance was presented by the rivet shown at 100 diameters in photomicrograph Fig. 54. In this case, however, the crack is filled with the products of corrosion. The Jevons modification of the Fry etch showed that cracking had occurred in the area of highest local stress. Corrosion not directed by stress will usually proceed uniformly rather than by penetrating the metal in the form of cracks. Stressed areas carry a different electrolytic potential than unstressed areas.

A boiler tube started to leak at a point about an inch below the tube sheet. Examination showed that the tool used to expand the tube in place had been inserted carelessly so that a gouge was cut on the inner surface of the tube. Corrosion started at that point and gradually proceeded through the tube until leakage occurred. Photomicrograph Fig. 55 at 100 diameters shows a section of the crack. The black area represents the crack; the dark gray, products of corrosion; and the remainder is naturally the steel.

Slag, Forging Shuts, and Seams. Another source of failures in boiler materials is due to slag inclusions, forging shuts, and seams. Failure can readily occur due to mechanical causes under these conditions. Such a failure is shown in photomicrograph Fig. 56 at 100 diameters. These types of failure are generally regular in direction.

FAILURE THROUGH INTERCRYSTALLINE CRACKING

Parr and Straub[17] have described a phenomenon in boiler materials which they identify as caustic embrittlement. An accurate description of the effect they produced in the laboratory would be intercrystalline cracking in steel due to local tensile stresses in excess of the elastic limit, and to contact with solutions of caustic soda whose concentrations are in excess of 100 grams per liter. In very carefully controlled experiments they showed that, if the above conditions were met, intercrystalline cracking would take place regardless of the quality or previous thermal treatment of the steel. They pointed out the similarity between

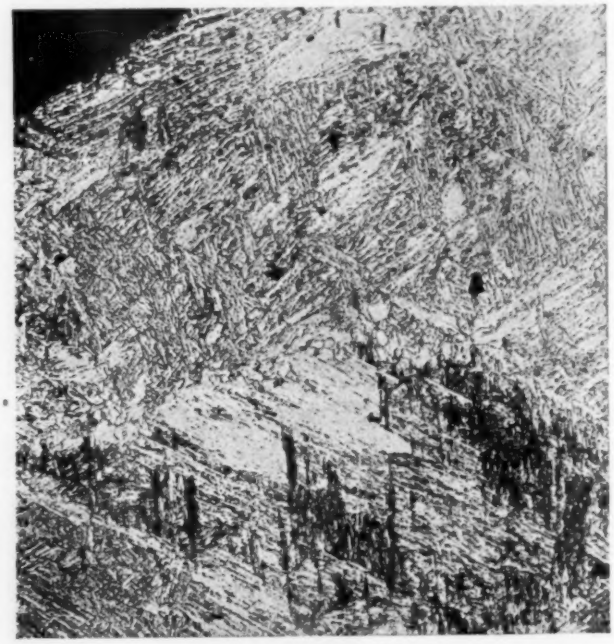


FIG. 63 TYPICAL SECTION AT 100 DIAMETERS FROM A TUBE WHICH BURST DUE TO OVERHEATING

these cracks and those found in boiler materials; that due to capillary action, high concentrations of sodium hydroxide were not improbable in the seams of the boilers, and that stresses near the rivets were probably quite high.

Photomicrographs Figs. 57 and 58 show at 100 and 1000 diameters a typical intercrystalline crack in a boiler tube. The course of the crack in the ferrite is shown at 4000 diameters in photomicrograph Fig. 59. Photomicrograph Fig. 60 taken at 1000 diameters of a cracked blow-off pad shows corrosion taking place in the pearlitic constituent. These failures have been considered due probably to caustic embrittlement. Many boiler failures, however, show cracks such as those presented at 100 diameters in photomicrographs Figs. 61 and 62 which were taken of a cracked blow-off pad. The nature of the failure is not clearly defined because of the peculiar grain structure, and because the cracking takes place both through and around the crystals.

FAILURE BY OVERHEATING AND BURSTING

An entirely different type of structure is exhibited by boiler tubes which have burst in an explosion. Sometimes material lodges in the tube and prevents the circulation of the water or scale formation permits overheating. As the temperature of the

tube increases the tensile strength of the metal diminishes. A point is reached where the pressure exerts stresses in excess of the yield point and failure takes place with explosive violence. Photomicrograph Fig. 63 shows the structure of the metal. It apparently had been heated above 1600 deg. fahr., and was quenched by the rush of steam around it.

Conclusion

The characteristic appearance of failures in boiler metals due to various common causes has been discussed. Bending, compression, tension, and impact according to the conditions described in this paper all produce fractures which are predominantly through the grain. Failure in such cases is accompanied by deformation of the crystals. Impact failures under some conditions may be produced with very little deformation. Fatigue cracks are always transcrystalline without deformation of the grain. Caustic embrittlement, as found in metal which has been in service in alkaline liquors, produces intercrystalline cracks. The actual mechanism of this phenomenon is not completely understood although it would seem that it is a result of the embrittling action which occluded hydrogen exerts upon the grain boundaries of iron. The causes of many actual failures of boiler material can be more or less satisfactorily traced by comparison with fractures produced in the laboratory.

Acknowledgments

This work was done at the suggestion and with the financial assistance of The Detroit Edison Company. Special acknowledgments are due C. F. Hirshfeld, Chief of Research of that company, and J. H. Walker, Assistant Chief of Research, for their friendly interest and their helpful and constructive suggestions.

Discussion

F. G. Straub.[35] In reading this most interesting paper one is impressed with the fact that much time must have been spent by the authors in preparing the material. They are to be congratulated upon their technique in preparing their specimens for micro-examination. The micrographs which they have shown illustrate the points they emphasize in an excellent manner.

This paper should help clear up the confusion which must have resulted from the publication of recent articles which have attempted to place the responsibility of most boiler-metal failures on corrosion fatigue. The authors show very clearly the different types of failure which result from stress, stress and corrosion, and so-called caustic embrittlement.

The writer is of course primarily interested in the intercrystalline cracking which they describe. The results of the work at the University of Illinois agree in almost every detail with the results of the work reported.

Some engineers have the impression that we ascribe all cases of boiler-metal failure to embrittlement caused by caustic. In fact we have, like the authors of this paper, found various types of failures brought about by vari-

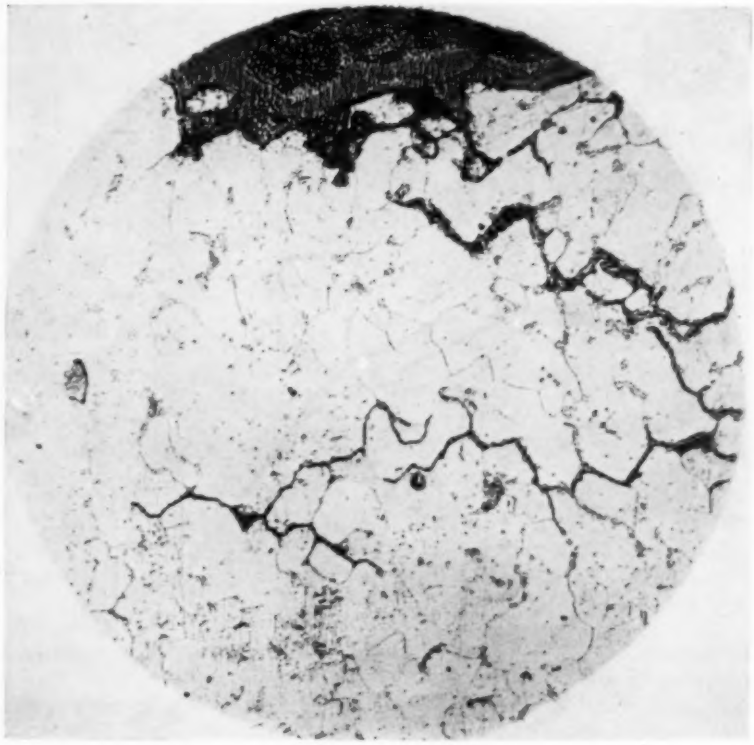


Fig. 64 Intercrystalline Cracks in Superheater Header. × 250

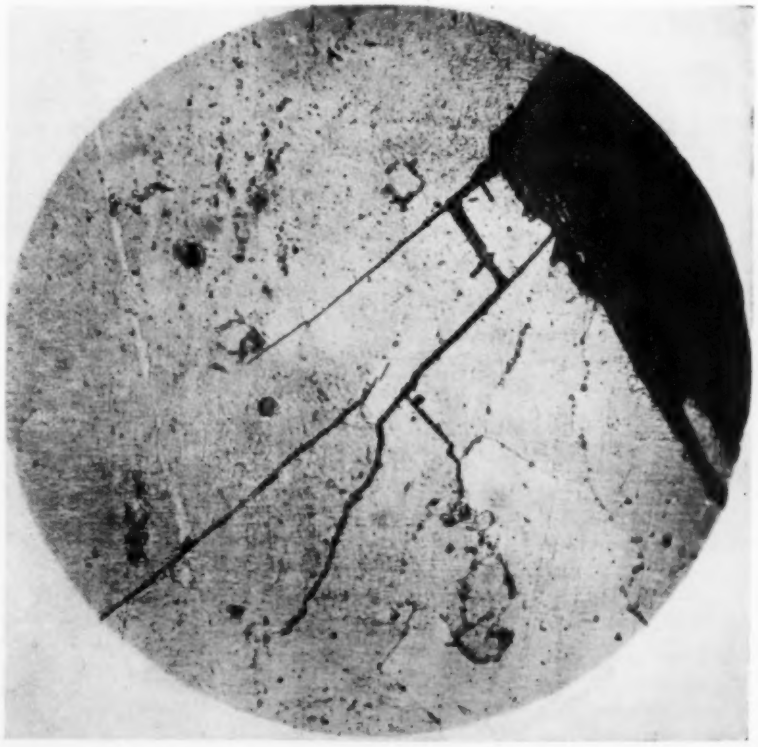


Fig. 65 Cracks in Same Header as Shown in Fig. 64, but Transcrystalline and Following Nitride Needles. × 250

[35] University of Illinois.

ous causes. The writer would like to discuss a few cases of failure which proved very interesting and almost misleading, and which illustrate that to arrive at definite conclusions as to causes one must make a thorough study of all contributing factors, in the same manner as the authors of this paper have done.

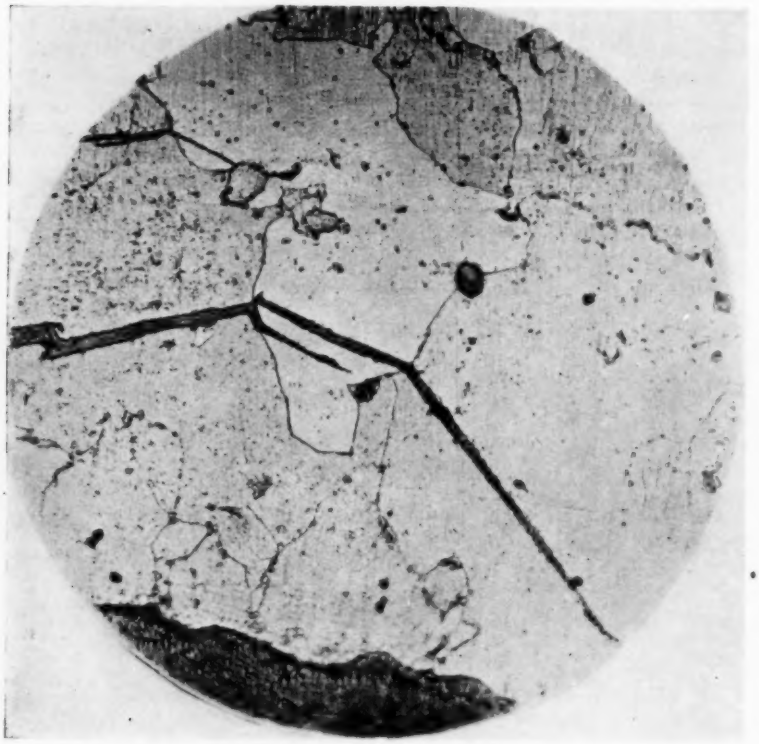


FIG. 66 CRACKS IN SAME HEADER AS SHOWN IN FIG. 64, BUT TRANSCRYSTALLINE AND FOLLOWING NITRIDE NEEDLES. × 250

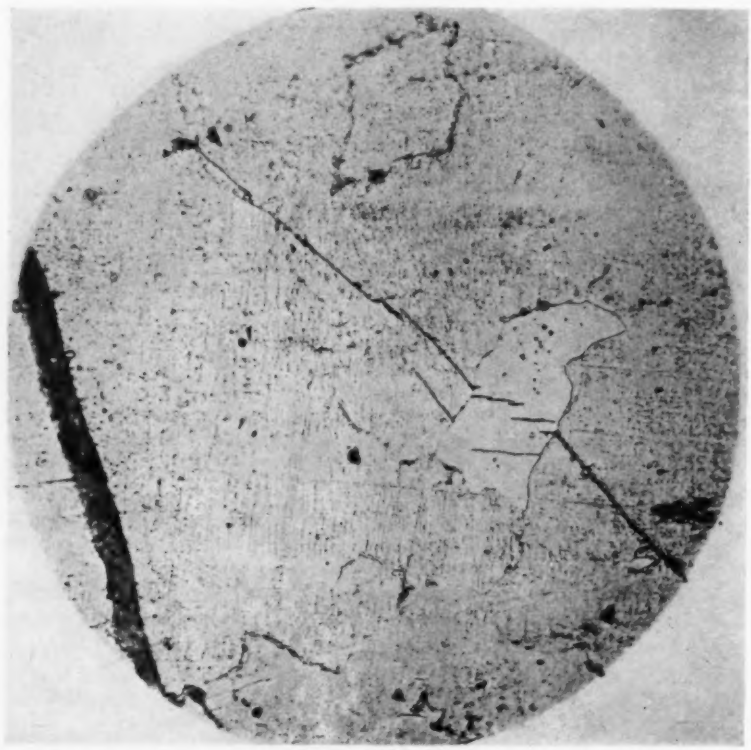


FIG. 67 CRACKS IN SAME HEADER AS SHOWN IN FIG. 64, BUT TRANSCRYSTALLINE AND FOLLOWING NITRIDE NEEDLES. × 250

In a large power plant using an older type of superheater, cracking was encountered in the superheater headers. The cracks were starting from the edge of the handholes. There had been evidence of high caustic concentration in the boilers and carryover to the superheaters. Microexamination of the polished specimen gave the typical intercrystalline crack shown in Fig. 64. The plant operators were ready to attribute their difficulty to caustic embrittlement. A more thorough study of the specimens cut from the cracked header indicated that caustic was not the cause of the cracking. The micrographs taken of these sections are shown in Figs. 64 to 67, inclusive. Further study showed that some of the rods supporting the header had broken, and the header without proper support was bending. The gases striking the metal were between 1200 and 1500 deg. fahr. The conditions of operation were such that this cracking could be readily explained on a basis of high temperature, stress, and high nitride.

The second case involved tube ends that were brittle and appeared to be laminated. The ends would break off. The fracture did not have a bright metallic luster. The tube could be separated into two layers running lengthwise of the tube. Microexamination of one of these tubes gave the result shown in Fig. 68. This indicates that the tubes were burned, apparently during manufacture.

The writer is extremely interested in the statement that the authors make relative to the fact that they have encountered typical intercrystalline cracking in steel which has been in contact with concentrated phosphate solutions. Our research has indicated that small amounts of phosphate tend to retard the embrittling action of caustic soda. It has been known that concentrated nitrate solutions produce this same type of cracking. We have found that small amounts of nitrate introduced in concentrated caustic solutions stop the action of the caustic.

H. C. DINGER.[36] The authors are to be congratulated upon this paper, which presents a mass of definite and useful information upon a very much discussed subject. In giving definite data as to the observed character of the cracks produced by definitely known causes much valuable information can be learned, and this information can then be used in investigation of actual cracks.

The paper, however, has not sufficiently considered the character of the cracks produced by cyclic stress accompanied by corrosion. This is the actual condition that is generally presented to the metal in the boiler. It also seems to be a fact that many if not most of the actual cracks in boilers show some evidence of both transcrystalline and intercrystalline cracking. The evidence presented by the authors seems to indicate that

[36] Captain, U. S. Navy, Officer in Charge, U. S. N. Experiment Station, Annapolis, Md. Mem. A.S.M.E.

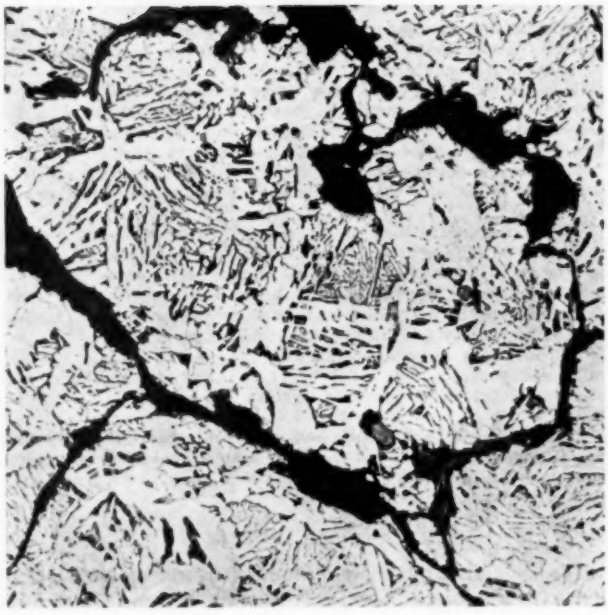


FIG. 68 MICROPHOTOGRAPH OF BURNED TUBE. × 100

true intercrystalline cracks are produced by stress in the presence of very high caustic concentration. In actual boilers these conditions are not produced either in the seams or elsewhere, but we do have variations of cyclic stress in the presence of variable corroding medium. We therefore have some kind of *corrosion fatigue* in all boilers, unless perhaps we could absolutely exclude oxygen from the water. There is some evidence that a caustic condition tends to accelerate the formation of intercrystalline cracks, but we also have many cracks partially intercrystalline where no caustic was present. We must then assume that corrosion tends to intercrystalline failure and stress to transcrystalline, or perhaps slow failure is pro-intercrystalline while rapid failure, due to higher stress, is more likely to be transcrystalline. The amount of oxygen and corroding salts present also, no doubt, accounts for variations.

The authors' deductions as to slippage, deformation, and final failure though crystal cracks present a logical picture of what may take place. This is in entire agreement with the phenomenon of corrosion fatigue. A real explanation of boiler cracks is really not possible unless the varying effects of corrosion accompanied by cyclic stress are con idered.

So-called "caustic embrittlement" appears to be but one of many variations of possible conditions. If we can keep the water in the least destructive condition (this is not yet fully known, but would seem to be deaerated water controlled at about nine pH), we can lessen the effect of corrosion. If we design our pressure parts to avoid stress concentration and cyclic stress as far as possible, we can lessen the destructive effect of cyclic stress. In view of the fact that neither of these can be very easily controlled where uncertainty exists, it is well to have an adequate margin of safety and in the design to consider that both corrosion and fatigue are likely to be present. One thing is fairly well established and it is that it is extremely desirable to keep the water away from the highly stressed metal of a riveted joint. Sealing of joints by inside calking or other means, such as by cements or lasting coatings, is extremely desirable. The coating of the water side of all joints by a durable protective film that will effectively keep the mixture of air and water away from the stressed surface of the metal ought to give some protection.

D. S. MCKINNEY AND MAX HECHT.[37] It may be considered that the contribution the authors have made to the literature on fractures in boiler metal will be of distinct benefit to workers closely associated with plant operation. The comparison of

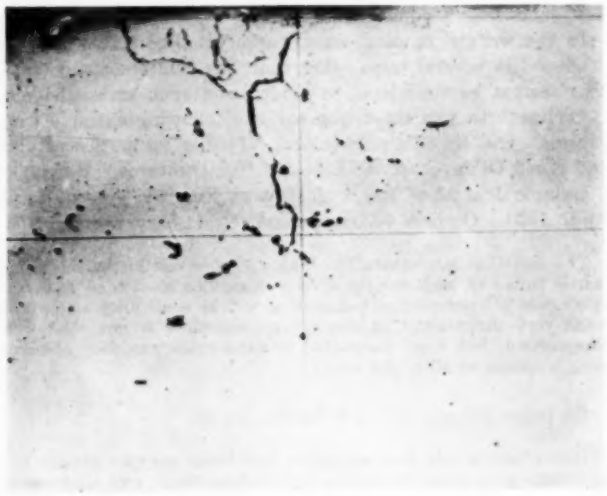


FIG. 69 SPECIMEN AS BURNT, PARTIALLY MELTED, AIR-COOLED, COLD-WORKED, UNETCHED. × 100

photomicrographs of boiler metal failures due to known causes with those found in boilers should produce more exact knowledge of the conditions accompanying boiler metal failure.

Under the heading of "Failures Through Intercrystalline Cracking," the authors refer to Figs. 57 to 62 inclusive. In the writers' opinion, Figs. 61 and 62 exhibit a structure characteristic of severe overheating.

In their own laboratory intercrystalline cracking has been pro-

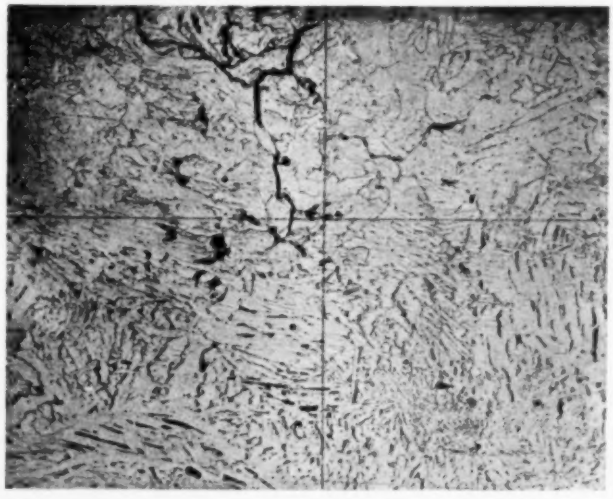


FIG. 70 SAME SPECIMEN, ETCHED WITH 4 PER CENT NITAL. × 100
(Cracks noted in Fig. 69 are intercrystalline.)

duced by abnormal heat-treating and cold-working new boiler metal (a rivet), carbon 0.2 per cent. The photomicrographs given in Figs. 69 and 70 show the cracks before and after etching.

The absence of evidence of overheating or cold-working cannot be taken as conclusive evidence that this abnormal treatment

[37] Duquesne Light Company.

has not been given to the steel, as these evidences may be removed by annealing. However, cracks once formed cannot be healed by annealing.

Freeman and Quick have shown that intercrystalline cracking is found in steel of varying carbon content, including Armco iron, exhibiting "secondary brittleness."

In the writers' opinion, since intercrystalline cracks may be produced in several ways other than "caustic embrittlement," they cannot be considered as evidence of such embrittlement.

Pertinent to this discussion are the following extracts from Freeman and Quick's publication, "Tensile Properties of Rail and Some Other Steels at Elevated Temperatures," Bureau of Standards Journal of Research, Report No. 164, Vol. 4, No. 4, April, 1930. On page 573, it is stated:

> The fact that intercrystalline failure is observed in the secondary brittle range of such widely diverse materials as Armco iron, rail steels, and a 1 per cent carbon steel as well as some alloy steels indicates very definitely that secondary brittleness is not related to composition, but must be related to some grain boundary phenomenon common to all of the steels.

On pages 590 and 591, it is further stated:

> It has been shown that secondary brittleness may be present in a relatively pure iron, rail steels, high-carbon steels, and alloy steels. The low ductility has been shown to be related apparently to a grain boundary condition of the steel as was indicated by the marked intercrystalline shattering found in specimens tested in the secondary brittle range. The cause then must be related to one or more of the following: some structural condition, an element, or nonmetallic inclusions which may be present in steels of widely varying composition. The influence of free cementite has been suggested. The fact that annealing below the Ac_1 transformation affects the degree of secondary brittleness indicates that it may be related to a constituent capable of solid solubility or precipitation at temperatures below the A_1 transformation.
>
> A study has been made of the tensile properties of rail steels and some other steels at elevated temperatures. Special study was made of the tensile properties in the approximate temperature range of 400 to 700 deg. cent. It has been found that the ductility of all rail steels, as measured by percentage elongation, and reduction of area, decreased with increase in temperature over a portion of this temperature range. In some instances the decrease in ductility is very marked, the elongation and reduction of area values becoming less between approximately 500 to 650 deg. cent. than at normal atmospheric temperatures. The same phenomena have been found to exist in Armco iron, a 1 per cent carbon steel, and some alloy steels. This range of temperature in which low ductility becomes manifest has been termed the "secondary brittle" range to distinguish it from the well-known "blue brittle" range occurring at 200 to 300 deg. cent. and the "hot-short" range which is usually associated with the A_3 transformation and high sulphur content. The phenomenon has been termed "secondary brittleness."
>
> The degree of secondary brittleness varies markedly between individual heats, and the temperature of the maximum effect also varies.
>
> The data indicate that the secondary brittleness is not confined to individual rails or ingots, but is apparently a property of the heat as a whole. It has been shown that the degree of secondary brittleness is decreased by annealing at a temperature (700 deg. cent.) slightly below the Ac_1 transformation as well as at the usual annealing temperature above the Ac_3 transformation.
>
> The fact that secondary brittleness is found in other steels than rail steels suggests that other types of unexplained failures, such as internal cracks in ingots or forgings, may be due to secondary brittleness.
>
> The data indicate that equalization of temperature and slow cooling of rails and other steel shapes through their secondary brittle range are desirable procedures to follow in order to prevent formation of shatter cracks.

Authors' Closure

The point raised by Professor Straub on intercrystalline cracking due to overheating or burning is very interesting. Overheating of low-carbon steel, however, cannot be brought about at the temperature of the furnace gases, given as 1200 to 1500 deg. fahr. A temperature of between 2200 and 2400 deg. fahr. is necessary.

The authors were not familiar with "secondary brittleness" which Mr. Hecht mentioned, and have found his comments very interesting.

The authors appreciate the importance of corrosion fatigue as a probable mechanism of boiler cracking, as Captain Dinger points out. Before submitting the paper, they communicated with Mr. D. J. McAdam, Jr., who has done a great deal of work in this line. The authors requested a photomicrograph on a specimen having an intercrystalline crack due to corrosion fatigue. His reply was that up to the present he had no photomicrographs or specimens with intercrystalline cracks due to corrosion fatigue. For this reason the authors did not discuss this phase of failure at greater length.

FSP-53-17

Frictional Resistance and Flexibility of Seamless-Tube Fittings Used in Pipe Welding

By SABIN CROCKER[1] and ARTHUR McCUTCHAN,[2] DETROIT, MICH.

The trend toward the use of welded piping has led to the development of seamless-tube fittings. These fittings are a decided departure from both the cast fittings and long-radius pipe bends heretofore used, and their introduction has occasioned considerable speculation as to their relative merits and correct application. This paper is confined to a discussion of: (a) the relative friction loss in elbows and bends of different radii, and (b) the relative flexibility of pipe lines made up with long-radius bends as distinguished from cast fittings and from seamless-tube fittings having a radius of $1^1/_2$ pipe diameters. The authors have compiled and analyzed the available data on friction loss in elbows and bends, and present a detailed comparison of the flexibility of piping made up with seamless-tube fittings, long-radius bends, and cast fittings. Seamless-tube fittings (R/d — $1^1/_2$) are found to compare to advantage with the older types of construction in both phases considered.

THE application of fusion welding to the fabrication and erection of steel pipe lines has progressed rapidly in recent years both in the extent of use and in new types of construction made possible thereby. A decided improvement in welded construction has been effected through the development of ready-made elbows and returns of a radius comparable to long-radius cast fittings and forged from seamless tubing. For convenience these will be referred to in this paper as seamless-tube fittings.[3] Such fittings are made at the present time in sizes from 1 in. to 18 in., inclusive, in standard iron-pipe size, as shown in Fig. 1, and from sizes $1^1/_2$ in. to 12 in., inclusive, in extra strong. The three standard types, i.e., 45- and 90-deg. ells and the 180-deg. returns, are made, and all fittings are scarfed 45 deg. to facilitate their application. The usual center-line radius of such fittings is $1^1/_2$ internal pipe diameters, although other radii can be obtained.



S. CROCKER



A. McCUTCHAN

Before proceeding with a presentation of the friction loss and flexibility data which constitute the principal subjects of this paper,[4] a few photographs will be introduced to illustrate how seamless-tube fittings are applied in the construction of welded pipe lines. They are adapted readily to the fabrication of branch connections equivalent to tees and the like, as shown in Fig. 3. Where flanges are required, the seamless-tube fitting can be welded on, as shown in Fig. 4(*a*), which compares to advantage with the equivalent piece built up with a screwed elbow and screwed flange, Fig. 4(*b*).

Where welded construction is used, seamless-tube fittings have a decided advantage over the mitered square-corner job made in the field as regards strength, reduced friction loss, and appearance. Typical examples of mitered elbows are shown for comparison with a seamless-tube elbow assembly in Fig. 5. Examination of Fig. 5 indicates that decidedly less field labor is required in cutting and welding in the case of the seamless-tube elbow.

That the assembled product made up with seamless-tube fittings makes a good-looking job and provides easy turns for fluid flow is illustrated in Figs. 6 and 7. The noticeable absence of flanges adds to the appearance of the piping ensemble, and has a decided advantage in the application of covering. The general effect is entirely different from anything that can be obtained with mitered elbows, and should do much toward developing a preference for welding over older types of construction.

It is the purpose of this paper to discuss in some detail the use of seamless-tube fittings from a piping-design standpoint, comparing them with mitered fittings, cast fittings, and long-

[1] Engineer, Engineering Division, The Detroit Edison Company. Formerly Piping Engineer, Drafting and Surveying Bureau of same company. Mem. A.S.M.E. Mr. Crocker was graduated in 1915 from the University of Michigan, with degree of B.M.E., and has been employed since by The Detroit Edison Company with the exception of two years (1917–1919) spent with the U. S. Naval Reserve Force and the Emergency Fleet Corporation in engineering duties.

He has taken an active part in A.S.A. piping-standardization projects and in A.S.T.M. committees, writing specifications for piping materials.

He is co-author with S. S. Sanford of paper on "The Elasticity of Pipe Bends" presented at the Annual Meeting of the A.S.M.E. in 1922; and also co-author with J. H. Walker of the "Piping Handbook."

[2] Engineer, Engineering Division of The Detroit Edison Company. Mr. McCutchan was educated at Monmouth (Illinois) College, and the University of Michigan, from which latter institution he received the degree of B.S. in Mechanical Engineering in 1926. Since graduation he has been engaged in mechanical-engineering studies with The Detroit Edison Company.

[3] For a more detailed description of the product and the method of manufacture, see "The Economics and Advantages of Pipe Welding With Stock Fittings," by Willard P. Curley, *Journal of the American Welding Society*, April, 1930, pp. 42 to 51.

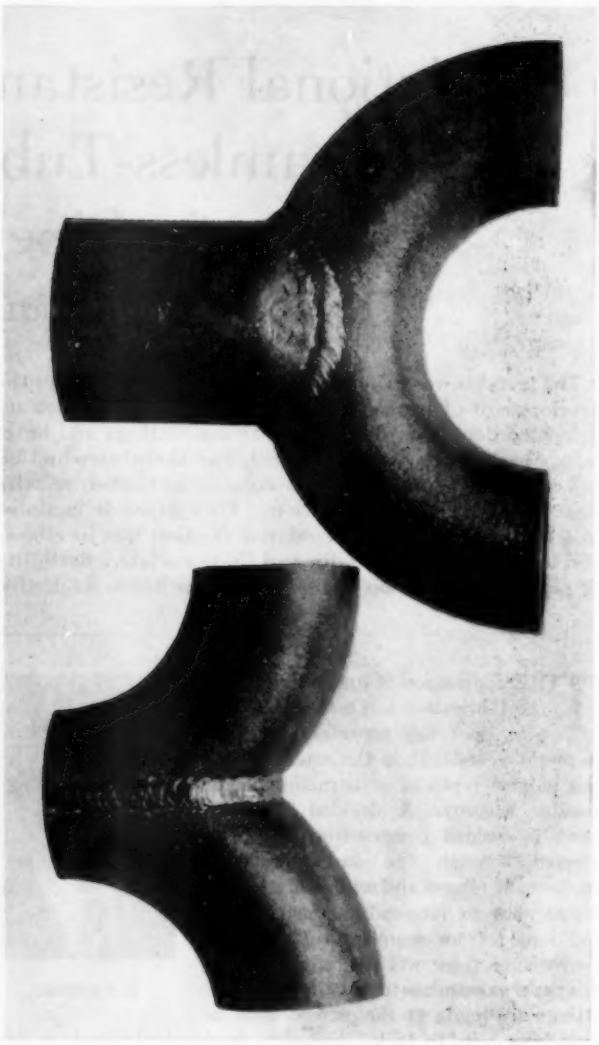
Presented at the Annual Meeting, New York, N. Y., December 1 to 5, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[4] Prepared with the cooperation of Tube Turns, Inc., who manufacture in the U. S. A. the type of seamless-tube fittings referred to.



FIG. 1 SEAMLESS-TUBE FITTINGS SHOWING ELBOW, 45-DEG. ELBOW, AND RETURN



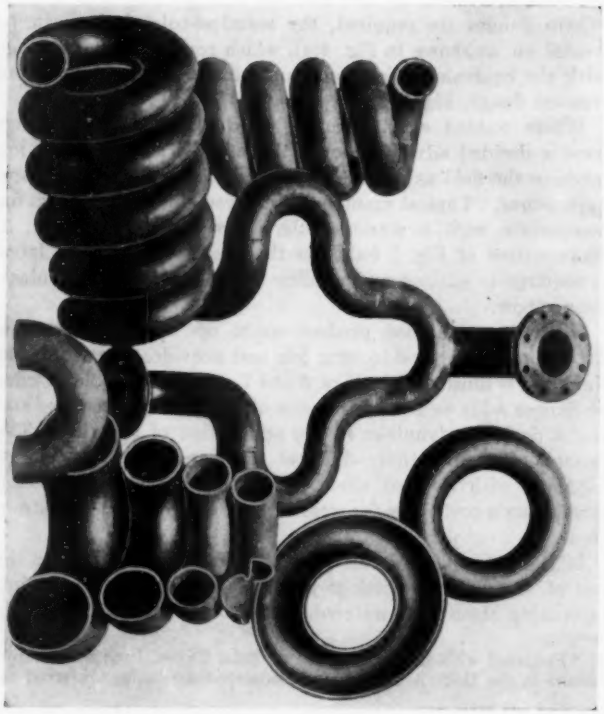


FIG. 2 CONTINUOUS PIPE COILS (ONE-PIECE), ALSO WELDED ASSEMBLIES MADE BY USING SEAMLESS-TUBE FITTINGS

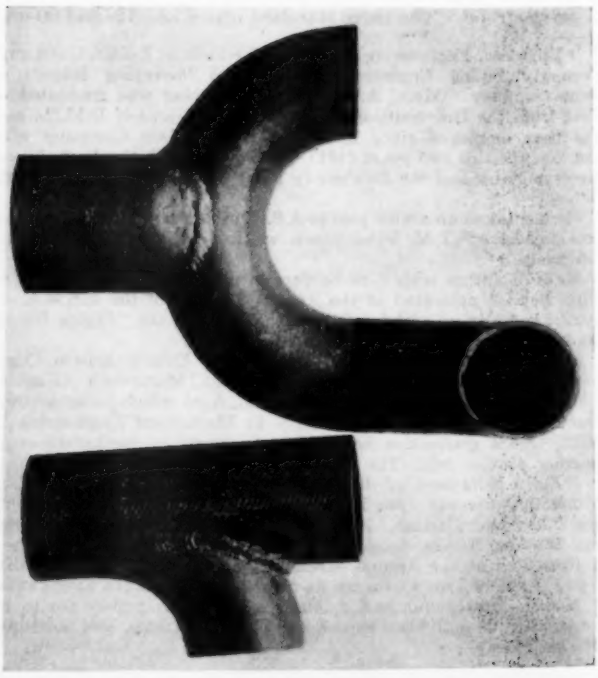


FIG. 3 TEES AND ASSEMBLED FITTINGS MADE BY WELDING SEAMLESS-TUBE FITTINGS

radius pipe bends as regards pressure drop and flexibility. In the latter connection a comparison will be made of the elastic properties of piping made up with long-radius pipe bends as distinguished from seamless-tube fittings having a radius of $1^1/_2$ pipe diameters.

I—PRESSURE DROP IN ELBOWS AND BENDS

A misconception is more or less prevalent to the effect that short-radius bends and fittings cause appreciably greater pressure drop than do long-radius bends. This idea is not borne out by published results, provided the mean radius of curvature is from $1^1/_2$ to 2 times the inside pipe diameter. More data on the frictional resistance in bends are available for air and water than for other fluids. Some of the leading references on this subject are quoted in succeeding paragraphs.

Air. Fig. 8, reproduced from "Fan Engineering" of the Buffalo Forge Co., is based on data presented in a paper by Frank L. Busey on "Loss of Pressure Due to Elbows in the Transmission of Air Through Pipes or Ducts," A.S.H. and V.E. Transactions 1913, page 336. The pressure drop in percentage of a velocity head is given for elbows of different radii, the center-line radius being expressed in percentage of the pipe diameter. It is evident from this diagram that very good results are obtained with $R/d = 1^1/_2$, and that almost nothing is to be gained by making R/d greater than 2.

Further data on the flow of air through elbows of different radii are given in an article by William D. Edwards entitled "Pressure Losses in Air Ducts," published in *The Heating and Ventilating Magazine*, Feb., 1928, page 69. Fig. 9, which is reproduced from that article, shows the equivalent resistance of different-radius elbows and bends expressed as the product of a bend constant and pipe diameter. The bend constants serve as indices of the pressure drop through each type of bend, and are plotted in Fig. 10 as Resistance Factor vs. Mean Radius of Bend. Inspection of Fig. 10 indicates that here again no appreciable reduction in pressure drop is shown for a mean radius of bend greater than $1^1/_2$ times the inside pipe diameter. In this instance the nearest approach to a mitered elbow is given a friction loss of between 11 and 12 times that of a bend having a radius of $1^1/_2$ pipe diameters.

Water. That similar relations of pressure drop to bend radius exist for other fluids is shown by the curves of Fig. 11 which give fractional velocity heads lost in screwed elbows and in water-pipe bends of different radii. These curves were plotted from data taken from several sources[a] as indicated. Deduction has been made in each case for the friction loss which would have occurred for a corresponding length of travel through straight pipe, so that the results represent loss which is directly chargeable to the type of bend or fitting. The curves show a rapid decrease in elbow friction as R/d increases from less than one until it reaches $1^1/_2$ or 2. For values of from 2 to 4 the decrease in elbow friction diminishes slightly, and for values of R/d above 5, with the exception of the Weisbach curve, there is a rapid increase in bend friction. A similar tendency to increase at values of R/d greater than $2^1/_2$ was observed in large-diameter pipes by Williams, Hubbel, and Fenkell,[6] who found a point of minimum friction at R/d equal to $2^1/_2$. For values of R/d exceeding 5 or 6 several of these investigators observed fluctuating friction losses which did not appear to bear a definite relation to bend radius. This portion of Fig. 11 has been drawn in as smooth curves which the authors believe should represent average results. The authors of the papers from which these data were taken attributed the increase in friction loss at values of R/d above 4 or 5, to the greater length of pipe travel over which flow disturbance persisted, acting to counterbalance the lesser degree of disturbance produced in longer-radius bends. Or, in other words, they took the view that a long-radius bend tends to maintain the disturbance and attendant high frictional loss over a greater length of travel than does a short-radius bend, hence producing a greater total loss although the loss per unit length of travel is not so great. This fits in with the theory that elbow or bend loss is caused in part by increased turbulence of flow in the bend and beyond, as well as by shock resistance to change in direction. In short-radius fittings or 90-deg. mitered elbows, the predominating loss is that due to shock, while in long-radius bends, where the change in direction is more gradual, the loss is due principally to increased friction. The foregoing leads to the conclusion that medium-radius bends of smooth curvature and full cross-section are preferable from a pressure-drop standpoint to either mitered elbows or long-radius bends.

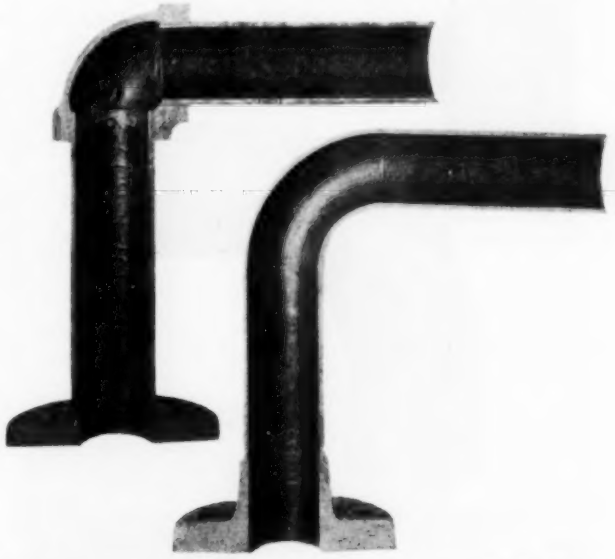


FIG. 4(*a*) FLANGED ASSEMBLY MADE WITH SEAMLESS-TUBE ELBOW AND WELDING NECK FLANGE

FIG. 4(*b*) FLANGED ASSEMBLY MADE WITH SCREWED ELBOW AND SCREWED FLANGE

It will be noted that three of the curves plotted in Fig. 11 refer to a flow velocity of 5 ft. per sec. These writers reported similar observations at other flow velocities ranging from 2 to 15 ft. per sec. in the same and other sizes of pipes, the general shape of the curves being similar, although the fractional velocity heads lost in bend friction varied with the velocity and pipe size. It seems reasonable to suppose that with sufficient data available it would be possible to formulate an equation involving radius of curvature and diameter of pipe, and velocity, density, and viscosity of the fluid, which would be more widely

[a] These data were published in the following references:

a "Mechanics of Engineering," by Dr. Phil. Julius Weisbach, (Coxe's translation), page 895.

b "An Experimental Study of the Flow of Water in Pipes," by Augustus V. Saph and Ernest H. Schoder, Trans. A.S.C.E. paper 964, vol. 51, 1903. See also "Curve Resistance in Water Pipes," by Ernest H. Schoder, Trans. A.S.C.E., March to June, 1909, vol. 62.

c G. J. Davis, Trans. A.S.C.E., vol. 62, 1909, pp. 97–112.

d A. W. Brightmore, *Proc. Inst. of Civil Engrs.*, vol. 169, pp. 315.

e "Selection of Elbows Can Regulate Pipe-Line Pumping Costs," by Prof. J. H. Pound, *Oil Field Engineering*, Dec. 1, 1929.

[6] Trans. A.S.C.E., vol. 47, 1902, pp. 1–335.

applicable to the flow of water and other fluids in bends.[7] The curves presented in Fig. 11 are offered merely as typical of conditions in bends of different curvature with a view to illustrating the characteristic tendency for variance in pressure drop with R/d.

Other Fluids. The above data apply to the flow of air and water as noted. The authors have been unable to locate any results of equivalent tests covering the flow of steam and other fluids through bends of different radii, but it is reasonable to

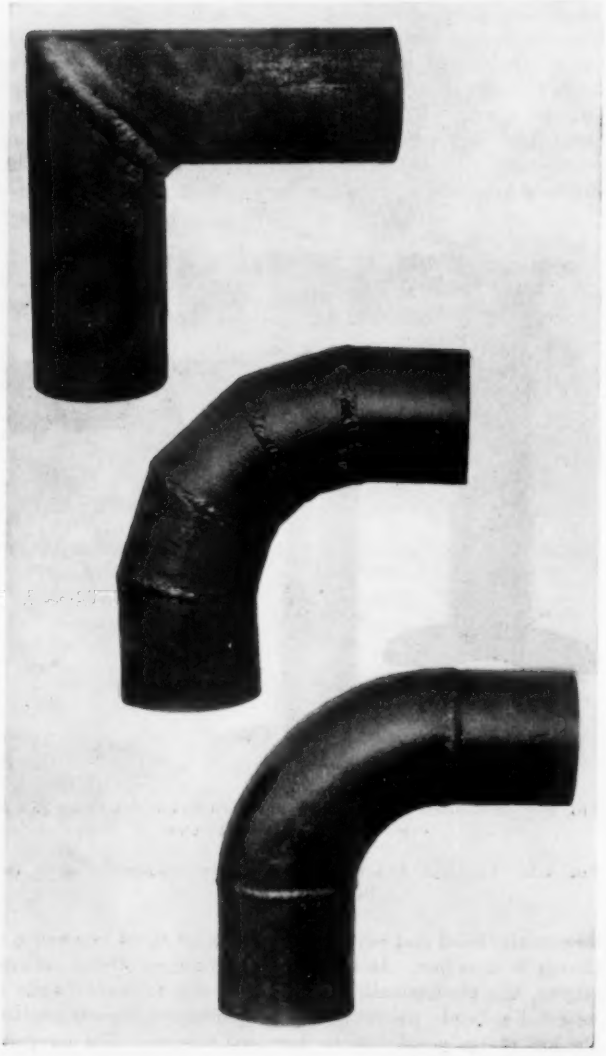


FIG. 5 COMPARISON OF TWO TYPES OF MITERED ELBOWS WITH A SEAMLESS-TUBE ELBOW

assume that such fluids exhibit similar characteristics. In an article by M. Hottinger in *Gesundheits-Ingenieur*, vol. 51, 1928, page 723, dealing with the flow of water through bends and fittings, the following statement is made: "The tests made in the Laboratory for Heating and Ventilating at the Polytechnic Institute, Berlin, concerning pressure losses in different elements

[7] See "Piping Handbook" (McGraw-Hill Book Co., Inc.), by Walker and Crocker, pp. 84 to 99, for description of a rational formula applying to the flow of fluids in general. The authors believe it would be possible to introduce the effect of curvature into such formulas, thus obtaining a widely applicable set-up for dealing with bend friction.



FIG. 6 WELDED PIPE CONSTRUCTION IN A BUILDING, USING TWO SEAMLESS-TUBE ELBOWS



FIG. 7 WELDED PIPE CONSTRUCTION IN A TUNNEL, USING SEAMLESS-TUBE ELBOWS

of a distributing system, gave results showing that the coefficients of resistance are the same for pipes carrying water (cold or hot)[8] and steam of low or high pressure. (See Bulletin No. 5–1913 of this Laboratory.) Therefore my experiments on bends and fittings were made only with cold water from 48 to 54 deg. fahr." While the above statement is somewhat broad

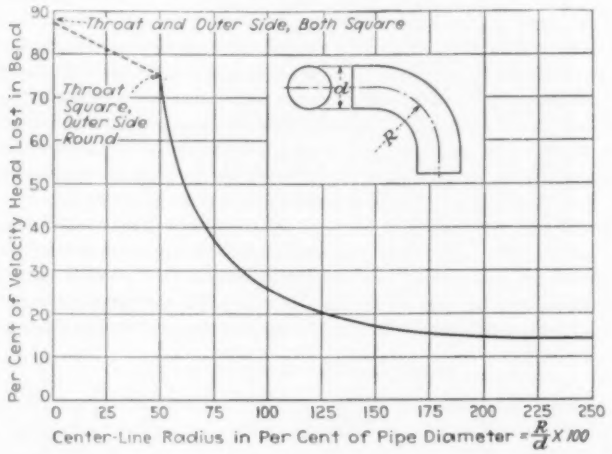


FIG. 8 HEAD LOST IN 90-DEG. BENDS CARRYING AIR

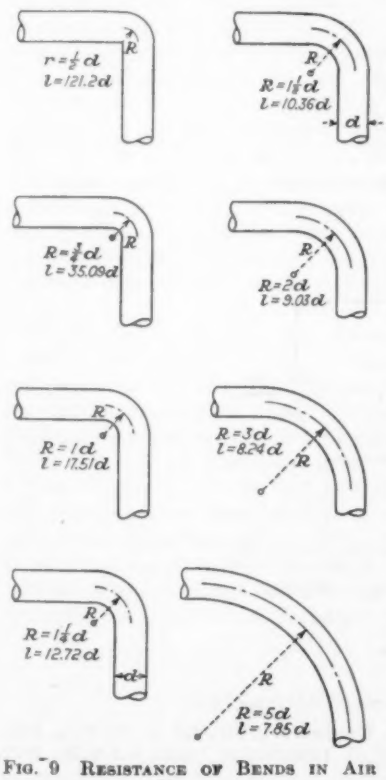


FIG. 9 RESISTANCE OF BENDS IN AIR PIPING

(l = equivalent length of straight pipe expressed in diameters.)

to accept literally, it does tend to support the idea that the relation of bend friction to R/d for steam would plot into a shape similar to those given in Fig. 11 for water.

RADII OF AMERICAN STANDARD CAST-IRON AND STEEL ELBOWS

In view of the relatively high frictional resistance observed in elbows where R/d is less than $1^1/_2$ it is of interest to note the approximate values of this ratio for different classes of American Standard screwed and flanged elbows set down in Table 1. These values were computed on the assumptions that (*a*) the bore of the fitting agrees with the nominal size up to 12 in., and with the corresponding pipe bore for sizes 14 in. and larger; (*b*) the curvature is a circle whose center is determined by lines passing through the back of the flange or through the back of the band in the case of screwed fittings. While these assumptions are not exact in all cases, it is deemed that they approximate actual conditions well enough for the present comparison, as these details will vary to some extent with different manufacturers. Examination of Table 1 indicates that all the screwed fittings lie in the high-pressure-drop range of R/d; that standard

[8] These tests were made at temperatures ranging from 59 to 194 deg. fahr.

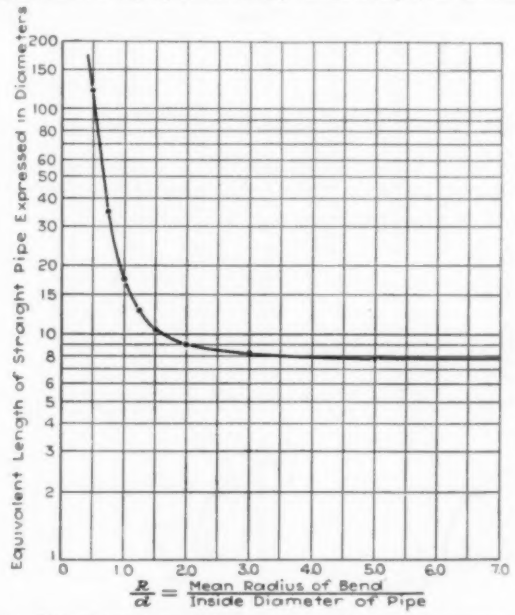


FIG. 10 RESISTANCE OF BENDS IN AIR PIPING—PLOTTED FROM DATA IN FIG. 9

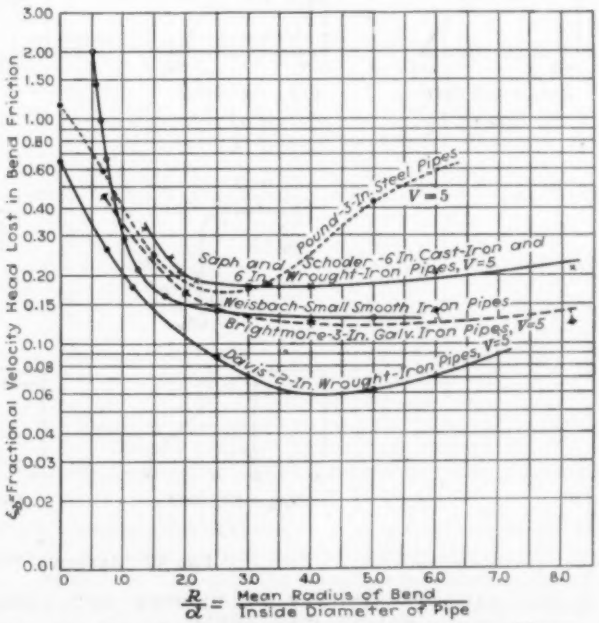


FIG. 11 HEAD LOST IN 90-DEG. BENDS CARRYING WATER

($h_b = \epsilon_b V^2/2g$, where h_b = head lost, ft. of water; ϵ_b = fractional velocity head lost in bend friction above loss in corresponding length of straight pipe; V = water velocity, ft. per sec.; and g = 32.2 ft. per sec. per sec.)

flanged fittings for pressures up to 600 lb. fall in the low- or moderate-pressure-drop range for sizes below 6 in. and in the high-pressure-drop range for larger sizes; that long-radius flanged fittings come almost entirely in the low- and moderate-pressure-drop range. These observations are of interest for comparison with seamless-tube fittings having a uniform R/d ratio of $1^1/_2$, which gives a consistent pressure-drop relation approaching the flat region of the curves.

TABLE 1 RADII OF 90-DEG. ELBOWS EXPRESSED IN PIPE DIAMETERS[1]

(A.S.A. Standard Dimensions)

Nominal size	125-lb. C.I. screwed	150-lb. M.I. screwed	250-lb. C.I. screwed	125-lb. C.I. flanged	125-lb. C.I. flanged, long-radius	250-lb. C.I. flanged	250-lb. C.I. flanged, long-radius	150-lb. steel flanged	150-lb. steel flanged, long-radius	250-lb. steel flanged	250-lb. steel flanged, long-radius	400-lb. steel flanged	600-lb. steel flanged
1	0.9	0.9	0.9	3.1	4.5	3.3	4.3	...	...	3.3	...	3.3	3.3
1¼	0.8	0.8	0.8	2.6	4.0	2.8	3.8	...	...	2.8	...	2.8	2.8
1½	0.7	0.7	0.7	2.3	3.6	2.4	3.4	...	...	2.4	...	2.4	2.4
2	0.7	0.7	0.7	1.9	2.9	2.1	2.8	1.9	2.9	2.1	2.8	2.3	2.3
2½	0.7	0.7	0.7	1.7	2.5	1.8	2.4	1.7	2.5	1.8	2.4	2.1	2.1
3	0.7	0.7	0.7	1.6	2.3	1.6	2.2	1.6	2.3	1.6	2.2	1.8	1.8
3½	0.7	0.7	0.7	1.5	2.2	1.5	2.1	1.5	2.2	1.5	2.1	1.7	1.7
4	0.7	0.7	0.6	1.4	2.0	1 4	1.9	1.4	2.0	1.4	1.9	1.4	1.7
5	0.7	0.7	0.6	1.3	1.9	1.3	1.8	1.3	1.9	1.3	1.8	1.3	1.6
6	0.6	0.6	0.6	1.2	1.8	1.2	1.7	1.2	1.8	1.2	1.7	1.2	1.5
8	0.6	0.6	0.6	1.0	1.6	1.0	1.5	1.0	1.6	1.0	1.5	1.0	1.3
10	0.6	0.6	0.6	0.9	1.5	1.0	1.5	0.9	1.5	1.0	1.5	1.0	1.3
12	0.6	0.6	0.6	0.9	1.5	0.9	1.4	0.9	1.5	0.9	1.4	0.9	1.2
14 O.D.	0.6	0.6	0.6	1.0	1.5	1.0	1.5	1.0	1.5	1.0	1.5	1.0	1.1
16 O.D.	0.6	0.6	0.6	0.9	1.5	0.9	1.4	0.9	1.5	0.9	1.4	0.9	1.1
18 O.D.	...	...	...	0.9	1.4	1.0	1.4	0.9	1.4	1.0	1.4	1.0	1.1
20 O.D.	...	...	...	0.9	1.4	0.9	1.4	0.9	1.4	0.9	1.4	0.9	1.1
24 O.D.	...	...	...	0.9	1.4	0.9	1.4	0.9	1.4	0.9	1.4	0.9	1.1
30 O.D.	...	...	...	0.8	1.4	0.8	1.3	...	...	...	...	...	...
36 O.D.	...	...	...	0.7	1.3	...	...	...	...	...	...	...	...
42 O.D.	...	...	...	0.7	1.3	...	...	...	...	...	...	...	...
48 O.D.	...	...	...	0.7	1.3	...	...	...	...	...	...	...	...

[1] Approximate values for R/d where R is mean radius of 90-deg. elbow and d is bore of fitting.

593 of the paper by Crocker and Sanford on "Elasticity of Pipe Bends," Trans. A.S.M.E., 1922. At that time the effect of flattening in the circular cross-section of a curved pipe during flexure was not as well understood as at present, and no attempt was made to take it into account in that comparison. The present investigation shows that such flattening of curved pipes during flexure has a decided effect on a comparison of this kind which cannot be ignored in computing either reactions or stresses.

A number of references are available for those who wish to familiarize themselves with the theory of flattening in the circular cross-section of a curved pipe during flexure.[9]

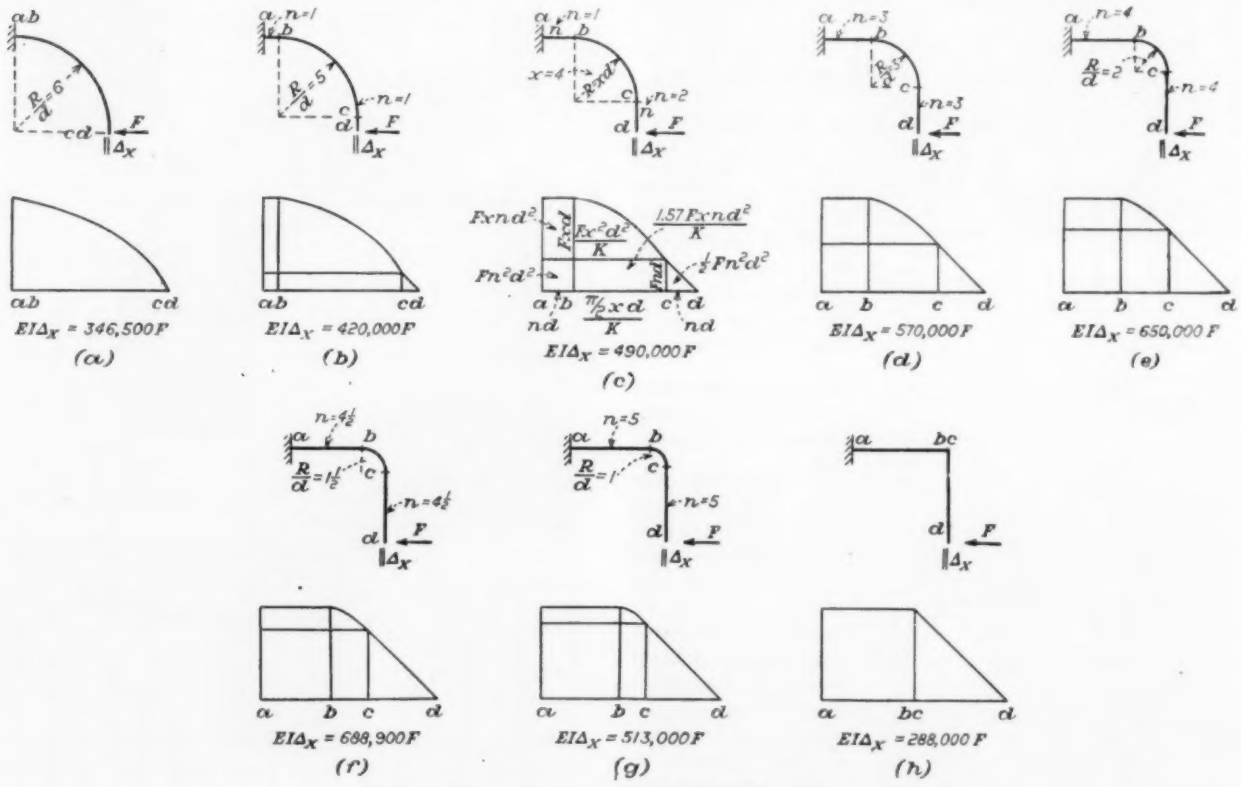


FIG. 12 RELATIVE FLEXIBILITY OF 90-DEG. TURNS

II—RELATIVE FLEXIBILITY OF ELBOWS AND LONG-RADIUS PIPE BENDS

In comparing the merits of different piping layouts applicable to any given job, question frequently is raised as to whether long-radius pipe bends or so-called "square bends" give the greater flexibility. In making such a comparison it is necessary to consider both major phases of flexibility, i.e. (*a*) the relative reactions set up at anchor points; and (*b*) the maximum stress produced at any point in the bend or pipe line. The maximum bending moment produced at any joint also is of interest, since the amount of bending moment obtaining has a bearing on the probability of keeping a joint tight. A comparison of reactions and stresses for four simple types of bends was made on page

[9] The principal references on this theory are:

a Paper by Dr. Th. Karman, Z.V.D.I., Bd. 55 (1911), p. 1889.

b Discussion by Dr. S. Timoshenko, Trans. A.S.M.E., 1922, p. 585.

c Papers by Prof. William Hovgaard, *Jl. Math. & Phys.* (M.I.T.), vol. vi, no. 2, 1926; vol. vii, nos. 3 and 4, 1928; vol. viii, no. 4, 1929. Also Proc. Nat. Acad. Sci., vol. 16, 1930; World Eng. Congress, Tokyo, 1929, "Deflections and Stresses in Pipe Bends;" Int. Cong. App. Mech., Stockholm, 1930, "Bending of Curved Pipe."

d "Stresses and Reactions in Expansion Pipe Bends," by A. M. Wahl, Trans. A.S.M.E., vol. 50, no. 15, p. 241, paper FSP-50-49, 1928.

e "Piping Handbook," by Walker and Crocker, McGraw-Hill Book Co., Inc., p. 550.

This theory has been well established by rigorous mathematical analysis and extensive tests on actual pipe bends. No derivation of this relation is included here since it is adequately covered in the references listed below. The terms and symbols used in applying the theory in this paper are in agreement with those given in reference 9e, and are defined for convenience in Appendix No. 3.

Several common forms of long-radius pipe bends have been selected as examples for comparison with their equivalent counterparts of the "square bend" variety as made up with seamless-tube fittings ($R/d = 1^1/_2$), American Standard flanged steel fittings and mitered elbows. Enough examples have been worked out to afford some degree of comparison and to indicate how the methods can be applied to other cases. Solution by the area-moment method described in the "Piping Handbook" by Walker and Crocker has been adopted as the one most universally applicable to problems of this sort. Complicated bends and pipe lines in three-dimensional space can be solved by area-moments, although such problems are too complicated for the successful application of analytical solutions.

Flexibility of 90-Deg. Turns

Elements Considered. The fundamental relations between displacement, reaction, and stress obtaining with a simple quarter-bend or 90-deg. turn are the basis of flexibility computations for practically all cases involving curved pipe. It is convenient, therefore, to take up first the case of a quarter-bend anchored at one end and having a force applied at the other end perpendicular to the axis of the pipe and lying in the plane of the bend as shown in Fig. 12. Ten-inch standard-weight pipe has been chosen for this comparison, using as the basis a quarter-bend having a radius of 6 nominal pipe diameters ($R/d = 6$). With this quarter-bend are compared other 90-deg. turns consisting of shorter-radius bends and fittings having sufficient tangent length on each end to give the same center-to-face dimension in each case. The other turns considered are made up with a seamless-tube fitting ($R/d = 1^1/_2$), a 250-lb. American Standard flanged steel elbow ($R/d = 1$), and a 90-deg. miter elbow.

Relation Between Deflection and Force. The relation between deflection and force is obtained by means of the area-moment diagrams shown in connection with Fig. 12. The general bending-moment diagram for a single force acting in the plane of the bend on a quarter-bend with tangents at each end is shown in Fig. 12(c). This diagram involves elements treated as cases II, III, V, and VII of the "Piping Handbook" (reference 9e). The rotations of the elements due to force F acting on the free end of the tangent length of pipe, Fig. 12(c), are represented by the following equations which are simply the areas of the bending-moment diagram to the left of each point indicated. (See Appendix No. 3 for definition of symbols.)

$$\psi_b = (F/EI)\ (xnd^2 + n^2d^2) \qquad \text{clockwise}$$

$$\psi_c = \psi_b + F\left(\frac{x^2d^2}{KEI} + \frac{1.57xnd^2}{KEI}\right) \qquad \text{clockwise}$$

In which $x = \dfrac{R}{d} = \dfrac{\text{mean radius of bend in inches}}{\text{nominal diameter of pipe in inches}}$

$n = \dfrac{l}{d} = \dfrac{\text{length of tangents in inches}}{\text{nominal diameter of pipe in inches}}$

K = rigidity multiplication factor which takes flattening of the cross-section of a curved pipe into account.

The deflections in the direction in which the force acts of the several parts of the bend are given by

$$EI\,\Delta x_c = \Sigma\psi_b xd \qquad \text{left}$$

$$\psi_F\,0.7854R = \frac{Fx^2d^2}{K}\,0.7854xd \qquad \text{left}$$

$$\psi_M 0.637\,R = \frac{1.57Fxnd^2}{K}\,0.637xd \qquad \text{left}$$

$$EI\,\Delta x_d = \Sigma\psi_c nd \qquad \text{left}$$

$$\psi_F{}^2/_3 l = {}^1/_2 Fn^2d^2/_3 nd \qquad \text{left}$$

90-Deg. Bend ($R/d = 6$). Where the radius of the bend is 6 times the diameter of the pipe ($R/d = 6$), the center-to-face dimension was chosen so that the force acts directly on the bend

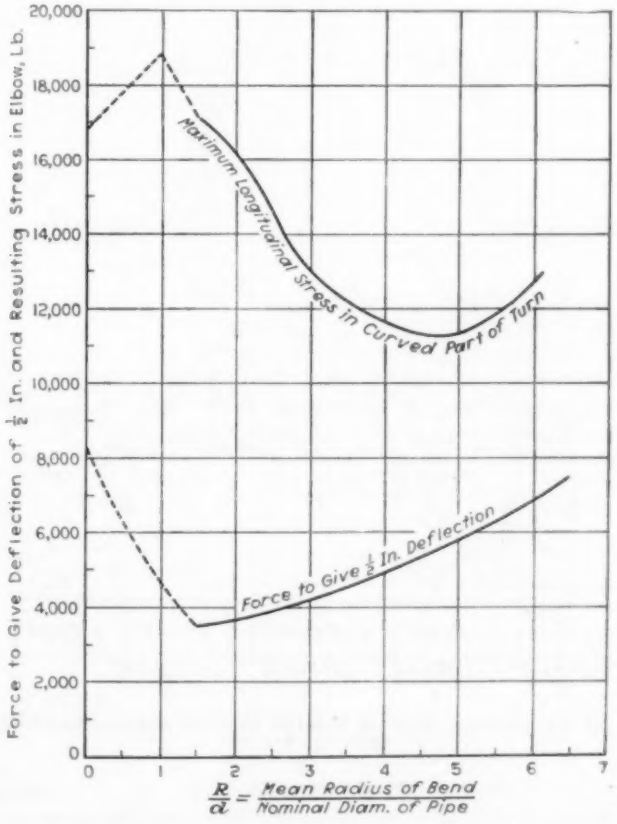


Fig. 13 Relative Flexibility of 90-Deg. Turns Made With Various Types of Fittings and Bends

[10-in. standard-weight pipe; center-to-face length = 60 in. in each case (Fig. 12).]

with no tangent lengths, hence $n = 0$ and the expression for rotation at b drops out and that for rotation at c simplifies to $\psi_c = Fx^2d^2/KEI$. Likewise the expression for deflection reduces to $EI\,\Delta x_c = (Fx^2d^2/K)\ (0.7854xd)$. Hence substituting values for x, d, and K, the relation between force and deflection for a

TABLE 2 RELATIVE FLEXIBILITY OF DIFFERENT TYPES OF 90-DEG. TURNS

Ratio R/d	Reacting force F	Bending moment at b, M	Stress in curved portion at b, S_{max}
6	6,960	417,500	12,700
5	5,730	344,000	11,340
4	4,920	295,000	11,650
3	4,230	254,000	13,000
2	3,710	223,000	16,200
$1^1/_2$	3,500	210,000	17,150
1 cast-steel elbow	4,680	281,000	18,800
0 miter	8,370	502,000	16,800

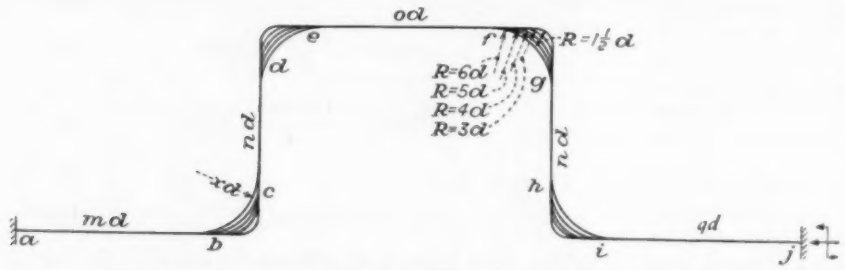


FIG. 14 SKETCH OF SQUARE BEND WITH ARCS OF DIFFERENT RADII (Compared in Fig. 16.)

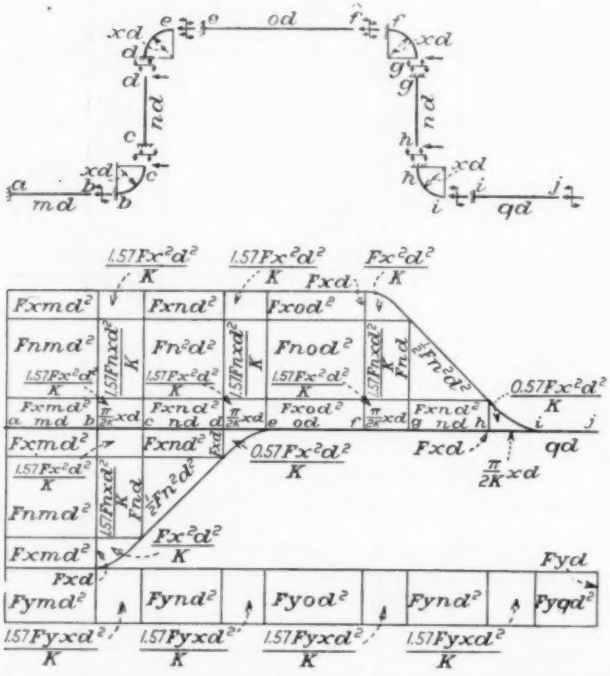


FIG. 15 GENERAL BENDING-MOMENT DIAGRAM FOR SQUARE BEND SHOWN IN FIG. 14

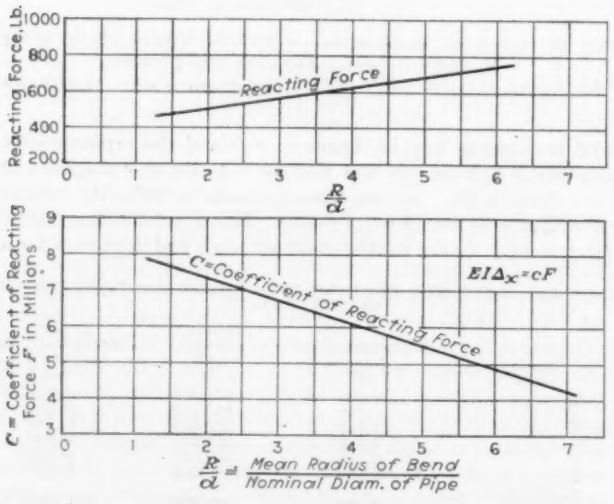


FIG. 16 RELATIVE FLEXIBILITY OF SQUARE BEND (FIG. 14) WITH ARCS OF DIFFERENT RADII

90-deg. bend acted on by a force at the end in the plane of the bend is found. $x = 6$, $d = 10$, $h = tR/r^2 = 0.365 \times 60/29.96 = 0.81$, $K = 0.49$ (Fig. 20).

$$EI\,\Delta_{Xc} = F\,\frac{(36 \times 100)}{0.49}\,(0.7854 \times 6 \times 10) = 346{,}500\,F$$

90-Deg.-Turn Seamless-Tube Fitting. For a seamless-tube fitting ($R/d = 1\frac{1}{2}$) having the same center-to-face dimensions as the bend formed to a radius of 6 pipe diameters, the relation determined between force and deflection is[10]

$$EI\,\Delta_X = 688{,}900\,F$$

90-Deg.-Turn 250-Lb. Flanged Steel Elbow. In the case of the 250-lb. American Standard flanged steel elbow ($R/d = 1$), the greater stiffness of the heavier wall of the fitting must be taken into account. This is done by dividing the length of the abscissa of the diagram representing the elbow by the ratio of the moments of inertia of the cross-section of the elbow and of the pipe comprising the balance of the turn.

The relation between deflection in the direction of the force F and the force acting at the end of the tangent for a turn involving a 250-lb. 10-in. flanged elbow, Fig. 12(*g*), is therefore

$$EI\,\Delta_X = 513{,}000\,F$$

90-Deg.-Turn Miter Elbow. The relation between deflection and force, Fig. 12(*h*), for a miter elbow is

$$EI\,\Delta_X = 288{,}000\,F$$

The relative flexibility of these several types of 90-deg. turns is shown by the equations given in Fig. 12. The greater the value of the coefficient of F, the greater the flexibility. This result is illustrated more effectively by assuming that the free end of each type of turn is displaced an equal amount by the application of a force as indicated. If a displacement of $\frac{1}{2}$ in. is assumed, then, substituting values of E and I for a 10-in. standard-weight pipe, the resulting forces found from the formulas of Fig. 12 give the curves of Fig. 13.

The maximum longitudinal stress S_{1max} which occurs in the middle surface at top of the turn in the curved portion is found by substituting the values of the forces determined in the foregoing into the ordinates of the bending-moment diagrams and applying a factor which takes into account flattening of the cross-section of a curved pipe under flexure. The value of this factor β may be found from Fig. 21 for the various values of h involved in this series of comparative shapes. (See later paragraph for an explanation of the significance of β.) For the particular case in which a seamless-tube fitting is employed, the stress may be found directly from Fig. 22 for standard-weight pipe. The maximum stress occurs at b of Fig. 12(*f*) and is $0.082 \times 3500 \times 60 = 17{,}150$ lb. per sq. in.

The magnitude of the forces required to produce a deflection of $\frac{1}{2}$ in. and the resulting bending moments and stresses produced in the 90-deg. turns of Fig. 12 are tabulated in Table 2.

While the results in Table 2 show the relative flexibility of different types of 90-deg. turns under a conventional condition of one end anchored and the other end free, they are of more or less academic interest in that the important question is not how much more or less flexible an element of this kind is demon-

[10] See Appendix No. 1 for derivation of relations between force and deflection for the several types of 90-deg. turns considered herein.

strated to be with respect to some similar element. The principal consideration is how the bend behaves when incorporated in an actual run of piping. Obviously, since the flexibility of only a portion of a given line will be affected by the substitution of a different type of bend or fitting, the magnitude of the differences found in the above comparison will be reduced when a whole section of piping between anchors is considered as a unit.

Expansion Bends Formed With Different-Radii 90-Deg. Arcs

A series of expansion bends of a given width and height which are composed of 90-deg. arcs of various radii and complementary lengths of straight pipe are illustrated in Fig. 14. This expansion bend is inserted in a straight run of piping rigidly anchored at each end. The dimensions of the various elements are given as ratios to permit a general solution from which proper deductions as to relative flexibility can be readily made. As this shape and the modifications to which it is susceptible are basic elements for providing flexibility in symmetrical piping layouts the general equations involved are given in Appendix No. 2.

The bending-moment diagram for piping arranged as shown in Fig. 14 is given in Fig. 15. From consideration of the requirement that the sum of the rotations of the elements of the line from one end to the other must be zero, an expression for determining the restraining moment at the ends may be developed. This relation is expressed in terms of ratios of length of the elements involved to diameter of pipe. The ordinate of the diagram for restraining moment Fig. 15 is designated as y. Its value is determined from the expression given in Appendix No. 2. The rotations of the elements between each point indicated and the left end of the line are given by the series of equations in Appendix No. 2.

The deflections corresponding to these rotations and to the effect of the forces and moments acting on each element may be found from the general deflection equations of Appendix No. 2. The algebraic sum of the deflections of the elements to the left and right gives the coefficient of the reacting force F. The desired relation between deflection and reacting force is thus obtained. The application of these equations, while tedious, gives directly the relations from which generalizations may be made.

This comparison of a series of square expansion bends of a given width and height installed in a straight run of pipe indicates that for the particular case shown in Fig. 14, the reacting force at the ends is approximately 50 per cent greater if 90-deg. bends having a center-line radius of 6 times the pipe diameter are used than if seamless-tube fittings ($R/d = 1^1/_2$) were employed.

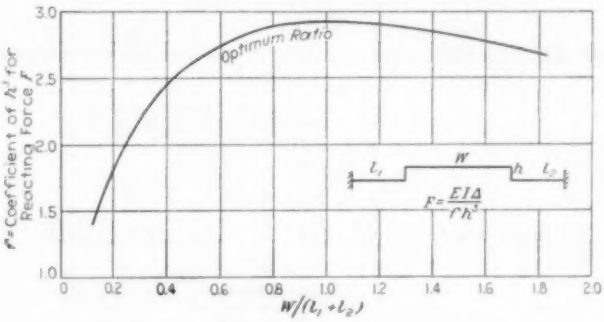


Fig. 18 Optimum Form of Square Bend
[Results obtained from Fig. 39 of Reference 9z.]

The effect of varying the radius of the 90-deg. bends forming the square expansion bend of Fig. 14 is shown graphically in Fig. 16, for an assumed deflection of 0.79 in.

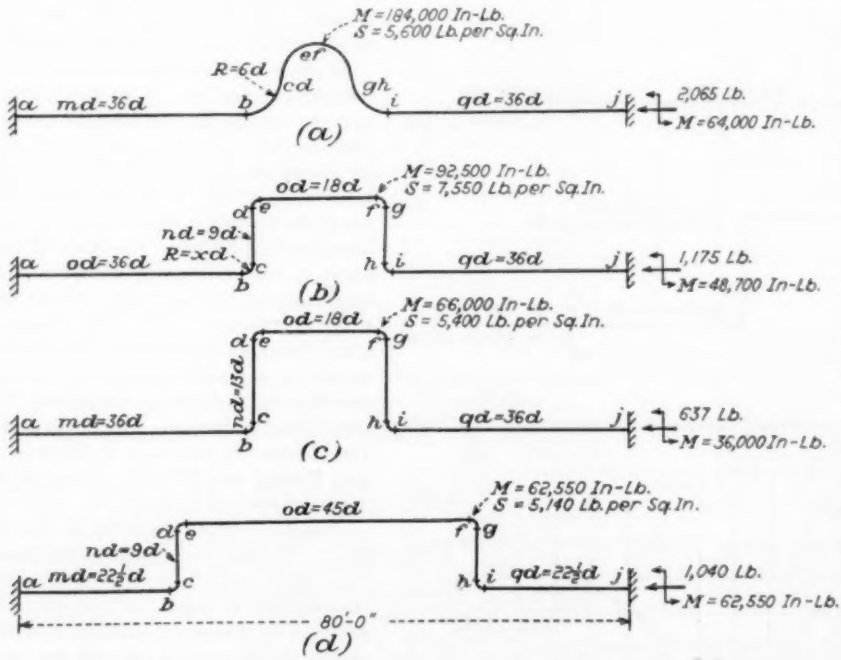


Fig. 17 Comparative Flexibility of Expansion U-Bend and Square Bends

Comparison of Expansion U-Bend and Various Square Bends

A comparison of the relative flexibility of an expansion U-bend and a bend composed of short-radius 90-deg. bends and straight lengths of pipe as shown in Fig. 17 may be made readily from the general equations just described.

Example. Assume a 10-in. line is to carry 15-lb.-gage saturated steam; distance between anchor points is 80 ft.; anchors restrain pipe both as to translation and rotation; and expansion from room to operating temperature is to be taken care of by a suitable expansion U-bend or a square bend formed of seamless-tube fittings and straight lengths of pipe. Required to determine the magnitude of the reacting forces and the maximum bending moment and stress in the line arising from expansion.

Expansion U-Bend. In the case of the expansion U-bend [Fig. 17(*a*)] $x = 6$, $m = 36$, $n = 0$, $o = 0$, $q = 36$, and $K = 0.49$ (see Fig. 20). The coefficient of the reacting force is found to be 2,728,000, hence the relation between the deflection to be cared for and the reacting force is given by

$$EI\,\Delta x = 2{,}728{,}000\,F$$

Square Bend Made With Seamless-Tube Fittings. The square-bend counterpart of the expansion U-bend, Fig. 17(*b*), which employs seamless-tube fittings has the following ratios: $x = 1^1/_2$, $m = 36$, $n = 9$, $o = 18$, $q = 36$, and $K = 0.14$ (see Fig. 20). The coefficient of the reacting force is found to give the relation

$$EI\,\Delta x = 4{,}799{,}200\,F$$

Flexural Rigidity of Pipe. The quantities on the left side of the equations are determined from the elastic properties of the pipe and expansion cared for.

E for steel pipe at 250 deg. fahr. may be taken as 29×10^6

I for 10-in. standard-weight pipe is 160.7 in.^{4}

$\Delta x = 1.51 \times 80/100 = 1.21$ in., where 1.51 represents the linear expansion of 100 ft. of steel pipe from 60 deg. to 250 deg. fahr.

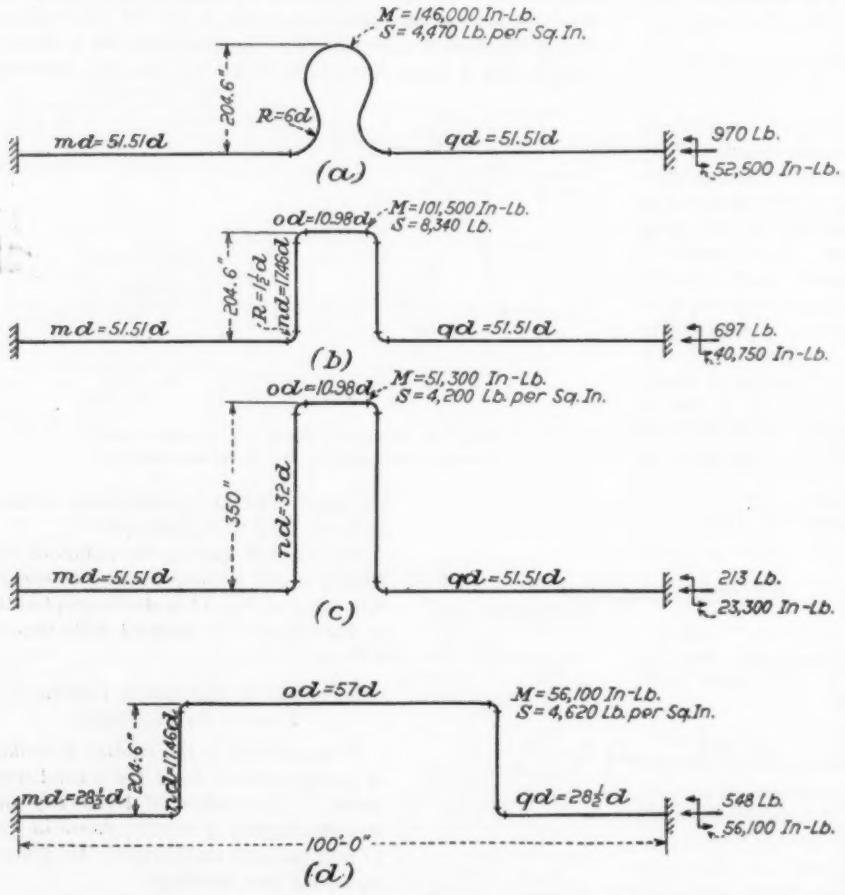


FIG. 19 COMPARATIVE FLEXIBILITY OF DOUBLE-OFFSET EXPANSION U-BEND AND SQUARE BENDS

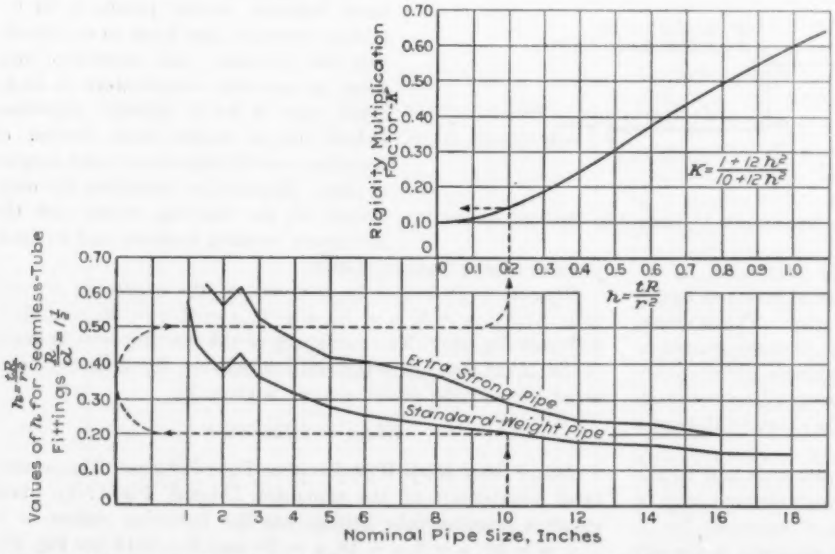


FIG. 20 VALUES OF RIGIDITY MULTIPLICATION FACTOR K FOR FLATTENING OF CIRCULAR CROSS-SECTION OF A CURVED PIPE DURING FLEXURE

Magnitude of Reacting Forces. The reacting force at the end of the line when an ordinary expansion U-bend [Fig. 17(*a*)] is inserted is therefore—

$$29 \times 10^6 \times 160.7 \times 1.21 = 2{,}728{,}000\,F$$
$$F = 2065 \text{ lb.}$$

The corresponding force when the U-bend is replaced by a square bend of the same overall dimensions fabricated of seamless-tube fittings [Fig. 17(*b*)] is found as

$$29 \times 10^6 \times 160.7 \times 1.21 = 4{,}799{,}200\,F$$
$$F = 1175 \text{ lb.}$$

The above values indicating that the latter method [Fig. 17(*b*)] of providing for expansion reduces the reacting forces to approximately 57 per cent of their former value [Fig. 17(*a*)] for the particular conditions used in this example.

Magnitude of Bending Stresses. The bending stresses set up in the curved portions of the line by thermal expansion are found, however, to be greater by about 35 per cent for the square-bend counterpart of the expansion U-bend due to the sharp curvature of the seamless-tube fitting. The effect of the short radius materially increases the magnitude of the longitudinal-stress multiplication factor β which takes into account the effect of flattening of the cross-section of a curved pipe during flexure (see Fig. 21). As indicated in Fig. 17, the maximum stress $S_{l\max}$ occurs at point f and is 5600 lb. per sq. in. in the expansion U-bend and 7550 lb. per sq. in. in the seamless-tube fitting.

Strength Criterion Employed for Curved Pipes. The maximum longitudinal stress $S_{l\max}$ in the middle surface of the pipe cross-section is taken as the strength criterion in the present analysis of bends and fittings in accordance with the conclusions reached by Prof. William Hovgaard in his elaborate study of the stress and strain relations in pipe bends. His conclusion is that, as long as the maximum longitudinal stress as determined by the formula below does not exceed about 20,000 lb. per sq. in. for ordinary steel-piping material, failure of the bend will not occur.

Professor Hovgaard recommends, however, that a total combined stress of 16,000 lb. per sq. in. be used for design purposes, including longitudinal tensile stress due to internal pressure and other stresses which may exist. As a general

principle, limiting the longitudinal bending stress $S_{i\max}$ to 12,500 lb. per sq. in. will provide sufficient margin to cover other stresses involved. Where torsional stress exists, however, it should be combined with the total longitudinal stress, and the maximum equivalent stress determined by the following equation (see Reference 9*c*, p. 559).

$$S_{eq.} = 2\sqrt{S_s^2 + \left(\frac{S_{\max} - S_{\min}}{2}\right)^2}$$

Where $S_{eq.}$ is the total equivalent stress which should not exceed 16,000 lb. per sq. in.; S_s is the shear stress due to torsion in lb. per sq. in.; $S_{\max}$ is the total longitudinal stress in lb. per sq. in. due to bending ($S_{i\max}$) plus the longitudinal stress due to internal pressure; $S_{\min}$ is the direct radial compression due to internal pressure, which at the mean radius of the pipe wall is approximately equal to one-half that pressure in lb. per sq. in. and

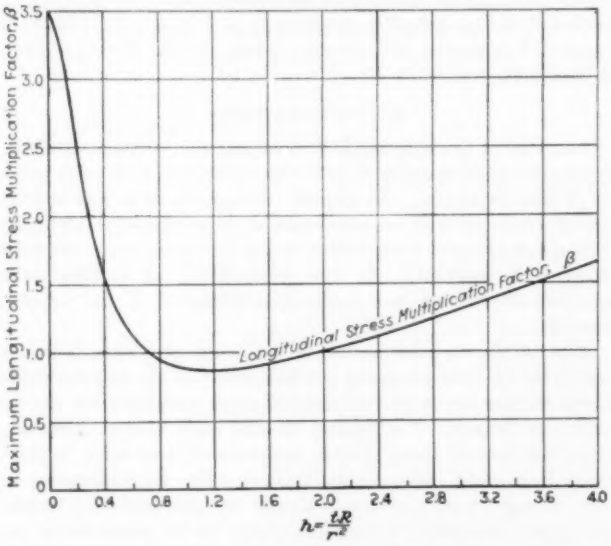


FIG. 21 STRESS MULTIPLICATION FACTOR β FOR MAXIMUM LONGITUDINAL STRESS IN MIDDLE SURFACE OF CURVED PIPE

$$S_{i\max} = \beta Mr/I;\ \beta = \frac{2}{3K}\sqrt{\frac{6h^2+5}{18}},\ \text{where } K = \frac{1+12h^2}{10+12h^2}$$

usually can be neglected. Since compressive stresses are considered as negative, the direct radial stress $S_{\min}$ should be added to $S_{\max}$ if it is taken into account in the computations.

Expression for Maximum Longitudinal Stress in Curved Pipes. The maximum longitudinal stress as developed by Professor Hovgaard is given by $S_{i\max} = \beta Mr/I$ (Reference 9*c*, p. 8), where

$$\beta = \frac{2}{3K}\sqrt{\frac{6h^2+5}{18}}$$

in which $K = (1 + 12h^2)/(10 + 12h^2)$ and $h = tR/r^2$

t = pipe wall thickness, inches
R = radius of center line of bend, inches
r = radius of pipe cross-section to the middle surface, inches.

Values of β are plotted against h in Fig. 21.

This equation is combined with r/I for standard-weight and extra strong pipe so that the stress in a short-radius bend may be obtained directly from the curves of Fig. 22 and the bending moment for each size of pipe. (See equation for maximum moment, Appendix No. 2.)

Comparison of Stresses in Expansion U-Bend and Its Counterpart. Thus in the comparison of the expansion U-bend and its square-bend counterpart made with seamless-tube fittings, it is only necessary to substitute the value of the reacting force F determined for the latter into the ordinates of the bending moment diagram of Fig. 15 to find the maximum bending moment which in this instance occurs at points e or f (see Appendix No. 2 for equations) and is 92,500 in-lb. The stress multiplication factor for a 10-in. standard-weight pipe is found from Fig. 22 as 0.082. The stress at e or f is therefore

$$0.082 \times 92{,}500 = 7550 \text{ lb. per sq. in.}$$

The corresponding stress determined for the expansion U-bend is found from the bending moment at the top of the bend 184,000

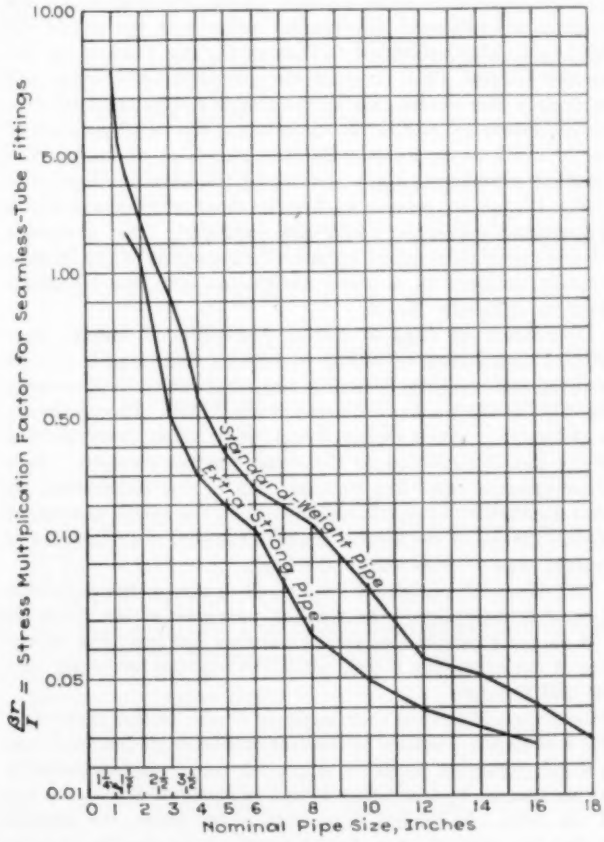


FIG. 22 STRESS MULTIPLICATION FACTOR β FOR MAXIMUM LONGITUDINAL STRESS IN MIDDLE SURFACE FOR SEAMLESS-TUBE FITTINGS

$$S_{i\max} = \left[\frac{2r}{3KI}\sqrt{\frac{6h^2+5}{18}}\right]M$$

in-lb. and the value determined for the longitudinal-stress multiplication factor from Fig. 21 for a bend formed of standard-weight pipe to a radius of 6 pipe diameters ($0.94 \times 184{,}000 \times 5.19/160.7 = 5600$ lb. per sq. in.

An examination of the foregoing comparison shows that the substitution of a square bend composed of seamless-tube fittings and straight lengths of pipe in place of an expansion U-bend has the effect of reducing the magnitude of the reacting forces and bending moments, but at the same time increases the stress in the curved portions of the line.

Bending Moments at Joints. The foregoing comparison deals with bending stresses in the pipe wall and with reactions at anchor points. A third phase of flexibility analysis concerns the maximum amount of bending moment existing at any joint or weld in the line. It is desirable, of course, to choose a design in which the bending moments at all joints are equalized or

minimized with a view to avoiding excessive moments at any of them. Too severe bending moments occurring at joints increase the liability of leaks in service and should be avoided by due consideration in design.

Methods of Reducing Stresses and Bending Moments in Square-Bend Counterpart of Expansion U-Bend. In order that the magnitude of the bending moment at the top of the bend composed of straight lengths of pipe and short-radius bends may not exceed that which is found at the ends of the expansion U-bend, two methods of procedure may be followed. If it is necessary or desirable to maintain the same width as occupied by the expansion U-bend, for some constructional reason, an increase in height of bend by approximately 45 per cent [Fig. 17(*c*)] will result in a lowering of the bending moment at the top of the bend to a value equivalent to that on the end joint of the expansion U-bend. The stress in the curved part of the line will be slightly less in the case of the higher square bend than in the expansion U-bend, while the reacting forces at the ends of the line will be approximately only one-third as large. The foregoing statements apply strictly to the example illustrated in Fig. 17, but are reasonably true for most conditions in which symmetrical expansion bends are employed. The particular relations may be obtained by substituting ratios of the lengths actually involved in a given installation into the equations given in Appendix No. 2.

Distribution of Pipe to Secure Flexibility. A much more efficient distribution of piping as regards total length of pipe involved and resulting flexibility may be obtained by increasing the width of the bend as indicated in Fig. 17(*d*). The maximum flexibility in a given piping layout exists when every portion of the line is subjected to the same bending moment. In a symmetrical straight line such as used in this illustration, an ideal distribution of pipe is obtained when the length of straight pipe at the top of the bend is equal to the sum of the lengths of straight pipe at each end of the bend.

In this case if a height of bend equal to that of the expansion U-bend is maintained, the bending moments at the joints as well as the stress in the curved portions are less than in the case of the expansion U-bend while the reacting forces are only one-half as large.

Conclusion Regarding Distribution of Pipe. In this connection it is well to call attention to the fact that in every practical case in which expansion bends are employed, much greater flexibility can be secured by inserting a length of straight pipe at the top of the bend. In general, for bends in one plane, the amount of straight pipe at the top of the bend should be equal to the sum of the lengths of straight pipe between each end of the bend and the anchor points. This relation is shown graphically for a particular square bend in Fig. 18. It will be noted that the optimum value of f (the coefficient of h^3 which determines the value of the reacting force) results when the ratio of the width to the sum of the lengths between the ends of the bend and the anchor points is unity.

Double Offset Expansion U-Bend

Relative Flexibility. A comparison of the flexibility of a double-offset expansion U-bend with that of a bend having the same overall dimensions made up of seamless-tube fittings is of interest, as this type of bend is often used where a considerable amount of expansion is to be absorbed (see Fig. 19). In this example a 10-in. standard-weight pipe is used as in the foregoing, but the steam pressure is 100 lb. gage (338 deg. fahr.) and the distance between anchors is 100 ft. The resulting forces at the anchors with the double-offset expansion U-bend are 40 per cent greater than with the square-bend counterpart fabricated with short-radius bends [Fig. 19(*b*)], while the maximum longitudinal stress is found to be about 86 per cent greater for the latter.

Methods of Reducing Stresses and Bending Moments in Square-Bend Counterpart of Double-Offset Expansion U-Bend. In this case as with the simple expansion U-bend, two alternatives suggest themselves to reduce the stresses and bending moments so that as good results will obtain as with the double-offset expansion U-bend. If the same width of bend is necessary, the height will have to be 70 per cent greater than that of the double-offset expansion U-bend. If the optimum width may be used as developed in connection with the simple expansion U-bend, the same height can be maintained and the bending moment at the joints will be only slightly greater while the maximum stress will be only 90 per cent as great and the reacting forces only a little more than one-half as large.

Again, it should be noted that the above statements refer to the particular case chosen for analysis but are indicative of the general conclusions which may be drawn. If a true comparison is desired for an actual installation it is necessary to substitute ratios of the lengths of the various parts into the equation given in Appendix No. 2.

III—CONCLUSIONS

An attempt has been made to point out the conclusions drawn from various phases of this paper in connection with each point as it was developed. In general, it may be said that welded piping made up with seamless-tube fittings compares favorably with other types of construction as regards appearance, pressure drop, and flexibility. It has pronounced advantages over mitered elbows and short-radius cast fittings in all the respects named.

Due to the complex nature of expansion problems, it is not advisable to draw sweeping conclusions as to the relative flexibility of pipe bends as distinguished from seamless-tube elbows and cast fittings. The typical specific cases chosen, however, show the general trend of such comparisons, and serve to illustrate how other cases can be treated. The suggestions as to how straight pipe and bends should be distributed to obtain maximum flexibility, the authors deem to be particularly important (see discussion in connection with Figs. 17, 18, and 19).

The rather sketchy nature of the data offered on pressure drop through turns of different radii is appreciated by the authors, who hope that interested engineers will call attention to references which may have been overlooked, or present unpublished data of their own bearing on the subject. The solution of piping-flexibility problems has received considerable attention of late and the authors would appreciate a frank discussion of how to make the methods they have used more readily understood and useful. It is especially desirable to further develop the application of these principles to special cases and work out short-cut methods for computing the flexibility of common types of expansion members.

Appendix No. 1

(1) *90-Deg.-Turn Seamless-Tube Fitting* [Fig. 12(*f*)]. The relation between deflecting force and the deflection is determined as follows: The values of x, n, and K become:

$$x = 1\tfrac{1}{2}, n = 4\tfrac{1}{2}, h = 0.203 \text{ (Fig. 20) and } K = 0.14 \text{ (Fig. 20)}$$

$$\psi_b = 1\tfrac{1}{2} \times 4\tfrac{1}{2} \times 100F/EI + 4\tfrac{1}{2} \times 4\tfrac{1}{2} \times 100F/EI = 2700F/EI$$

$$\psi_e = 2700(F/EI) + \frac{1\tfrac{1}{2} \times 1\tfrac{1}{2} \times 100F}{0.14EI} + \frac{1.57 \times 1\tfrac{1}{2} \times 4\tfrac{1}{2} \times 100F}{0.14EI} = 11{,}745F/EI$$

$$EI\,\Delta x_c = \Sigma\ 2{,}700F \times 1\tfrac{1}{2} \times 10 = 40{,}500F$$
$$1{,}585F \times 0.7854 \times 15 = 18{,}700F$$
$$7{,}460F \times 0.637 \times 15 = 71{,}300F$$
$$EI\,\Delta x_d = \Sigma\ 11{,}745F \times 4\tfrac{1}{2} \times 10 = 528{,}000F$$
$$\tfrac{1}{2}\,(4\tfrac{1}{2})^2\,100F \times \tfrac{2}{3} \times 45 = 30{,}400F$$
$$688{,}900F$$

$$EI\,\Delta_x = 688{,}900F$$

(2) *90-Deg.-Turn 250-Lb. Flanged Steel Elbow* [Fig. 12(*g*)]. The minimum wall thickness of a 250-lb. 10-in. elbow is $^{15}/_{32}$ in. The corresponding moment of inertia is 211 in.^{4} while the moment of inertia of the 10-in. standard-weight pipe is 160.7 in.^{4} The ratio by which the abscissa of the diagram must be divided is therefore 211/160.7 = 1.315. $x = 1$, $n = 5$, $h = tR/r^2 = 0.468 \times 10/27.4 = 0.171$. The value of K corresponding to $h = 0.171$ from Fig. 20 is 0.13.

The equations for rotation and deflections become

$$\psi_b = (1 \times 5 \times 100F/EI) + 25 \times 100F/EI = 3000F/EI$$

$$\psi_c = 3000F/EI + \frac{100F}{1.315 \times 0.13EI} + \frac{1.57 \times 1 \times 5 \times 100F}{1.315 \times 0.13EI} = 8164F/EI$$

and

$$EI\,\Delta x_c = \Sigma\ 3{,}000F \times 10 = 30{,}000F$$
$$584F \times 7.85 = 4{,}580F$$
$$4{,}580F \times 6.37 = 29{,}200F$$
$$EI\,\Delta x_d = \Sigma\ 8{,}164F \times 50 = 407{,}500F$$
$$\tfrac{1}{2} \times 2{,}500F \times 33.3 = 41{,}700F$$
$$512{,}980F$$

$$EI\,\Delta_x = 512{,}980F$$

(3) *90-Deg.-Turn Miter Elbow* [Fig. 12(*h*)]. In the case of the miter elbow, $x = 0$, $n = 6$, and $K = 1$.

$$\psi_c = (F/EI)n^2d^2 = 36 \times 100F/EI = 3{,}600F/EI$$
$$EI\,\Delta x_d = \Sigma\psi_c nd = 3{,}600F \times 60 = 216{,}000F$$
$$\psi_F{}^2/_3 nd = \tfrac{1}{2} \times 36 \times 100 \times 40F = 72{,}000F$$
$$288{,}000F$$

$$EI\,\Delta_x = 288{,}000F$$

Appendix No. 2

GENERAL EXPRESSIONS INVOLVED IN COMPUTATION OF FLEXIBILITY OF SYMMETRICAL SQUARE BENDS, U-BENDS, ETC.

(1) *Ordinate of Resisting Moment.* Let y be the coefficient of the restraining moment; then its value may be found from the equation

$$y = \frac{(2x + n)\left(\frac{3.14x}{K} + n + o\right)}{m + 2n + o + q + \frac{6.28x}{K}}$$

In which m, n, o, q, and x are ratios of lengths to pipe diameter as indicated in Fig. 14.

(2) *General Equations for Rotations of a Square Bend Composed of Arcs and Straight Lengths of Pipe* (Fig. 15).

$\psi_b = Fmyd^2/EI$ counterclockwise

ψ_c = $\psi_b + \frac{1.57Fxyd^2}{KEI} + \frac{Fx^2d^2}{KEI}$ counterclockwise

= $\frac{1.57Fx^2d^2}{KEI}$ clockwise

ψ_d = $\psi_{c-} + Fnyd^2/EI + Fn^2d^2/2EI$ counterclockwise

= $\psi_{c+} + Fxnd^2/EI + Fn^2d^2/EI$ clockwise

ψ_e = $\psi_{d-} + \frac{1.57Fxyd^2}{KEI} + \frac{0.57Fx^2d^2}{KEI}$ counterclockwise

= $\psi_{d+} + \frac{3.14Fx^2d^2}{KEI} + \frac{1.57Fxnd^2}{KEI}$ clockwise

ψ_f = $\psi_{e-} + Foyd^2/EI$ counterclockwise

= $\psi_{e+} + 2Foxd^2/EI + Fond^2/EI$ clockwise

ψ_g = $\psi_{f-} + \frac{1.57Fxyd^2}{KEI}$ counterclockwise

= $\psi_{f+} + \frac{Fx^2d^2}{KEI} + \frac{1.57Fxnd^2}{KEI} + \frac{1.57Fx^2d^2}{KEI}$ clockwise

ψ_h = $\psi_{g-} + Fnyd^2/EI$ counterclockwise

= $\psi_{g+} + Fn^2d^2/2EI + Fxnd^2/EI$ clockwise

ψ_i = $\psi_{h-} + \frac{1.57Fxyd^2}{KEI}$ counterclockwise

= $\psi_{h+} + \frac{0.57Fx^2d^2}{KEI}$ clockwise

ψ_j = $\psi_{i-} + Fqyd^2/EI$ counterclockwise

= ψ_{i+} clockwise

(3) *General Equations for Deflection of a Square Bend Composed of Arcs and Straight Lengths of Pipe* (Fig. 15).

$EI\,\Delta x_a = \Sigma\ \psi_b R = \psi_b xd$ left

$\psi_F 0.7854R = \frac{Fx^2d^2}{K}\,0.7854xd$ left

$\psi_M 0.637R = \frac{1.57Fxyd^2}{K}\,0.637xd$ left

$\psi_M 0.637R = \frac{1.57Fx^2d^2}{K}\,0.637xd$ right

$EI\,\Delta x_d = \Sigma\ \psi_c l = \psi_c nd$ left

$\psi_M{}^1/_2 l = (Fxnd^2 + Fn^2d^2)^1/_3 nd$ right

$\psi_M{}^1/_2 l = (Fnyd^2)^1/_3 nd$ left

$\psi_F{}^2/_3 l = (^1/_3 Fn^2d^2)^2/_3 nd$ left

$EI\,\Delta x_e = \Sigma\ \psi_d R = \psi_d xd$ left

$\psi_M 0.363R = \left(\frac{3.14Fx^2d^2}{K} + \frac{1.57Fxnd^2}{K}\right) 0.363xd$ right

$\psi_M 0.363R = \left(\frac{1.57Fxyd^2}{K}\right) 0.363xd$ left

$\psi_F 0.624R = \left(\frac{0.57Fx^2d^2}{K}\right) 0.624xd$ left

$EI\,\Delta x_g = \Sigma\ \psi_f R = \psi_f xd$ left

$$\psi_F 0.7854R = \left(\frac{Fx^2d^2}{K}\right) 0.7854xd \qquad \text{left}$$

$$\psi_V 0.637R = \left(\frac{157.xndF}{K} + \frac{1.57Fx^2d^2}{K}\right) 0.637xd \qquad \text{left}$$

$$\psi_M 0.637R = \left(\frac{1.57Fxyd^2}{K}\right) 0.637xd \qquad \text{right}$$

$$EI\,\Delta\Sigma_{\Delta h} = \Sigma\,\psi_o l = \psi_o nd \qquad \text{left}$$

$$\psi_F{}^2/_3 l = ({}^1/_2 Fn^2d^2)^2/_3 nd \qquad \text{left}$$

$$\psi_M{}^1/_3 l = (Fxnd^2)^1/_2 nd \qquad \text{left}$$

$$\psi_M{}^1/_3 l = (Fnyd^2)^1/_2 nd \qquad \text{right}$$

$$EI\,\Delta_{X'} = \Sigma\,\psi_n R = \psi_h xd \qquad \text{left}$$

$$\psi_F 0.624R = \left(\frac{0.57Fx^2d^2}{K}\right) 0.624xd \qquad \text{left}$$

$$\psi_M 0.363R = \left(\frac{1.57Fxyd^2}{K}\right) 0.363xd \qquad \text{right}$$

(4) *Equation for Maximum Bending Moment in Square Bend* (Fig. 15).

$$M = (2x + n - y)Fd$$

Appendix No. 3

LIST OF SYMBOLS

The symbols used in this paper are defined below:

- Δ_X = deflection along the X-axis, inches
- F = generalized force acting as shown, pounds
- l = length of cantilever or section of pipe, inches
- E = modulus of elasticity of pipe material at the working temperature of the piping, pounds per square inch
- I = moment of inertia of pipe cross-section, inches to the fourth power
- K = rigidity multiplication factor $= \dfrac{1 + 12h^2}{10 + 12h^2}$
- β = longitudinal stress multiplication factor $= \dfrac{2}{3K}\sqrt{(6h^2 + 5)/18}$
- h = tR/r^2 = pipe-bend ratio
- t = thickness of pipe wall, inches
- R = mean radius of bend, inches
- r = mean radius of pipe cross-section, inches
- d = nominal diameter of pipe, inches
- ψ_M = rotation due to a moment at the end of a cantilever; = area of rectangular bending-moment diagram, divided by EI
- ψ_F = rotation due to a transverse force at the end of a cantilever; = area of triangular bending-moment diagram, divided by EI
- $\psi_{b\,c,etc.}$ = algebraic sum of the rotations of the elements between the point designated by the subscript and the fixed end of the section of piping; = algebraic sum of the areas of the bending-moment diagrams to the left of the point designated by the subscript, divided by EI
- $m, n, o, q,$ and x = ratios of lengths of pipe to pipe diameter involved in computation of flexibility of symmetrical square bends, U-bends, etc.

Discussion

GEORGE J. STUART.[11] When designing a piping system, the problem of caring for expansion is one of great importance and one concerning which little specific information is at hand. This paper, together with the authors' previous work, is therefore most timely.

Figs. 17 and 19 of the paper and the conclusion that the seamless-tube-fitting method compares favorably with the pipe-bend method of providing flexibility are interesting.

The capacity of a pipe bend to take care of expansion is limited by the maximum stress set up in the pipe. It will be noted from Fig. 19 that for the conditions given the maximum stress in the pipe bend is 4470 lb. per sq. in., whereas, for the same conditions, the stress in the seamless-fitting method is 8340 lb. per sq. in., or an increase of 79 per cent.

It will also be noted that the maximum stress, Fig. 19 (*b*), comes at a weld, while in the pipe bend the maximum stress is taken by the pipe itself.

The authors have suggested a method of reducing the stresses by means of increasing the proportion. The stresses can undoubtedly be reduced in this way, but in actual practice the space available for expansion bends will only rarely permit this expedient; furthermore, when space permits, the method is applicable to pipe-bend as well as to seamless-tube fitting.

It always is desirable to minimize the number of joints in a pipe line. The seamless-tube-fitting expansion U-bend has necessarily eight welded joints, while the equivalent bend has but two joints, which may be either flanged or welded.

It is interesting to compare the quantity of labor and material entering into the two methods. The amount of pipe entering into the seamless-tube-fitting construction is almost the same as in the pipe-bend construction—more exactly, 0.9 as much if the pipe bend has a radius of 5 diameters. If the radius is 6 diameters, 0.95 as much pipe is used, and if the radius is increased to 7 diameters, the quantity of pipe in the two methods is the same. To this must be added the expense of making eight welds and the cost of four seamless-tube fittings. This cost of welding and fittings more than offsets the expense of bending the pipes.

In view of the foregoing, the writer believes that the pipe-bend construction has the following advantages over the equivalent seamless-tube-fitting construction:

1 Lower stress per unit of flexibility
2 Fewer joints
3 If any joints are necessary in the pipe-bend construction, they can be placed at points of low stress
4 More economic of space for equal stress and flexibility
5 Less expensive.

A. M. HOUSER[12] AND S. HIRSCHBERG.[13] Those of us who have dealt with piping flexibility problems can appreciate that the grapho-analytical method based on area moment diagrams introduced by the authors to this application is a decided advancement in solving complicated piping layouts. The authors should be commended for this admirable contribution.

It is interesting to note, however, that throughout the paper the authors emphasize the preferable use of seamless-tube elbows having a mean radius of curvature of $1^1/_2$ times the inside pipe diameter. They treat this subject from two phases, namely, pressure drop and flexibility. The writers realize that treatment of this subject from all angles would entail an article of consider-

[11] Engineer, Power Piping Society, Pittsburgh, Pa. Mem. A.S.M.E.

[12] Product Engineer, Crane Company, Chicago, Ill. Mem. A.S.M.E.

[13] Crane Company, Chicago, Ill.

able length, yet the writers feel that the illustrations and problems as presented by the authors may be misleading to the piping designer not intimate with this subject. The writers wish to bring out some points not discussed by the authors which will indicate that a ready-made elbow need not be confined to a mean radius of $1^1/_2$ times the inside diameter nor should it be made of seamless tubing exclusively; also, that expansion bends made of continuous parts of a circle have many advantages over square bends.

Pressure Drop in Elbows and Bends

Prof. F. E. Giesecke in 1927 conducted a series of tests on short-radius and long-radius standard 90-deg. elbows to determine the comparative friction losses of each. The report of this investigation can be found in the University of Texas Bulletin No. 2712. Quoting Professor Giesecke:

> Comparing a short radius 90-deg. elbow with a long-radius elbow, the friction in a long-radius elbow is about 63 per cent of that in a short-radius elbow, for a velocity of 3 ft. per sec. and a 3-in. pipe. This percentage is reduced slightly as the velocity is increased, and increased as the pipe diameter is increased.

Comparing this statement with the authors' statements and references would indicate that there is considerable difference of agreement of results. According to the authors' curves there is a much more decided rapid decrease in elbow friction as R/d increases from 1 to 4, which is the comparative relation of the short-radius and long-radius elbows' mean radii of curvature.

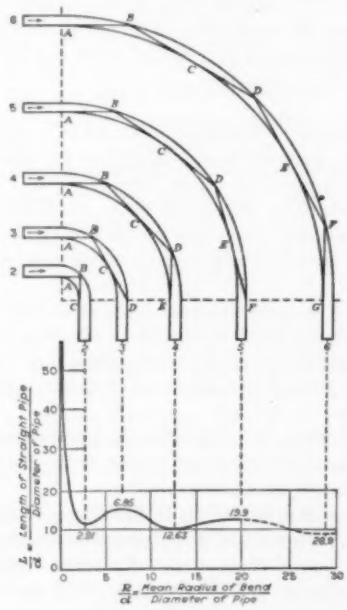


Fig. 23

Fig. 23 is reproduced from *Engineering*, of London, in the November 13, 1925, issue, based on data presented in a paper by Prof. J. Eustice. Professor Eustice showed by colored filaments the motion of water in its flow through curved glass pipes for the purpose of ascertaining the causes of increased resistance to flow due to curvature of pipe. The average curve from results of experiments made by several investigators on circular pipes from 2 in. to 6 in. is shown in Fig. 23. An inspection of the curve indicates that the friction loss in curved pipes with respect to R/d follows a peculiar law of maxima and minima and that the minimum pressure drop for the least R/d is decidedly at 2.91 and not 1.5 to 2 as indicated by the authors. Also, that there is not as decided a rapid decrease in pressure drop when R/d increases from 1 to $1^1/_2$, as the authors contend.

Professor Eustice states that after a series of calculations based on theoretical assumptions, it was found that for right-angled bends, the results which agree most closely with the experiments are obtained when the line AB is drawn as a tangent to the inner circle of the curved pipe and continued as at A, B, C, D, E in Fig. 23. If the joint with the tangent pipe is at E, the fluid passes along the straight pipe with minimum resistance. This is illustrated for right-angled bends in Fig. 23, cases 2, 3, 4, 5, and 6. In each of cases 2, 4, and 6 where the lines BC, DE, and EG, respectively, are tangent at the outlet, there is a minimum. In each case of cases 3 and 5, where the lines BCD and DEF, respectively, cut the curve at the outlet, there is a maximum.

Another consideration of pressure drop in right-angled bends must be made which is discussed in the "Handbook on Hydraulics" by H. W. King. In designing a pipe line, it may be found

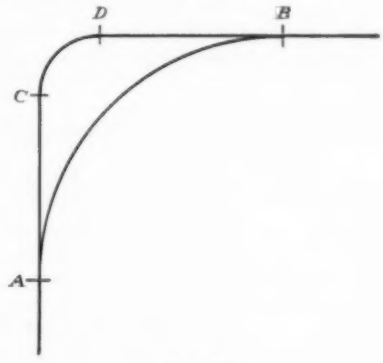


Fig. 24

that the total loss of head in pipe line between two given points will be less by using a curve of greater radius due to shortening the length of pipe. This may be seen from Fig. 24. Assuming that the radius of the bend CD is from 4 to 7 ft., the radius giving minimum excess loss of head, the bend AB having a greater radius than CD; the total loss of head in the pipe AB may be less than the total loss of head in the pipe $ACDB$ because of its shorter length. This would tend to show that in Fig. 12 of the authors' paper pressure drop is sacrificed when comparing Fig. 12 (*f*) with Fig. 12 (*a*).

Relative Flexibility of Elbows and Long-Radius Pipe Bends

Comparing the merits of so-called "square bends" made up of cast-steel fittings and straight pipe as contrasted with "square bends" made up of the seamless-tube elbow and straight pipe, the writers wish to call to the attention of the authors the following:

In Fig. 12 (*g*) of the authors' paper is shown the relation between deflection and force of $EI\,\Delta_x = 513{,}000\,F$ for 250-lb. American standard flanged steel elbow with $R/d = 1$. A comparison of the relative flexibility of cases Fig. 12 (*f*), seamless-tube elbow, with Fig. 12 (*g*) cast-steel elbow, taking tensile strengths in consideration, is as follows:

Seamless-Tube Fitting	*Cast-Steel Fitting*
$R/d = 1^1/_2$	$R/d = 1$
$EI\,\Delta_x = 688{,}900\,F$	$EI\,\Delta_x = 513{,}000\,F$
Max. longitudinal stress = 17,150	Max. longitudinal stress = 18,800
Tensile strength = 54,000	Tensile strength = 70,000

Therefore, using the permissible longitudinal stress of 12,500 lb. per sq. in. as recommended by the authors for seamless steel tubing, the permissible longitudinal stress for cast steel using the same factor of safety would be approximately 15,500. In other

words, using the same factor of safety, it can be computed by simple mathematics that the deflection obtained by the cast-steel elbow bend would be slightly greater than that obtained by use of the seamless-tube elbow, not considering any additional flexibility due to the screw joint over the rigid welded joint. Furthermore, if the authors would have determined the relation between deflection and force for the seamless-tube elbow as in Fig. 12 (*g*) for the cast-steel elbow, the coefficient would have been greater than 688,000 as of $R/d = 1^1/_2$, as in Fig. 12 (*f*), although the longitudinal stress may have been greater.

Comparing Fig. 12 (*a*) with Fig. 12 (*f*), it can be readily seen that for $R/d = 1^1/_2$ the coefficient of the relation between deflection and force shows a decided advantage over $R/d = 6$, yet the longitudinal stress for $R/d = 1^1/_2$ is much greater than when $R/d = 6$. This means that if the permissible stress is 12,500 the deflection would be considerably less for $R/d = 1^1/_2$ than for $R/d = 6$, although the force would be greater in the latter. It must be remembered that the condition selected by the authors is the most beneficial to the $R/d = 1^1/_2$ bend, and here a lesser deflection is obtained than $R/d = 6$, using the same permissible fiber stress.

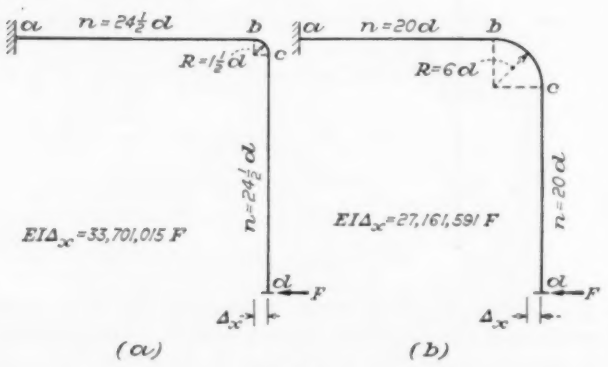


FIG. 25

Suppose a more practical situation, as shown in Fig. 25, where the face-to-face dimension is 26 pipe diameters.

Assuming the same permissible stress of 12,500 and diameter of pipe $d = 10$ in., the two bends as shown in Fig. 25 compare as follows:

Bend $R/d = 1^1/_2$	Bend $R/d = 6$
$\Delta_x = 4^1/_4$ in.	$\Delta_x = 8^1/_2$ in.
$F = 585$ lb.	$F = 1450$ lb.

Upon comparison of the coefficients of the relation between deflection and force in Fig. 24, as contrasted with the authors' Fig. 12 (*a*) and (*f*), the pronounced difference of the coefficients as pointed out by the authors' case becomes insignificant in the case illustrated in Fig. 24.

As indicated, the bend having $R/d = 6$ offers twice the amount of deflection as the bend with $R/d = 1^1/_2$. Is not this the more practical and most economical way of getting the most out of an expansion bend with the same factor of safety? The providing of anchorage of 585 lb. as compared to 1450 lb. to obtain twice the flexibility should not present a difficult problem of structural designing.

From inspection of Fig. 19 of the authors' paper it can be seen that considerably more space and pipe will be required to make up a square bend of seamless-tube elbows and straight pipe as compared with the fabricated double-offset expansion U-bend. The double-offset expansion U-bend requires considerably less pipe and can take up a minimum of space.

One point the authors fail to discuss is that of concentration of stress at the weld. The authors agree that the maximum fiber stress occurs at (*b*) in Fig. 12 of their paper. From an inspection of the illustration shown in Fig. 5 of the authors' paper it can be seen that no tangents are integral with the seamless-tube elbow. The weld is made at the point where the radius meets the straight pipe to which the elbow is welded, and this is where the maximum fiber stress occurs. An elbow having tangents would have a decided advantage over the elbow described by the authors.

The authors also confine their discussion to an elbow made of seamless-steel tubing; is there anything wrong with an elbow forged in two halves and welded together having the same thickness as pipe and made of steel having similar characteristics as tubing? After all, is not a welded line as strong as its welded joints? It may be of interest to state that ready-made tees for welding can be obtained on the market which are made by forging them in halves and welding together.

CONCLUSIONS

In concluding, the writers believe that the seamless-tube elbow and welded elbows having tangents (forged in halves) should be confined to such use as a fitting and not as an integral part of an expansion pipe bend unless in extraordinary cases.

The writers do not agree with the authors that the seamless-tube fittings, as pointed out, compare favorably with long-radius pipe bends as regards flexibility, and have not pronounced advantages, as pointed out by the authors, over the cast fittings as regards appearance, pressure drop, and flexibility.

A fitting made of seamless-tube having tangents has a decided advantage over the seamless-tube described by the authors because of the pronounced decrease of stress concentration due to expansion or other stresses.

J. ROY TANNER.[14] The authors are to be commended for the systematic arrangement of existing data on the losses which occur in changing the direction of the flow of fluids. Such data are more incomplete than one could wish, especially in respect to test conditions and this accounts for the divergence in the specific determination of losses under given conditions by different observers.

Enough observers agree to make it apparent that up to $1^1/_2$ pipe diameters in radius the friction loss in bends is greatly reduced through increasing the radius of the bend and thereafter more slowly, reaching the minimum at about 5 diameters, after which it may increase.

For low velocities the difference in loss between $1^1/_2$ and 5 diameters is small, but for steam at high velocities it becomes appreciable as shown by all observers.

The data from Fig. 8 from the Buffalo Forge Company indicate that for bends with radii of $2^1/_2$ pipe diameters the pressure loss is only 72 per cent of those of $1^1/_2$ diameters. Fig. 9, from Edwards, shows the loss at 5 diameters to be 76 per cent of that for $1^1/_2$ diameters. Fig. 11, from Davis, shows that the loss at 5 diameters is only 43 per cent of that at $1^1/_2$ diameters, and from Weisbach 75 per cent.

These figured losses differ numerically, but all indicate that enough pressure drop can be prevented through the use of pipe bends of about 5 diameters in radius instead of shorter bends or fittings to make their use preferable where space permits.

Another point to be taken into account by the designer is plant investment. From this point of view, in the writer's experience, under present-day conditions and taking into account the cost of additional joints entailed by short-radius bends and the additional pipe necessary to equalize line length, the pipe bend of customary design has the advantage.

[14] Pittsburgh, Pa. Mem. A.S.M.E.

The writer is glad of this opportunity to second the plea of the authors for more reliable data on this subject.

ROBERT CRAMER.[15] This is an excellent presentation of some of the problems which confront the designer of piping systems for modern conditions of high pressure and temperature. One of the merits of the paper is the fact that it refrains from giving explicit instructions or "recipes" for design, which usually cover only some very special case. The authors rather present a thoughtful analysis of some assumed cases, thereby stimulating the attentive reader to apply similar thoughtful methods to his own problems.

A case which came under the writer's observation several years ago illustrates this point. In a straight high-pressure steam line leading from a boiler plant to a group of steam turbines, an expansion bend similar to Fig. 19 (*a*) was installed, without making a close estimate of the flexibility of the bend or the reactions produced. With the first heating up of the system one of the turbines was dislocated from its foundation. The bend was then replaced by an expansion joint of the stuffing-box type.

The authors mention in their conclusions that the data offered on pressure drop are sketchy. This is true of practically all information available on this subject. A thorough review and analysis of all the available data would seem desirable.

In the presentation of data referring to flow of water by the authors the friction in the straight pipe which is replaced by long-radius elbows is taken into consideration. If this were also done in the case of air flow, a picture somewhat different from that presented in Fig. 8 would appear.

For instance, comparing the largest elbow, $R/d = 5$, with the smallest, not only is the elbow with radius $^1/_2d$ equivalent to 121.2 diameters, but the installation of this elbow requires an additional amount of straight pipe of 9 pipe diameters, as compared with the elbow of radius $5d$.

This difference (less than $7^1/_2$ per cent) does not seem large, but the percentage is much greater in the intermediate cases, as is shown in the following table:

R/d	0.50	0.75	1.00	1.25	1.50	2.00	3.00	5.00
$2(5 - R/d)$	9.00	8.50	8.00	7.50	7.00	6.00	4.00	0.00
l/d	121.2	35.09	17.51	12.72	10.36	9.03	8.24	7.85
Total equivalent length in diameters	130.2	43.59	25.51	20.22	17.36	15.03	12.24	7.85

The curve representing these figures, when plotted, will show a much steeper inclination at $R/d = 5$ than the curve of Fig. 10. This seems to indicate that, from the standpoint of friction loss, it is desirable to make the radii of all bends as large as the space limitations permit.

WILLIAM HOVGAARD.[16] The seamless-tube fittings referred to are, it is supposed, of the same type as the so-called "tube-turns." Last year the writer made a series of tests on a 12-in. tube turn bent to 180 deg. with a radius of $1^1/_2$ times the diameter. The wall thickness was $^3/_8$ in. An abstract of the results was given in a paper read before the International Congress of Applied Mechanics in Stockholm in August, 1930. The writer will here merely mention that although the curvature of this pipe was much sharper than assumed in his theory on the bending of curved tubes the deflections as well as stresses corresponded remarkably well with the results of the formulas. As in bends of more easy curvature it was found that the increments in the deflection ΔX remained proportional to the increments in the corresponding force F up to a longitudinal stress of 20,000 lb. per sq. in. It appears that as in case of ordinary pipe bends a total working longitudinal stress of 16,000 lb. per sq. in. is permissible. When torsional stresses exist, the principal stress should be found by the formula given in the paper, and stress so determined should then not exceed 16,000 lb. per sq. in.

This is essentially a practical engineers' paper, and will undoubtedly prove very useful to piping engineers. It brings the problems involved in the design of pipe lines up to the most recent practice and presents to the profession a wealth of valuable information, theoretical and practical.

The proposed area-moment method is very ingenious and practical. It enables the designing engineers to solve otherwise unapproachable three-dimensional problems.

The writer wishes to add that the representation of the stress multiplication factor β, as given in Fig. 21, and which was published in one of his papers on pipe bends, was suggested by Mr. A. Edwards of the Stone & Webster Engineering Corporation.

The correct references to the writer's various papers on this subject are as follows:

Journ. of Math. & Phys., M.I.T., Vol. VI, No. 2, 1926; Vol. VII, No. 3, 1928; Vol. VII, No. 4, 1928; and Vol. VIII, No. 4, 1929.

World Eng. Congress, Tokyo, 1929, "Deflections and Stresses in Pipe Bends."

Intl. Congress of Applied Mechanics, Stockholm, 1930, "Bending of Curved Pipes."

W. R. TOWNSEND.[17] The frictional resistance data furnished are valuable and important from a practical point of view as an assembly of results of previous tests. The discussion has so far demonstrated that there is a considerable difference of opinion regarding results under varying conditions of flow, and has brought out additional data. Further information will doubtless be uncovered on this subject.

It would be well if test results could be given for the second half of the paper, on flexibility; the authors do not make clear whether tests have been conducted, and it is assumed from the paper that the results are on a theoretical basis only.

It is important from the standpoint of the designer to know the practical results and to know how the piping will act when installed. The results of actual tests will be awaited with interest.

W. P. CURLEY.[18] Speaking as a representative of the manufacturers of the seamless tube fittings referred to by the authors, the writer considers their paper a valuable and timely contribution, recognizing as it does the increasing popularity of pipe welding and the somewhat different problems regarding both frictional resistance and flexibility, which must be considered in designing a piping layout having in view welding and stock welding fittings.

Several engineers referred to in the authors' paper are in accord with the theory regarding the rigidity multiplication factor K, which corrects for flattening in the circular cross-section of a curved pipe.

The derivation of K assumes, however, that the stresses are acting on an undeformed section of the pipe, which is true in the case of the seamless tube fitting, but not always true of the pipe bend. Before flexure takes place, the former has a true circular cross-section and a uniform wall thickness corresponding with that of the adjoining straight length of pipe, whereas the pipe bend may have initial ellipticity. It should be emphasized, therefore, that the corrections considered in the paper for flattening cover only the distortion of the circular cross-section during flexure, and do not compensate for any inherent flattening, thinning, and buckling which may have occurred in fabricating the bend from a straight piece of pipe.

[15] Consulting and Chief Engineer, Sewerage Commission, Milwaukee, Wis. Mem. A.S.M.E.

[16] Professor, Massachusetts Institute of Technology, Cambridge, Mass.

[17] Engineer, Baker and Spencer, Inc., New York, N. Y. Assoc. Mem. A.S.M.E.

[18] Tube-Turns, Inc., Louisville, Ky.

A. M. Wahl, in a paper entitled "Stresses and Reactions in Expansion Pipe Bends" (Trans. A.S.M.E., Vol. 50, No. 15, p. 241, Paper FSP-50-49, 1928), has investigated the effect of stress due to permanent flattening or initial ellipticity of the cross-section produced during fabrication of the bend, and has found that while this flattening does not materially influence the stress due to bending moments, it does greatly increase the stress due to internal pressure, and that these stresses, when superimposed on those due to bending moments, may result in stresses sufficiently high to produce permanent set in the piping.

In a more recent paper, "Stresses in Turbine Pipe Bends," (*Mechanical Engineering*, Vol. 51, No. 11, November, 1929, A. M. Wahl, J. W. Bowley, and G. Back), the same author and two co-authors, J. W. Bowley and G. Back, record the results of a test in which actual strain measurements were made around the circumference of a 10-in. extra-strong pipe bend. The test results are compared with the theory developed in the previous paper and show good agreement with the correctness of the theory. Both test and calculations bear out the authors' contention that, because of permanent flattening in fabricating the pipe bends, it might be desirable to specify certain limits regarding the permissible initial ellipticity of the cross-section.

In connection with the flattening theory, it is evident that if a piece of pipe is partly bent and partly straight, as in the case of the square expansion loop, and the point of tangency is subject to a bending moment, the bent pipe will tend to flatten the adjoining straight section, and in turn the straight pipe will have a tendency to restrict the full flattening of the curved pipe. It would be of interest if the authors will clear up these points, since they would seem to influence the location of the point of maximum stress.

In conclusion, considering the several indeterminate factors which may be introduced during the fabrication of a pipe bend, such as ellipticity of cross-section, thinning of outside wall and fiber strains set up due to localized heating during bending, and the bending and torsional stress which may be introduced in the case of flanged bends at the time of installation due to imperfect alignment, all of which tend to have an additive effect on the bending stresses set up when the bend is installed and operating, it seems that, although a maximum stress for design purposes is suggested by the authors to cover both types of loop, a somewhat lower stress-safety factor might be used in laying out the square loop with seamless tube fittings, which has everywhere the full strength and true cross-section of the pipe in the line.

JOHN H. ZINC.[19] The writer is much interested in this paper and discussion because, as a piping contractor, it has been necessary for him to purchase and use both welded fittings and pipe bends. Our industry has accepted and is promoting pipe welding under a control specification supported by the best-known engineering standards. There is no question about it being a new and better type of pipe construction and takes with it the generous use of the tube-turn welded fitting covered by this discussion as well as others. The results obtained by the authors as to flexibility and pressure drop has been confirmed by our experience. It is believed that these tests should be continued to more accurately determine the value of pipe bends for compensation of expansion and contraction with particular reference to incorporated stresses as a result of the bending operation and their relation to the elastic limit of the pipe.

The fact that the tube-turn fitting has no incorporated stresses due to manufacture has prompted us to use them with straight pipe in fabricating our own welded expansion loops of rectangular design with more confidence as to flexibility, anchorage, and convenience of installation to meet our varying structural conditions. Further research in this direction will undoubtedly do much to correct the errors and differences of opinion now in evidence by our present empirical rules in applying values to pipe bends for expansion purposes.

It must not be inferred from this statement that the writer does not recognize that pipe bends have a place in our work, but he must disagree with the statement made by a previous discussor that they properly meet stress conditions and are cheaper. The question of cost will of course depend upon the design of the bend and competitive conditions, but, generally speaking, in our welded piping installations we are eliminating many fabricated pipe bends and using instead the welded fitting and straight pipe for bend construction because of greater flexibility and economy and, further, because we are enabled to control the pipe fabrication within our own shops or on the job itself.

May the writer also refer to the statement of a previous discussor wherein he recommended the use of fittings and straight pipe in making expansion bends as foolproof, cheaper in first cost, and representing an efficiency of only about 5 per cent less than other types. From a practical standpoint, the writer disagrees with him that the fitting expansion bend would be more foolproof because the threaded and flanged joint is not as good a guarantee of tightness, strength, and flexibility as the welded joint, to say nothing of the maintenance cost. As to first cost, in our own shop we can make a welded rectangular bend much cheaper by using a welded design with welded fittings than screw or flanged fittings, and the writer thinks this is generally true.

One representative raises an objection to the welded joint in expansion bends and states that in the design covered in the paper the maximum stresses are concentrated at the weld. It would be interesting to know the basis of this conclusion because it is at variance with the writer's opinion of stress distribution, but assuming he is correct, where else would it be better to concentrate stresses than at the weld, inasmuch as the weld should be equal to or is stronger than the pipe itself? The writer is of course supporting quality welding and passing out of consideration poor welding, as may be expected from workmen who are not able to qualify under similar specification requirements as laid down by our national association. We are prepared to deliver welded pipe installations to meet the most rigid requirements of pressure and temperature. It is the writer's opinion that welding will soon be recognized as the ultimate in pipe installation and that its use will demand a complete revamping of our present methods of determining design for compensation of pipe forces in service.

The writer trusts that his discussion will be accepted in the constructive way intended. He wishes to express his appreciation to the authors for the invaluable service they have rendered the industry, and strongly urges that, as indicated by them, further tests should be made to establish known quantities so that accurate and more readily understood rules for computing flexibility of expansion bends may be made available.

ANDREW A. BATO.[20] Of the points dealt with in the paper that concerning the frictional resistance is showing the longest history of investigations. This is probably because the fact is rather surprising that a greater radius of curvature in a bend does not, beyond a certain limit, mean less resistance, and this fact needs repeated confirmation for every case. Although this fact was discovered more than 25 years ago, the General Electric Company saw fit to conduct experiments to secure further data for the design of large crossover elbows of steam turbines, as published in *General Electric Review*, June, 1927, by Wirt. The results of the tests made with an 8-in. pipe and connected elbows of various shapes and designs, also with square ducts, with air up to 90 lb.

[19] President, The Heat and Power Corporation, Baltimore, Md.

[20] Consulting Engineer, East Orange, N. J. Mem. A.S.M.E.

pressure not only confirm the fact that greater radius does not necessarily mean less resistance, but reveal some new ones; for instance, that the best results could be obtained with an elbow the inner side or throat of which shows a round bend and the outer a sharp, mitered corner.

Before discussing the questions of flexibility and strength a modification in the method of calculating will be presented as it helps greatly in visualizing and demonstrating the various points. The writer showed this method three years ago, and it deserves to be demonstrated more in detail. It is based on the fact that, if the moment at the virtually free end of a pipe line (for instance, of that shown in Fig. 15 of the paper) and the force F acting at the same point are replaced by one single force, parallel to F, but shifted to a distance from the point j so as to give a moment at j equal to the original moment at that point, this shifted force F_M in Fig. 26 of this discussion paper will go through the so-called

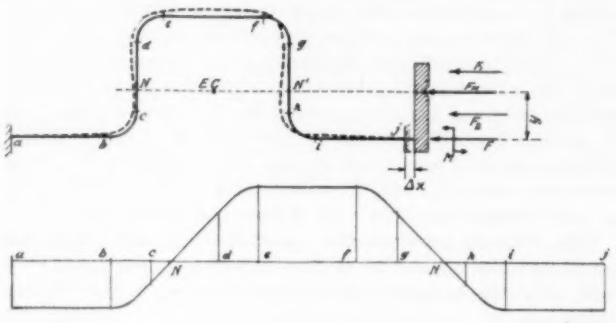


FIG. 26

elastic center EC of the pipe line. EC is nothing but the center of gravity of all elements dl/KEI; that is, of the elements of pipe length divided by the corresponding rigidity factors and modulus of inertia and elasticity. In case of a pipe line made of the same size and type of pipe in its whole length, EC will be very near to the center of gravity of the pipe line itself, the only deviation being caused by K being less than one for the relatively short elbows. The distance y to which the force F has been shifted is the one used in the paper; it is called "ordinate" or "coefficient" of the restraining moment, but it is nothing else but the arm of the force F_M, at the same time the distance of the elastic center for the line connecting the two ends of the pipe line. This can be proved by checking the formula given for y in Appendix No. 2 of the paper, and it can be proved in general for any pipe line held at both ends with the aid of well-known mathematical considerations.

The physical working of this method is visualized on Fig. 26, where the right hand or free end of the pipe line is held by an absolutely rigid arm on which F_M is acting in a distance y from the free end. F_M will have the same effect as F and M together; that is, it will move the point j in a straight line toward a and will at the same time keep the tangent to the piece of pipe ij parallel to itself in j, with no rotation. If F would be shifted into the positions F_1 or F_2, j would move in other directions, and the end at j would rotate. The case is similar to that of a coil spring which will bend out sideways, trying to get out of the way of the force, with the end rotating, unless it will be compressed by a force acting strictly in its axis.

With this modification all the pipe designers will have to do will be to find EC, a relatively simple operation that can be performed graphically, with not much chance for an error, and draw the line containing F_M. The magnitude of F_M itself can then be figured by using the F_M–EC line as the X axis for the system of coordinates. The moments and restraining moments for the end points as well as for any point can be found by multiplying F_M by the distance of the F_M–EC line from those points, which for the purpose of further calculations can be scaled off directly from a drawing, like that in Fig. 26. For the application of the area-moment method to calculate the rotations and deflections, the second diagram of Fig. 26 is drawn, corresponding to the second diagram of Fig. 15 of the paper, with the difference that the ordinates in Fig. 26 give directly the resulting moments. Clockwise above the a-j line and counterclockwise below that line, also the neutral point N where the moment is zero, while in Fig. 15 these ordinates have to be determined by adding and subtracting three lengths or items.

The greatest saving in labor is in the calculation of the rotations and deflections. Of all the figuring in Appendix No. 2, called by the authors "tedious" with right, about 40 per cent can be saved, as the moment areas that have to be added and deducted in Fig. 15 are shown with their net values in Fig. 26.

Furthermore, this presentation helps the designer to bear in mind certain simple rules he has to observe. For instance, the fact brought out by the authors that of the layouts shown in Figs. 17 and 19 the most flexible is the one in which the top of the bend is as long as the sum of the two bottom parts, and that in this case top and bottom are subjected to the same bending moment, can be visualized better if we bear in mind that the elastic center of the bend, in this case identical with the center of gravity of the line, will be just half-way between the top and the bottom of the bend; consequently, the force F_M passing through this center will give equal moments to top and bottom, and will make them work equally, utilizing them for flexibility to the same extent.

This result is also shown in Fig. 18 of the paper, and could be obtained without calculating a number of cases and drawing a curve, simply with the aid of the foregoing considerations.

This "force through elastic center" method points out sharply the importance of the strength of welded joints against bending; this because it shows that wherever the resulting bending moment is a maximum there is always a joint. A paper on the usability of welded joints for pipe lines subject to heat expansion could touch this point by giving figures on the strength of such joints as compared to that of flanged ones. Although the spreading of the use of welding seems to indicate that a welded joint compares favorably with a flanged one, as the costs of the necessary highly skilled labor seem to be offset by the savings on flanged fittings, industrial plants not using the services of firms specializing in piping work but laying their own piping should be warned against making such joints with ordinary steam fitters who are not trained especially in welding pipes, as according to tests quoted by Walker and Crocker in their book on piping both yield point and ultimate strength are about 22 per cent lower for the welded joint than for the pipe itself. On the other hand, joints made by special methods and by trained help proved to be stronger than the pipe.

A very general rule to insure greater flexibility can be arrived at by contemplating the equation giving the force F_M for bends of the discussed type:

$$F_M = \frac{EI\,\Delta x}{\int \frac{y^2 dl}{K}}$$

It is evident that the greater $\int \frac{y^2 dl}{K}$, where dl is the element of length—that is, the more pipe is laid at a possibly great distance from the F_M line—the smaller will F_M be. This is another reason why a square bend is more flexible than a U-bend, and not only

because welded fittings are used. For a fair judging, square bends made with welded joints should be compared with square bends of the same design and layout, but made with flanged joints. The only step in this direction indicated in the paper is a comparison between 90 deg. elbows with welded joints of standard good for 250 lb., which again offers no fair basis. This was probably done because flanged elbows of steel for 125 lb. are not readily obtainable, although they could be used where square expansion bends are preferred to U-bends, but specially skilled labor for welding is not available. For such cases flanged steel fittings may be of good service, as they are more foolproof and if combined with Van Stone connections of sufficient strength they can be readily installed by anybody.

To demonstrate the significance of the layout as compared with that of the seamless tube fittings, the writer figured out the force F acting at the end of the square bend Fig. 19 (*d*) of the paper for the case in which 90-deg. flanged turns are used instead of seamless ones. The turns were assumed to have the dimensions of standard turns, only made of cast steel instead of cast iron. The result obtained for F was 685 lb. as compared with 970 lb. for the double offset U-bend, Fig. 19 (*a*) and 548 lb. for the square bend, Fig. 19 (*d*). With other words for the case in question a 29.5 per cent decrease of F is due to the layout and a further decrease of 14 per cent to the use of the seamless fitting. Naturally the longer the straight sections of the pipe line will be in proportion to the turns the less will be the significance of the seamless fittings from the point of view of flexibility.

It is very hard to find a basis for comparing and demonstrating the significance of the ratio between the radius of the bend R and the pipe diameter d. Where the turn alone was used as such a basis, with a small piece of tangential piping, as in Fig. 12, while R/d changed from $1^1/_2$ to 6, the force necessary to bend it increased 100 per cent, while in the case when the turns were inserted into a square bend, Fig. 14, this increase was only 60 per cent, as the influence of the straight sections came into the foreground. The reader therefore is warned against generalizations on the basis of the paper, and is advised to use the very fine work of Shipman, presented two years ago, for more general information.

All in all, the authors did a nice piece of work in applying the scientific developments of the last few years to a practical problem.

G. K. Sauerwein.[21] This paper will prove of value to many users of piping who are just a little hesitant about the use of welded joints. Such a statement by a prominent piping engineer from a conservative utility should do much to dispel their fears.

That the friction loss in welded lines is less than in lines with screw joints needs no proof. The data given to show that the same is true as regards welded lines when compared with standard flanged construction are very assuring and helpful to the user, because it means saving in pumping power.

The authors have shown further by mathematical investigation that the use of welding elbows with proper design will also give greater flexibility to the lines. This, too, is a fact of great significance to the user. It affects the maintenance and life of the hot-water and steam lines. For our own purpose we arrived at the same conclusion after making some simple comparative tests.

The use of welding and of seamless tube elbows is already spreading at a rapid rate. Except in the most unusual cases the first cost of a job can be reduced by the use of welding. As the price of welding elbows falls by reason of increased use and willing labor becomes more skilful, the initial cost of welded piping will be still further reduced.

Much evidence is being accumulated which shows that maintenance expense is also less for welded lines than for lines with other types of joints. This is true for our own case at Harvard University when we have a great deal of piping in buildings, in tunnels and conduits, and in heating plants. On the older screwed and bolted piping, fittings are failing, and Van Stone lips are cracking off, gaskets giving out, flanges cracking, bolts failing, etc. Wherever practicable, welded work is replacing screwed or bolted work that gives out. It costs less to make a welded joint than to remake a gasketed and bolted joint.

On all of the new work now being done at the university, totaling in value about $20,000,000, welded piping is being specified for lines 2 in. and larger except for valves, in buildings, in tunnels, and in conduits, and also in the new Medicine School power plant. Welded piping in a power plant presents a very nice appearance.

A. E. R. de Jonge.[22] The writer would desire to join the others in thanking the authors for their very interesting paper. There is, however, one question he would like to have cleared up. The authors have confined themselves in their paper to the analysis of the elasticity and flexibility of plane pipe bends and their combinations and have not covered the flexibility of pipe lines in space. The writer would therefore ask Mr. Crocker if any experiments have been made on pipe lines in space and if these have been compared with the calculations by the method given in the "Piping Handbook" by Walker and Crocker.[23]

[Mr. Crocker answered this question in the affirmative and mentioned that the results of these experiments compared very well with the mathematical analysis as given in the "Piping Handbook."]

This is a very interesting statement by Mr. Crocker, as one would have expected considerable deviation of the test results from those of the analysis owing to the latter not being correct. It is always an unpleasant task to have to point out errors committed by others and doubly so if committed by men who by their past contributions have shown that they have the advance of the art very much at heart.

The authors' errors occurred in their mathematical analysis and make their results incorrect. This is to be deplored the more as they are contained in an otherwise very useful handbook, which probably will find its way to every engineer interested in pipe-line problems. Due to the reason that the spreading of this error is thus most difficult to check, the writer has felt forced to draw attention to it. The errors occur in the cases [drawing a sketch on the blackboard] of a 90-deg. pipe bend acted upon by a force perpendicular to the plane of the bend, by a moment acting perpendicular to the plane of the bend and tangentially to its free end and by a moment acting as a twisting moment on the free end, in which class there has been an error committed by interchanging the bending and twisting components whereby sine and cosine in the integrals have been reversed. Besides, in their elementary cases IX and X the authors have omitted one angular component of deviation completely, thereby getting a simple plane set-up for the determination of the components of reaction in the case of a space pipe line where they should get a space set-up.

The writer would add that he has made very careful experiments with bends and has found their results in good agreement with his formulas for these cases, but differing rather widely from the authors' formulas.

In referring to the whole problem of the flexibility of pipe lines, there is no necessity for calculating plane pipe lines according to the rather complicated method of the authors. As Mr. Bato has shown before, the question of plane pipe bends can be dealt

[21] Engineer, Lehman Hall, Harvard University, Cambridge, Mass. Mem. A.S.M.E.

[22] New York, N. Y.

[23] Published by the McGraw-Hill Book Company, New York.

with much easier by making use of the center of elastic weights of the pipe line as a whole. This method is not new at all and is well known to civil engineers who have been using it for the calculation of arched girders for several decades. This method becomes particularly simple if the theory of the ellipse of elasticity is used. By means of the ellipse of elasticity the solution of any plane piping problem can be readily accomplished, and the writer cannot understand why this method, to which he had drawn attention at the meeting of the Society in 1928,[24] is not used more extensively by American engineers. This method presents an almost purely graphical analysis which can be easily carried out on the drafting board.

When we, however, go into space the problem becomes very much more complicated. Here the elasticity ellipse changes into an ellipsoid of elasticity. If the pipe line has in any way a symmetrical set-up, the axes of the resulting ellipsoid can be easily obtained, and the solution of the flexibility problem presents little difficulty. If, however, the set-up is unsymmetrical, as in most practical cases, the ellipsoid lies oblique in the space of the pipe line, and the problem has to be solved to find the principal axes of this oblique ellipsoid. This is a problem of the third degree and not of the second. Thus the solution cannot be tackled by means of compass and ruler solely, if a graphical solution is chosen, nor by quadratic equations, if the analytical

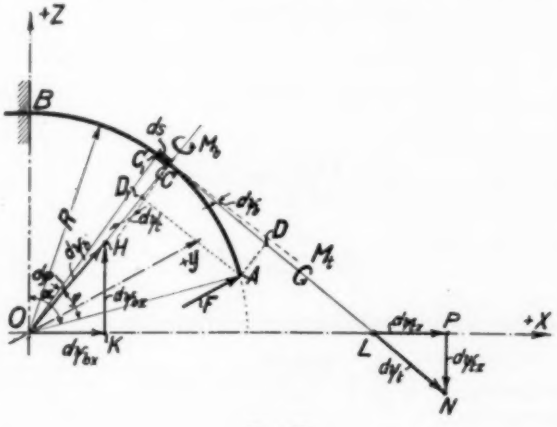


FIG. 27

method of deduction is adopted. In the former case use has to be made of other curves (hyperbolas and ellipses) and in the second case of a cubic equation. This explains why the problem of space pipe lines has been such a difficult one to solve. The writer arrived at the solution of this problem some time ago, and had hoped to be able to present it at this meeting, but was not able to complete the paper.

GENERALIZED FORMULAS AND THEIR DERIVATION

Having been advised by the authors that they propose to publish the derivations of the formulas for their cases VIII, IX, and X ("Piping Handbook") as revised by Dr. Timoshenko in their closure, the writer herewith submits his own formulas, which will be found to embrace several other cases besides the 10 fundamental cases of the authors. As there are many occasions where 30-, 45-, or 60-deg. and even $22^1/_2$-deg. bends are used, it may be convenient to have the necessary formulas handy, and it was for this reason that the writer has derived formulas of a general nature covering these cases.

The derivation of these general formulas is simple and will be given for the case of a bend of an angle α fixed at one end and acted upon at the free end by a force F perpendicular to the plane of the bend.

In Fig. 27 let AB be the bend fixed at B; R the radius of the center line of the bend; CC_1 an element ds; α the angle subtended by the bend; φ the angle AOC; O the center of the bend; $d\varphi$ the angle subtended by the element ds; CDL the tangent of the bend in C; AD a perpendicular from A onto the tangent; AD_1 a perpendicular from A onto the radius OC; F the force acting in A perpendicular to the plane of the bend. Further, let OX, OY, and OZ be the axes of a right-hand-screw space reference system, OY being perpendicular to the plane of the bend.

The force F acting on the lever AD_1 produces a bending moment M_b at C about the radius OC, the rotational direction of which moment is indicated by a circular arrow around the produced radius OC. The reaction F at C producing pure shear in the element ds is of little importance and will be neglected.

Likewise the force F acting on AD produces a twisting moment M_t at C about the tangent CD, the rotational direction of which is indicated by a circular arrow around the tangent CD. Thus the bending moment acting at C is:

$$M_b = FR \sin \varphi \quad \ldots\ldots [1]$$

and the twisting moment acting at C is:

$$M_t = FR(1 - \cos \varphi) \quad \ldots\ldots [2]$$

These moments produce deformations of the element ds.

The angular rotation of the part CA of the bend due to the bending moment, considering ds elastic only, is:

$$d\psi_b = \frac{M_b ds}{EI} \quad \ldots\ldots [3]$$

and that due to the twisting moment:

$$d\psi_t = \frac{M_t ds}{GJ} \quad \ldots\ldots [4]$$

where E = modulus of elasticity
G = modulus of rigidity (shear)
I = plane moment of inertia of pipe cross-section, and
J = polar moment of inertia of pipe cross-section.

Further there is:

$$G = \frac{E}{2(1 + \mu)} \quad \ldots\ldots [5]$$

where μ is Poisson's ratio, which may be taken for steel as $\mu = 0.3$. Since for circular cross-section

$$J = 2I \quad \ldots\ldots [6]$$

then

$$GJ = \frac{EI}{1 + \mu} = \frac{EI}{1.3} \quad \ldots\ldots [7]$$

Substituting this value into Equation [4],

$$d\psi_t = (1 + \mu) \frac{M_t ds}{EI} = 1.3 \frac{M_t ds}{EI} \quad \ldots\ldots [8]$$

Since these angular rotations do not lie in the same plane they cannot be composed algebraically. They can be represented, however, by vectors, using a right-hand screw system throughout, the vectors pointing in such a direction that the rotation, if viewed in the direction of the vector, takes place clockwise or in a "right-hand" sense.

Having established this relation, it is an easy matter to find the correct direction of rotation, the vectors or their components parallel to the reference axes being counted positive if acting in

[24] A.S.M.E. Trans., vol. 51, 1929, APM-51-18, p. 210.

the direction of the positive axes and negative if of opposite sense. Components in the same axis or parallel to it can then be added algebraically.

Thus the rotation due to bending $d\psi_b$ may be represented by the vector $\overrightarrow{OH}$, and the rotation $d\psi_t$ due to twisting by vector $\overrightarrow{LN}$, both being correct as to sense.

The components of these rotations along OX and OZ are:

$$\left.\begin{aligned} OK &= d\psi_{bx} = d\psi_b \sin(\alpha - \varphi) \\ LP &= d\psi_{tx} = d\psi_t \cos(\alpha - \varphi) \end{aligned}\right\} \quad \ldots\ldots\ldots [9]$$

$$\left.\begin{aligned} KH &= d\psi_{bz} = d\psi_b \cos(\alpha - \varphi) \\ PN &= d\psi_{tz} = -d\psi_t \sin(\alpha - \varphi) \end{aligned}\right\} \quad \ldots\ldots\ldots [10]$$

The two Equations [9] added give the total rotation around the X-axis and the two Equations [10] likewise that around the Z-axis. Thus,

$$\begin{aligned} d\psi_x &= d\psi_{bx} + d\psi_{tx} \\ &= d\psi_b \sin(\alpha - \varphi) + d\psi_t \cos(\alpha - \varphi) \end{aligned}$$

or with Equations [3] and [8],

$$= \frac{M_b ds}{EI} \sin(\alpha - \varphi) + (1 + \mu) \frac{M_t ds}{EI} \cos(\alpha - \varphi)$$

and substituting from [1] and [2] gives

$$d\psi_x = \frac{FR^2}{EI} \sin\varphi \sin(\alpha - \varphi)\, d\varphi + (1 + \mu) \frac{FR^2}{EI} (1 - \cos\varphi) \cos(\alpha - \varphi)\, d\varphi,$$

If $ds = Rd\varphi$; and by making

$$\frac{FR^2}{EI} = C \quad \ldots\ldots\ldots\ldots\ldots [11]$$

$$d\psi_x = C[\sin\varphi \sin(\alpha - \varphi) + (1 + \mu)(1 - \cos\varphi)\cos(\alpha - \varphi)]\, d\varphi$$

Integrating this equation over the length AB of the bend, or over the angle α:

$$\psi_x = C \int_0^\alpha [\sin\varphi \sin(\alpha - \varphi) + (1 + \mu)(1 - \cos\varphi)\cos(\alpha - \varphi)]\, d\varphi$$

or

$$\psi_x = C \cdot \left(1 + \frac{\mu}{2}\right) [\sin\alpha - \alpha\cos\alpha]$$

and substituting for C its value in [11], we find for the total rotation around the X-axis:

$$\psi_x = \frac{FR^2}{EI} \left(1 + \frac{\mu}{2}\right) [\sin\alpha - \alpha\cos\alpha] \quad \ldots\ldots [12]$$

In a similar manner we find the rotation around the Z-axis. From Equations [10], [11], [1] to [3], and [8],

$$\begin{aligned} d\psi_z &= d\psi_{bz} + d\psi_{tz} \\ &= C[\sin\varphi \cos(\alpha - \varphi) - (1 + \mu)(1 - \cos\varphi)\sin(\alpha - \varphi)]d\varphi \end{aligned}$$

and by integration over the whole length of the bend:

$$\psi_z = C \int_0^\alpha [\sin\varphi \cos(\alpha - \varphi) - (1 + \mu)(1 - \cos\varphi)\sin(\alpha - \varphi)]d\varphi$$

whence

$$\psi_z = \frac{FR^2}{EI} \left[\left(1 + \frac{\mu}{2}\right) \alpha \sin\alpha (1 + \mu)(1 - \cos\alpha)\right] \ldots [13]$$

For the displacement of A (free end) in the direction of the force we find the deflection:

$$d\Delta = d\psi_b \cdot R \sin\varphi + d\psi_t R(1 - \cos\varphi)$$

or using Equations [1], [2], [3], [8], [10] and putting $ds = Rd\varphi$,

$$d\Delta = RC[\sin^2\varphi + (1 + \mu)(1 - \cos\varphi)^2]d\varphi$$

and by integration over the whole length of the bend from A to B:

$$\Delta = RC \int_0^\alpha [\sin^2\varphi + (1 + \mu)(1 - \cos\varphi)^2]d\varphi$$

whence

$$\Delta = \frac{FR^3}{EI}\left[\left(2 + \frac{3}{2}\mu\right)\alpha + \frac{\mu}{2}\sin\alpha\cos\alpha - 2(1 + \mu)\sin\alpha\right] [14]$$

Equations [12], [13], and [14] are the general expressions for a bend of any angle α. If we put $\alpha = \frac{\pi}{2}$ and $\mu = 0.3$, we obtain the equations for the quarter-bend, thus:

$$\psi_{x_{90}°} = 1.15 \frac{FR^2}{EI} \quad \ldots\ldots\ldots\ldots [12a]$$

and for:

$$\psi_{z_{90}°} = \frac{FR^2}{EI}\left[1.15 \frac{\pi}{2} - 1.3\right]$$

or

$$\psi_{z_{90}°} = 0.5064 \frac{FR^2}{EI} \quad \ldots\ldots\ldots\ldots [13a]$$

Also

$$\Delta_{90°} = \frac{FR^3}{EI}\left[2.45 \frac{\pi}{2} - 2.6\right]$$

or

$$\Delta_{90°} = 1.24846 \frac{FR^3}{EI} \quad \ldots\ldots\ldots\ldots [14a]$$

From these equations we find the radii of rotation:

$$r_x = \frac{\Delta}{\psi_x} = \frac{1.24846}{1.15} R = 1.0856R \quad \ldots\ldots [15a]$$

and

$$r_z = \frac{\Delta}{\psi_z} = \frac{1.24846}{0.5064} R = 2.4654\, R \quad \ldots\ldots [16a]$$

Similarly, for a 45-deg. bend, $\alpha = \frac{\pi}{4}$ and $\sin\alpha = \cos\alpha = \frac{\sqrt{2}}{2}$,

$$\psi_{x_{45}°} = 0.1745 \frac{FR^2}{EI} \quad \ldots\ldots\ldots\ldots [12b]$$

$$\psi_{z_{45}°} = 0.05287 \frac{FR^2}{EI} \quad \ldots\ldots\ldots\ldots [13b]$$

$$\Delta_{45°} = 0.16075 \frac{FR^3}{EI} \quad \ldots\ldots\ldots\ldots [14b]$$

$$r_x = 0.9212\ R \quad \ldots\ldots\ldots\ldots [15b]$$

$$r_z = 3.0404\ R \quad \ldots\ldots\ldots\ldots [16b]$$

The equations developed above form a generalized case corresponding to the authors' case VIII. In a similar manner all other cases can be treated. If one does not confine oneself to quarter-bends, and the writer has not been able to do so in his practice, then 24 general cases result, from which all specific solutions can be obtained by simple substitution of the proper angle of the bend and its trigonometric functions.



FIG. 28 GENERAL CASES FROM WHICH SPECIFIC SOLUTIONS CAN BE OBTAINED BY SUBSTITUTION OF THE PROPER ANGLE OF BEND

Assuming the angular bend to be fixed to a vertical plane, four cases occur:

1 The coordinate system may pass with one of its axes through the fixed end of the bend.

2 The coordinate system may pass with one of its axes through the free end of the bend.

3 The forces and moments may act at the free end of the bend parallel and perpendicular to the fixing plane.

4 The forces and moments may act at the free end of the bend parallel and perpendicular to the free flange.

In all we get thus 24 general cases, which are given in Fig. 28. These cases allow of the numerical calculation of all pipe bends and also of pipe lines if their end reactions are known. It is assumed that the latter have been found. The writer will show how they can easily be found in a later communication.

The factor K in the formulas is the rigidity factor of the pipe section.

AUTHORS' CLOSURE

The discussion has brought forth much of interest regarding pressure drop and flexibility in piping. That certain points which seemed clear to the authors actually need amplification was demonstrated by several additional examples presented in the discussion, either to bring out new phases of the problem, or to develop points which were not explained in sufficient detail. In this connection the authors would again call attention to their statement in the conclusion to the paper that they had chosen typical specific cases for flexibility analysis to show the general trend of such comparisons and how other cases could be worked out, rather than attempting to draw sweeping conclusions. Other points outside the original scope of the paper have been introduced in the discussion and require explanation in this closure.

It was to be expected, of course, that some conclusions drawn by the authors might be misconstrued, or would not be accepted by all. This is apt to be the case when dealing with a new type of construction, especially one involving welding, which is still in a stage of development, involving more or less controversy between enthusiasts for welding and those distrustful of its use. The authors have striven to present the facts as they saw them in an impartial way, calling attention to the advantages or limitations of the product under discussion, and pointing out how to minimize these limitations in laying out pipe lines. In some instances identical points have been raised in two or more discussions and can best be taken up collectively in the closure, while other points require individual attention.

Welded Construction. Much has been said for and against the use of welded construction. Some are against it on the ground that it is a new and relatively unperfected art; others oppose it as upsetting the established use of screwed or flanged fittings and screwed or Van Stoned pipe. Both of these objections are bound to be overcome by time and progress—those who oppose welding now may be its earnest advocates five or ten years hence. An interesting exposition of this point is quoted from an editorial in *Metal Progress* for January, 1931:

It is recorded that the first use of iron pipe (other than cast iron) occurred in 1815, when Murdock promoted a gas-lighting plant for London. The inventor collected old musket barrels, which he screwed together to form continuous tubes reaching from his coal-gas generators to the burners.

Suppose, if you can, that modern equipment for fusion welding had been available to Murdock a hundred years ago, and to the inventors who developed butt-welded and lap-welded iron pipe—not only available, but well enough known so a welded joint was the standard joint in 1887 (when the first steel pipe was made).

In other words, picture a situation where piping or plumbing contractors had acquired long experience with the oxyacetylene cutting and welding blowpipe and had organizations well skilled in the use of these tools. Architects and designers would be familiar with the properties and capabilities of welded joints; pipe and valve manufacturers would be producing lines of goods especially designed for welded joints.

Suppose such a piping contractor in such an age had been approached by a salesman with this new idea: That pipe joints should be made by screwing together instead of welding. How far do you suppose he would get in his argument?

Here are the "advantages" which such a change would involve:

In the first place, the all-purpose tools so familiar (the cutting and welding equipment) would be superseded by hacksaws, pipe cutters, and a series of thread-cutting dies and die holders, a different die for each diameter of pipe to be used.

In the second place, the contractor would be invited to purchase and carry in stock a large number of ells, tees, branches, reducers, plugs, flanges, bolts and gaskets—a complete supply for each diameter of pipe in use, multiplied by two if light-pressure and heavy-pressure work were in prospect.

Third, the owner of the installation would be asked to assume the added expense of buying pipe heavier than really necessary to carry the pressure—pipe with walls thickened from end to end to give enough stock for the threading at the joints.

For a "bottle tight" joint, industry would get a mechanical fitting with many chances of leakage—a joint which, if by excellent workmanship is originally tight, is certain to be the source of eventual trouble, either through the loss of valuable contents of the pipe or by damage to surrounding property by escaping liquid.

Lastly, the prospective purchaser of the screwed joint would be asked to endure multiplication of work, expense, and time when additions or alterations to existing pipe systems were to be made.

To balance these inherent characteristics and disadvantages of the screwed pipe joint, what could be offered in compensation to an industry, a firm or a workman already possessing welding equipment and welding skill?

The authors do not imply that fusion welding as done in the field has reached a state of complete perfection, but the process has made rapid strides in the past few years and those who do not employ it to some extent at least are not keeping abreast of the times. As pointed out in the quotation, the practice of making the whole length of each piece of threaded pipe thick enough to stand threading for a short distance at each end is



FIG. 29 TYPICAL EXAMPLES OF FAILURE FROM CORROSION OF THREADED PIPE ENDS

extremely wasteful of material. On the other hand, if the extra thickness of pipe wall between the threaded ends is required to resist failure from corrosion, why deliberately create weak spots in the structure by threading the joints? The tendency for failure in the threaded portions of standard-weight pipe operating even at low pressures such as obtain in building heating and plumbing systems is not widely enough appreciated. Fig. 29 shows typical examples of failure from corrosion in such piping. The three specimens at the left are "black" steel pipe taken from a "dry" return line in a low-pressure building-heating system after about five years of service. The threaded portions completely rusted through were on the lower side of horizontal runs where condensation washed along more or less continuously. The upper portions, which were not so continuously immersed, were not so badly affected. The two specimens at the right are galvanized steel pipe taken from the city water piping in a residence after ten years use. All specimens were removed from service because of leaks in the threads. The remaining wall thickness in the unthreaded portions of all specimens would have been ample for many more years of service had failure not occurred in the threads. While these specimens which happened to be available at the time represent relatively small nominal sizes of pipe, the same observations hold true for larger sizes which lend themselves more readily to welding. These observations are offered as indicating superior corrosion-resisting properties for welded construction over screwed for pipe of the same material and wall thickness.

Concern has been expressed in the discussion regarding bending moments and stress concentrations at welds. The authors agree that this is a matter worth serious consideration, especially where service conditions involve high temperature or high pressure or both. While it is well known that the strength of good fusion welds frequently exceeds the strength of the normal pipe cross-section, especially where a bead is built up at the weld, this favorable strength relation does not exist, of course, with poorly made welds. The problem therefore resolves itself into controlling the process and the personnel so as to obtain uniformly satisfactory results. Present practice seems to indicate that a consistent weld efficiency of 75 to 80 per cent is within reason to expect. This is borne out by the tentative rules laid down for fusion-welded construction in the A.S.M.E. codes for the construction of boilers and unfired pressure vessels which allow a fusion-weld efficiency of 80 per cent. This efficiency compares to advantage with the relation set up in the Boiler Code between low-carbon seamless steel pipe (allowable stress by formula, 9000 lb. per sq. in.) and lap-welded steel pipe (allowable stress, 7000 lb. per sq. in.), which gives an efficiency for a furnace-welded pipe of $7000 \div 9000 = 0.78$, or 78 per cent. On this basis fusion-welded pipe joints would be considered equal to the lap-welded pipe which is commonly used in moderate-pressure pipe bends for temperatures not exceeding 750 deg. fahr. It is to be expected that as the technique of fusion welding is perfected and more confidence is placed in its reliability, the application of welded construction to piping operating at higher pressures and temperatures will be a matter of course. As an example of what is being done at the present time, high-pressure piping in the power and central heating plants of the Consolidated Gas, Electric Light, and Power Company of Baltimore, Md., is operating satisfactorily with welded joints. Welds for pressures above 350 lb. gage are made up with reinforcing sleeves, and the turns are made with long-radius bends having tangents on the ends to accommodate the sleeves.

Criticism also has been directed at built-up bends on the ground that joints or welds at points of maximum bending moment can be avoided by the use of conventional types of expansion U-bends made in one piece. In a limited sense this is true, but in many actual pipe lines flexibility is obtained by taking advantage of directional changes in the lines rather than through the insertion of expansion U-bends. Where expansion is taken up by the spring in runs of pipe at right angles to one another, it usually is impossible to prevent high bending moments from being applied at some of the joints. Under such circumstances long-radius bends have no advantage over seamless-tube fittings as far as avoiding bending moments at joints goes. The authors employed expansion U-bends in their comparisons as representing simple cases where the analysis could be followed easily, but not with the idea of advocating the replacement of such bends by built-up bends where the former could be used to advantage.

Question has been raised regarding concentration of stress at welds due to the absence of tangents at the ends of seamless-tube elbows as made at present. This may appear objectionable from theoretical considerations, but so far as the authors are aware no actual failure has been attributed to the absence of such tangents. The construction without tangents possesses certain advantages from a manufacturing standpoint in that the fittings can be made as 180-deg. bends and then cut off into forty-fives, nineties, or into any desired angle as required. From a design and installation standpoint, the absence of tangents is desirable in that it provides the maximum radius of curvature for a given center-to-end dimension. While the fittings described by Messrs. Houser and Hirschberg as being forged in halves and welded together

possess tangents at the ends, it would appear that this feature is offset by the presence of two longitudinal welds and a shorter radius of curvature. Consideration should be given the fact that hoop tension in a pipe wall due to internal pressure is for practical purposes exactly twice the longitudinal stress in the wall due to the same pressure. From this it follows that longitudinal welds are subject to twice the bursting stress existing in circumferential welds. If transverse stresses due to flattening of the cross-section of a curved tube during flexure are considered, high stress concentration may occur in the longitudinal welds. This condition should be taken into account in setting safe working stresses for welds in the two positions, bearing in mind that the conditions under which a weld is made also affect its reliability.

The convenience of having ready-made tees, crosses, and the like manufactured by Mr. Houser's method is attractive, and there is bound to be extensive use for both these and seamless-tube elbows. Failure to mention welded fittings forged in halves was not intentional, as they are a recent development which was not announced until after presentation of the paper. It would seem also that there was use for standard sizes of long-radius bends made up with tangents on the ends for welding.

Frictional Resistance. Several engineers discussing the paper have intimated that the authors stated the frictional resistance of seamless-tube elbows having a radius of $1^1/_2$ pipe diameters was less than for long-radius pipe bends. Such is not the case as the authors merely strove to point out that the data indicated not much was gained by using a radius greater than $1^1/_2$ to 2 pipe diameters. Professor Eustice's curve offered by Messrs. Houser and Hirschberg in Fig. 23 tends to support this statement, and at the same time explains in a most interesting manner the observations of certain earlier investigators regarding fluctuations in pressure drop for radii exceeding $2^1/_2$ diameters. Williams, Hubbell, and Fenkell reported having observed the same phenomenon for large pipes in their classical paper from which the following is quoted:[25]

> *Curvature and Other Effects.* The foregoing experiments, taken as a whole, prove beyond question or doubt that the hitherto accepted notions of the laws governing curve resistance are wholly in error. As was suggested in the discussion of the velocity curve, the resistance due to curvature is proved to be very much less in the short than in the long-radius curves for the same total length of pipe and angle of deflection. From the experiments upon the 30-in. pipe, as well as those on the 12-in., it is apparent that the 90-deg. curve introducing the least resistance in a total length of 80 diameters is one of between 2 and $2^1/_2$ diameters radius.

The test-data curves of Williams, Hubbell, and Fenkell show increasing resistance with increasing radius from 2 or $2^1/_2$ diameters up to a secondary maximum point at 6 or 8 diameters, and then continue with an undulating shape similar to that obtained by Professor Eustice. The authors are indebted to Messrs. Houser and Hirschberg for calling attention to the most interesting and valuable exposition of this phenomenon by Professor Eustice which had escaped their notice. This offers a logical explanation for an observed flow characteristic the reason for which previously was not understood as being due to periodical changes in direction of tangential flow vortices. Professor Eustice's curve (Fig. 23) shows a minimum point at $R/d = 2.91$ and a secondary maximum at 6.85, the general shape of the curve bearing out the authors' observation that pressure drops decrease rapidly as R/d ratios increase from less than unity to $1^1/_2$ or 2, and that little is to be gained by using ratios greater than $1^1/_2$ or 2.

The following plausible and clearly stated explanation of the reasons for an increase in bend friction for R/d ratios greater than 2 to 3 was offered by J. L. Campbell[26] in discussing the Williams, Hubbell, and Fenkell paper:

> Undoubtedly, with a given velocity, the intensity of disturbance, and consequently of resistance, in a sharp turn is greater than in a gentle turn; but with equal angular deflections it acts through a shorter distance. As this greater intensity of the shorter turn multiplied by its lesser distance is, apparently, smaller than the lesser intensity of the long turn multiplied by its greater distance, the larger resistance of the latter is clearly due to the continued agitation of the water in the long curve.

The foregoing statement coupled with the explanation of Professor Eustice offers a rather satisfactory analysis of the sinuous-curve shape of Fig. 23. The greater number of times the vortices change direction in the longer-radius curves of Fig. 23 is incidental to the "continued agitation of the water in the long curve." The fact that with certain combinations the vortex emerges tangent to the bend and with others at an oblique angle accounts for the maxima and minima points on the curve. Eventually, as the radius of the bend increases, a point is reached where disturbance due to curvature is so slight that the overall pressure drop is less than the minimum occurring with medium radius bends of 2 to 3 diameters.

In referring to the tests of Professor Giesecke, Messrs. Houser and Hirschberg assign an R/d value of 4 to 3-in. "standard" long radius elbows. It is not clear from either Professor Giesecke's paper or the writers' statement just what a "standard" screwed long-radius elbow is. Apparently the only American standard approaching such a description is that for 150-lb. Long-Turn Sprinkler Fittings B16g-1929. Examination of this standard does not indicate any such favorable relation as $R/d = 4$ for long-radius screwed elbows. The R/d values based on "Radius of Sweep" dimensions taken from this standard compare as follows with the corresponding R/d values for 125-lb. standard screwed elbows as computed by the authors and shown in Table 1 of the paper:

R/d RATIOS FOR 125-LB. STANDARD ELBOWS AND 150-LB. LONG-RADIUS SCREWED ELBOWS

Nominal size, in.....	$2^1/_2$	3	$3^1/_2$	4	5	6
Standard.....	0.7	0.7	0.7	0.7	0.7	0.6
Long radius.....	1.4	1.3	1.3	1.3	1.2	1.2

Assuming that the above values truly represent curvature relations of fittings tested by Professor Giesecke, it would appear that his long-radius elbows have an R/d ratio of 1.3 instead of 4 and that the conclusion of Messrs. Houser and Hirschberg regarding the existence of a considerable difference between these tests and the curves of Fig. 11 is in error. In any event the foregoing comparison serves to emphasize the fact that long-radius elbows are, in the main, inferior to seamless-tube elbows as regards pressure drop.

In Fig. 24 Messrs. Houser and Hirschberg compare two right-angle turns of the same center-to-face dimension, one made with a seamless-tube elbow and the other with a long-radius bend. To carry this comparison through to a definite conclusion, the authors have assumed 12-in. pipe with $R/d = 1^1/_2$ for the seamless-tube elbow and $R/d = 5$ for the long-radius bend. On this basis 7.85 linear feet of pipe are required for the long-radius bend as against 9.36 feet for the seamless-tube elbow with tangents welded on. This corresponds to an excess of 19 per cent in the linear feet of pipe required for the latter. To get at the relative frictional resistance for the two types of bends, assume the equivalent resistance of the seamless-tube elbow as 12 diameters (12 ft. in this case) and that of the long-radius bend as 0.8 the resistance of the former, or 9.6 diameters (or feet). The total equivalent lengths for computing the resistance of the two bends then are:

[25] "Experiments at Detroit, Mich., on the Effect of Curvature Upon the Flow of Water in Pipes," by Gardner S. Williams, Clarence W. Hubbell, and George H. Fenkell, Trans. A.S.C.E., vol. 47 (1902), paper no. 911, p. 183.

[26] Trans. A.S.C.E., vol. 47 (1902), p. 221.

Seamless-tube elbow assembly	9.36	+	12.0	= 21.36 ft.
Long-radius pipe bend.....	7.85	+	9.6	= 17.45 ft.
Difference.....				3.91 ft.

The increased resistance of the seamless-tube elbow assembly over the long-radius bend is 3.91 ÷ 17.45 = 0.222, or 22.2 per cent. Considering bends as only a part of a complete piping system which is made up largely of straight pipe, it seems safe to say that the resistance of the whole system should not be increased more than 10 per cent by the use of the shorter-radius bends. In fact, if there are globe valves, stop and checks, turbine throttles, or similar valves in the line, the percentage increase chargeable to the use of seamless-tube elbows may be less than 10 per cent.

A one-line drawing of a 12-in. line approximating conditions

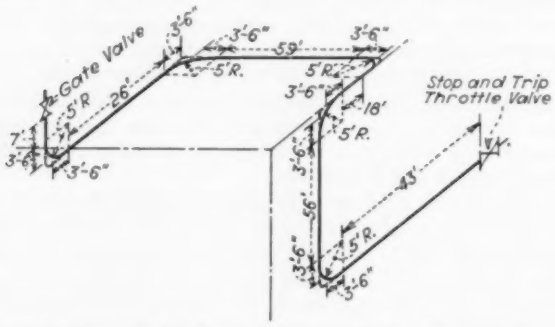


FIG. 30 TYPICAL PIPE LINE IN SPACE (12-IN. TURBINE LEAD)

for an actual turbine lead is shown in Fig. 30. The offsets in this line were required to drop from the steam-header elevation to the turbine-throttle level, and to clear the building structure and mechanical equipment. Assuming a steam flow of 350,000 lb. per hour at a pressure of 380 lb. gage and a temperature of 700 deg. fahr., the pressure drop computed by Fritsche's formula is approximately 3 lb. per 100 ft. In the case of long-radius bends the pressure drop due to change in direction is assumed to be 0.8 that of the seamless-tube elbows, although whether this relation holds true at such high velocities is uncertain. The equivalent lengths of the line with the corresponding pressure drops, including valves, are shown below for (*a*) turns made with long-radius pipe bends, and (*b*) turns made with seamless-tube elbows.

	R/d	Equivalent length, ft.	Pressure drop, lb. per sq. in.
(*a*)	5	403	12.09
(*b*)	1½	422	12.66

From which it appears that in this instance the excess friction loss in the line made up with seamless-tube elbows is less than 5 per cent greater than that in the line made up with long-radius bends.

The authors have not attempted to draw cost comparisons in the paper, but it is pertinent to point out that the 19 per cent excess in linear feet of pipe required in their analysis of Fig. 24 (which includes the seamless-tube elbow itself) plus the cost of making four welds would be balanced against the relative cost of fabricating a seamless-tube elbow vs. a long-radius bend, plus the cost of two pairs of flanges with their bolts and gaskets. On the other hand, if the long-radius bend is to be connected in the line by welds, there will be a reduction of one or two in the total number of welds required, which saving should be balanced against the convenience of using a ready-made seamless-tube elbow. Exception is taken to the generalized cost inferences drawn by Mr. Stuart on the grounds that they do not refer to any definite construction details and are not supported with cost data of any sort.

In commenting on Mr. Tanner's discussion, the authors would say that his analysis of the relative friction loss in bends having R/d ratios of 1½ and 5 affords an interesting comparison for design purposes. Attention is called to an error on his part in computing the relative losses from the Buffalo Forge Company's curve. Referring to the original curve in that company's handbook "Fan Engineering," which is drawn to a larger scale than the curve reproduced in Fig. 8 of this paper, it appears that the correct ratio is 81 per cent rather than the 72 per cent stated by Mr. Tanner. Based on a rough average from the curves of Figs. 8, 10, 11, and 23, the authors would suggest for design purposes that the resistance of long-radius pipe bends (R/d = 5 to 6) be considered as approximately 80 per cent of the resistance of seamless-tube elbows (R/d = 1½). This suggestion is offered as furnishing a working value for the time being until more complete data embracing more fluids, pipe sizes, and velocities are available. In this connection the authors would question the authority for Mr. Tanner's statement that "For low velocities the difference in loss between 1½ and 5 diameters is small, but for steam at high velocities it becomes appreciable as shown by all observers." As previously stated by the authors, they are not aware of any comprehensive tests on the pressure drop of steam flowing through bends of different radii.

In discussing the flow of air through elbows of different radii Mr. Cramer has assumed that the resistances given in Figs. 9 and 10 include an allowance for the measured length of travel along the center line of the bends which was not included in the curves for water flow in Fig. 11. Subsequent review of this point by the authors indicates that Mr. Cramer's assumption is in error and that the measured length of travel should be added to the bend resistance in each case. This practice is followed in most texts and papers on ventilation, although treatment of the subject often is not as clear as might be desired. The authors referred Mr. Cramer's interpretation of Figs. 9 and 10 to W. D. Edwards, who was the original source of these data, and received the following explanation:

> The linear feet of pipe, including the center line length of the bend, is separated from the resistance offered by the shape of the bend, or in other words, in my piping length I would measure and include this center line, to which I would add the bend resistance factor. This is in opposition to the interpretation of Mr. Cramer.

However, in the case of Fig. 8, which is reproduced from "Fan Engineering," subsequent inquiry indicates that in the opinion of the engineers of the Buffalo Forge Company and of the Carrier Engineering Corporation, the measured length of travel along the center line of the bends is included in the curve for pressure loss in bends presented by the late Frank L. Busey.

The authors regret that this distinction was not made clear in presenting their paper, and thank Mr. Cramer for calling attention to the oversight so that it can be corrected. Taking into account the measured length of travel gives the following tabular relation which is offered in lieu of that presented in Mr. Cramer's discussion:

R/d.....	0.50	0.75	1.00	1.25	1.50	2.00	3.00	5.00
Measured length in diameters = $2(5 - 0.215R/d.)$	9.79	9.68	9.57	9.46	9.36	9.14	8.71	7.85
Elbow resistance in diameters.....	121.20	35.09	17.51	12.72	10.36	9.03	8.24	7.85
Total equivalent length in diameters.....	130.99	44.77	27.08	22.18	19.72	18.17	16.95	15.70

FLEXIBILITY

General. In considering the discussion dealing with flexibility the authors feel that some agreement is needed regarding what constitutes an "expansion bend." The conventional expansion and double-offset expansion U-bends as shown in Figs. 17(*a*)

and 19(*a*) find appropriate application where there is a long run of pipe in one direction. Where there is opportunity to place sections of the line at right angles to one another it usually is possible to utilize the inherent flexibility of the line to good advantage and dispense with "expansion bends." An arrangement of this kind shown in Fig. 30 is typical of such a layout fitted in to clear structural and mechanical interferences in a power plant. Such layouts are far more common in steam power plants than those employing "expansion bends" for flexibility. Obviously the turns can be made with long-radius quarter-bends, seamless-tube fittings, forged fittings, or cast fittings as the designer prefers, and assembled with flanged or screwed joints or by welding. The authors discussed conventional types for ease in comparison, leaving it to the individual designer to use the methods indicated and make his own choice of layout.

Grapho-Analytical Method. The grapho-analytical method used in presenting information regarding the relative flexibility of symmetrical bends was employed to demonstrate its application in the simpler problems, and to suggest the manner in which it can be applied to problems in space. There are several alternate methods which can be applied successfully to symmetrical shapes in one plane such as those considered in this paper, but the alternate methods have definite limitations which render them unsatisfactory for application to pipe lines in space. The grapho-analytical method and the basic relations of the various elements of a pipe line which have been developed in connection with it, enable the designer to determine as accurately as he desires the relations between deflection and force throughout a complex piping layout.

Since in the application of the grapho-analytical method the ends of the run of piping under investigation are assumed as fixed, the sum of the bending-moment areas from one end of the structure to the other is zero. This fact enables the designer to check the numerical accuracy of his diagrams as he proceeds.

Maxwell's theorem of reciprocity of displacements also affords a convenient check on the accuracy of the final computations required for the solution of a pipe line in space. The coefficient of the F_y force in the expression for displacement along the X-axis is numerically equal to the coefficient of the F_x force for displacement along the Y-axis. The same equality exists between the coefficients of the F_z force along the Z-axis. Likewise the coefficients of the F_z force along the Y-axis and the F_y force along the Z-axis are equal.

The authors are pleased with the favorable reception accorded the grapho-analytical method. In considering the simplifications employed in other methods, it appears that the possibility of reducing the grapho-analytical method to three simple elementary principles has not been stressed sufficiently. If one is content to consider bends and fittings as square corners, all modifying factors may be neglected and the number of elements of the line reduced by at least 40 per cent in most cases. For preliminary layouts these simplifications are a great saving in labor and can be used to advantage, but the final layout should be checked to determine the actual stresses, bending moments, and thrusts on anchors as accurately as possible. The assumption that all turns can be considered as square corners may be in error as much as 40 per cent, depending upon the radius of the bend employed and the particular distribution of the piping.

Force-Through-Elastic-Center Method. The "force-through-elastic-center" method described by Mr. Bato affords a good picturization of the bending moments acting on a symmetrical expansion bend, and the explanation of the method, together with general conclusions drawn, constitutes a worth-while contribution to the paper. C. T. Mitchell described this method in detail in an A.S.M.E. paper entitled "A Graphic Method for Determining Expansion Stresses in Pipe Lines."[27]

The method as described by Mr. Mitchell was designed to give an approximation of the flexibility of both symmetrical lines and lines in space. The method in its present development seems to possess certain limitations which may be stated as follows:

a It cannot be used with any degree of accuracy for piping in more than one plane

b It neglects flattening of the cross-section of curved pipe during flexure

c Due to neglect of the effect of flattening of the cross-section, it does not correctly locate or evaluate the maximum bending stress in a pipe line

d It is not adaptable to dealing with a composite line made up with pipe of different diameters or wall thickness, or with fittings having a greater wall thickness than the pipe.

Attention is called to the general application of the grapho-analytical method as given in the "Piping Handbook" where congruent bending-moment areas above and below the base line are canceled directly and only net values used as suggested by Mr. Bato. In the comparison of a series of square bends made up with turns of different radii, which was the purpose of this paper, the expedient of expressing lengths in terms of pipe diameter was resorted to and certain general equations developed which facilitated such a comparison.

The majority of the practical problems in flexibility of piping which the authors have encountered are of the space variety as illustrated in Fig. 30. The solution of such a problem involves the construction of six bending-moment diagrams to determine the relations between deflection and force. It is customary to construct a resulting bending-moment diagram from these diagrams in order to correctly locate the point of maximum bending moment.

Regarding Mr. Bato's substitution of a 150-lb. cast-steel elbow (dimensions of 125-lb. cast iron) in place of the 250-lb. elbow used in the authors' comparison, the authors would explain that the pipe and seamless-tube fittings are suitable for 250-lb. W.S.P., so it seemed only fair to compare assemblies which could be used interchangeably.

Allowable Stress in Piping. Messrs. Houser and Hirschberg contend that the superior strength of cast-steel fittings should be taken into account in assigning allowable stresses, while Mr. Curley suggests that the true circular cross-section and uniform wall thickness of the seamless-tube fitting should permit of a higher working stress than in the case of a pipe bend which may possess considerable initial ellipticity and thinning of the pipe wall due to the process of forming a bend. Professor Hovgaard has offered supporting data as to maximum allowable stresses in pipe bends. As indicated in their paper, the authors have drawn liberally on the theoretical and test results presented in recent papers by Professor Hovgaard, and wish to take this opportunity of thanking him for his important contributions to the knowledge of the behavior of pipe bends and seamless-tube fittings.

The following specifications as to minimum physical properties of cast steel conforming to A.S.T.M. specification A95 and of open-hearth medium-carbon seamless pipe A53 are submitted with reference to the contention of Messrs. Houser and Hirschberg. The properties of seamless tubing (0.150 to 0.200 in. wall thickness) as given by the National Tube Company also are included as corresponding to the tensile strength data quoted by them.

---

[27] Trans. A.S.M.E., May-Aug., 1930, FSP-52-25, p. 167.

	Steel castings, A95	Seamless pipe, A53	Seamless tubing, National Tube Co.
Tensile strength, lb. per sq. in.....	70,000	62,000	54,000[a]
Yield point, lb. per sq. in.....	36,000	35,000	36,000
Elongation in 2 in., percentage.....	22	25	42

[a] Value used by Messrs. Houser and Hirschberg for comparison with tensile strength of cast steel.

An examination of these specifications shows that the yield points are practically the same for both classes of material, the principle variation in physical properties occurring in the value of tensile strength and elongation. In modern analysis of the strength of ductile materials, except in the case of fatigue, which seems to be more directly related to tensile strength, the yield point of the material is the criterion of the strength commonly employed, and factors of safety are based on the yield point rather than on the tensile strength. From this viewpoint which may be obtained from any recent work on strength of materials, this point of Messrs. Houser and Hirschberg does not appear to be well taken.

The example illustrated in Fig. 25 gives essentially the same information as developed in connection with the authors' Figs. 14, 16, 17, and 19: namely, that (1) for equal dimensions, the stresses are higher when using seamless-tube fittings in place of long-radius bends while the reacting forces and the bending moments at the joints are lower, and (2) the magnitude of the differences between the several types of 90-deg. turns depends largely upon their location and the general proportions of the piping layout.

If a flanged joint were located at point *b* of Fig. 25, as the discussers seem to favor, the bending moment would be $1450 \times 260 = 377{,}000$ in-lb. as compared with 152,000 in-lb. in the case of the seamless-tube-fitting assembly. While in the past, general practice has been to design piping to secure certain allowable stresses, the present tendency is to inquire as to just what constitutes "failure" due to lack of flexibility. The practical answer seems to be leaks in joints and excessive thrusts on equipment. An instance of the latter type of "failure" is cited by Mr. Cramer in his discussion where he mentions the dislocation of a turbine from its foundation. In view of the closer scrutiny being given the actual manifestations of lack of flexibility and the obvious objections to having excessive thrusts at anchor points, it does not appear that the proposal of Messrs. Houser and Hirschberg for "getting the most out of an expansion bend" by using greater thrusts and larger bending moments can be applied indiscriminately.

With respect to Mr. Curley's statements concerning initial ellipticity of pipe bends, what information is available seems to indicate that the initial flattening in carefully made bends should not exceed 1 to 2 per cent of the pipe diameter, and usually may be neglected. Cases have been reported, however, where initial ellipticity ran as high as 5 to 10 per cent, but it is believed that this represents the exception rather than the rule.

The maximum longitudinal stress in the middle surface of the pipe cross-section was used as the criterion of strength of curved pipe in bending throughout the paper, since this stress as explained in detail seems to bear a direct relation to the observed failure of a bend. The transverse bending stresses were shown by Professor Hovgaard to exceed the yield point of the material in some instances where no evidence of instability of the bend occurred. Mr. Curley reasons from the work of A. M. Wahl that the effect of internal pressure on the initial ellipticity of the cross-section of a bend should tend to augment the transverse stresses occasioned by flattening of the cross-section of a curved pipe during flexure. Actually, however, Mr. Wahl's papers show that the two transverse stresses in question tend to neutralize each other quite effectively around the pipe cross-section.

There is a possible correction in the value of K which was suggested by Harold E. Jenks in a discussion of the paper on "The Design of Steam Piping to Care for Expansion,"[18] by W. H. Shipman. Mr. Jenks shows that theoretically the expression for K should be modified when applied to sharp-radius turns by subtracting a factor j, whose value for a 10-in. standard-weight seamless-tube fitting is 0.307, from the expression for K as follows:

$$K = \frac{10 + 12h^2 - 0.307}{1 + 12h^2 - 0.307}$$

If this correction were used, the calculated flexibility of a 10-in. standard-weight seamless-tube fitting would be increased by approximately 20 per cent. As the tests made by Professor Hovgaard indicated that the usual expression for K represented the behavior of a seamless-tube fitting accurately enough for practical purposes, this modification was not taken into account in the flexibility comparisons of the paper.

Supporting Data and Recent Tests. Messrs. Zinc, Townsend, and de Jonge have inquired as to recent tests or supporting data tending to confirm the authors' methods for computing the flexibility of piping.

During the past summer extensive tests were conducted by the Research Department of The Detroit Edison Company with a view to verifying the 10 elementary cases for straight pipe and quarter-bends given in Chapter VII of the "Piping Handbook," by Walker and Crocker. It was deemed advisable to check the elementary cases as offering a more general proof of the method developed in the "Piping Handbook" rather than to attempt to deal with the complications involved in making and interpreting tests on complete pipe lines in space. This work demonstrated satisfactory agreement between test and theory, except that in Case X the measured deflection was considerably greater than that computed by formula. This pointed to the need for further check of the theory, and the problem was referred to Prof. S. Timoshenko, who was the first to call Karman's derivation of K to the attention of American engineers (see footnote 9(*b*)). The resulting derivation, which Professor Timoshenko presented under date of November 10, 1930, affords a basis for all the cases involving action of forces and moments on a quarter-bend. The derivation, which is reproduced in an appendix to this closure, furnishes an excellent example of the application of the Castigliano-theorem method to the analysis of such problems. The results of the recent Detroit Edison tests were presented by Messrs. E. T. Cope and E. A. Wert as an A.S.M.E. paper at a meeting of the Fuels and Steam Power Division held in Hartford, Conn., in June, 1931.

In replying to Mr. De Jonge's criticism of cases IX and X, the authors would point out that these cases apply to pipe lines in space and do not concern piping in one plane as treated in the present paper. While the authors question the propriety of discussing a matter distinctly outside the scope of the paper and should have hesitated to take the initiative in bringing it up at this time, they feel indebted to Mr. de Jonge for the opportunity to correct a previous error. It should be mentioned at this point that cases VIII, IX, and X have been corrected in the second edition (1931) of the "Piping Handbook," and that the derivation on which they are based is included as an appendix to this closure. The revised cases, also included in the appendix, have been referred to and checked by Mr. de Jonge.

In reply to Mr. Zinc's question as to the general reliability of methods for computing the flexibility of piping, the authors would assure him that present knowledge properly applied is sufficiently

[18] Trans. A.S.M.E., Sept.-Dec., 1929, FSP-51-52, p. 443.

accurate to preclude any difficulty due to lack of flexibility. They join with him in stressing the need for short-cut methods of applying present knowledge to practical every-day problems.

Conclusion. In conclusion the authors wish to thank all those discussing the paper who by reason of their interest and special knowledge of piping problems have contributed so much to its completeness and usefulness. Due to space limitations, only those points which appear to require further analysis have been considered in the closure. While separate acknowledgment of all the suggestions and views tending to confirm the findings of the paper has not been possible, the authors appreciate this support of their conclusions.

APPENDIX TO AUTHORS' CLOSURE

The following derivation embracing all the cases involving forces and moments acting on a quarter-bend was presented by Professor Timoshenko under date of November 10, 1930. This derivation is so comprehensive and concise that it may well serve as a model for such analyses.

A quarter bend is illustrated in Fig. 31 with all the possible forces and moments indicated thereon. Forces, moments, and angular and linear displacements are considered as positive when in the directions shown by arrows. The general set-up for the moments acting on a particle at point a of Fig. 31 about the three axes is as follows:

$$M_\xi = M_X \sin\varphi + M_Y \cos\varphi - ZR\,(1 - \sin\varphi)$$
$$M_\eta = -M_X \cos\varphi + M_Y \sin\varphi - ZR\cos\varphi$$
$$M_\zeta = M_Z + Y\,R\cos\varphi - XR\,(1 - \sin\varphi)$$

The potential energy stored in the bend by these moments is given by

$$U = \int_0^{\pi/2} \frac{M_\xi^2 R d\varphi}{2C} + \int_0^{\pi/2} \frac{M_\eta^2 R D\varphi}{2B} + \int_0^{\pi/2} \frac{M_\zeta^2 R d\varphi}{2KB}$$

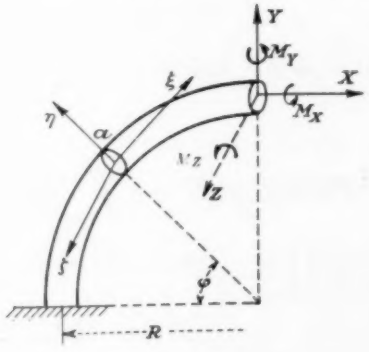


FIG. 31 QUARTER-BEND

in which $C = JG$ = torsional rigidity, $B = EI$ = rigidity in bending, and K is a rigidity multiplication factor which compensates for reduction in flexural rigidity of a curved pipe in the plane of the bend.

From the theorem of Castigliano, the partial derivative of potential energy with respect to any external force gives the displacement corresponding to that force

$$\frac{\partial U}{\partial F} = \int_0^{\pi/2} \frac{M_\xi \left(\frac{\partial M_\xi}{\partial F}\right) Rd\varphi}{C} + \int_0^{\pi/2} \frac{M_\eta \left(\frac{\partial M_\eta}{\partial F}\right) Rd\varphi}{B} + \int_0^{\pi/2} \frac{M_\zeta \left(\frac{\partial M_\zeta}{\partial F}\right) Rd\varphi}{KB}$$

The rotation of the free end of the bend around the X-axis is given by

$$\psi_X = \frac{\partial U}{\partial M_X}$$

$$\frac{\partial M_\xi}{\partial M_X} = \sin\varphi; \quad \frac{\partial M_\eta}{\partial M_X} = -\cos\varphi; \quad \frac{\partial M_\zeta}{\partial M_X} = 0$$

$$\frac{\partial U}{\partial M_X} = \int_0^{\pi/2} \frac{M_\xi \sin\varphi Rd\varphi}{C} - \int_0^{\pi/2} \frac{M_\eta \cos\varphi Rd\varphi}{B}$$

Substituting values of M_ξ and M_η from the general relations above

$$\frac{\partial U}{\partial M_X} = \int_0^{\pi/2} \frac{[M_X \sin\varphi + M_Y \cos\varphi - ZR\,(1 - \sin\varphi)]\sin\varphi Rd\varphi}{C} - \int_0^{\pi/2} \frac{[-M_X \cos\varphi + M_Y \sin\varphi - ZR\cos\varphi]\cos\varphi Rd\varphi}{B}$$

Performing the indicated multiplications and integrations the expression for rotation about the X-axis in terms of all the forces and moments is obtained

$$\psi_X = \frac{\partial U}{\partial M_X} = \frac{R}{4}\left[\frac{\pi M_X + 2M_Y - (4 - \pi)ZR}{C} + \frac{\pi M_X - 2M_Y + \pi ZR}{B}\right]$$

If the rotation due to the action of M_X alone is desired, the other forces and moments drop out and the expression reduces to

$$\psi_X = \frac{R}{4}\left[\frac{\pi M_X}{C} + \frac{\pi M_X}{B}\right]$$

If torsional rigidity is expressed in terms of rigidity in bending, the expression for rotation about the X-axis due to the moment M_X becomes

$$\psi_X = \frac{1.806\,M_X R}{EI} \qquad [\text{Case X}(a)]$$

The rotation of the free end of the bend around the Y-axis is given by

$$\psi_Y = \frac{\partial U}{\partial M_Y}$$

$$\frac{\partial M_\xi}{\partial M_Y} = \cos\varphi; \quad \frac{\partial M_\eta}{\partial M_Y} = \sin\varphi; \quad \frac{\partial M_\zeta}{\partial M_Y} = 0$$

$$\frac{\partial U}{\partial M_Y} = \int_0^{\pi/2} \frac{M_\xi \cos\varphi Rd\varphi}{C} + \int_0^{\pi/2} \frac{M_\eta \sin\varphi Rd\varphi}{B}$$

Substituting values of M_ξ and M_η from the general relations,

$$\frac{\partial U}{\partial M_Y} = \int_0^{\pi/2} \frac{[M_X \sin\varphi + M_Y \cos\varphi - ZR(1 - \sin\varphi)]\cos\varphi Rd\varphi}{C} + \int_0^{\pi/2} \frac{[-M_X \cos\varphi + M_Y \sin\varphi - ZR\cos\varphi]\sin\varphi Rd\varphi}{B}$$

which, when terms are multiplied together and indicated integrations are performed, gives the equation for rotation about the Y-axis in terms of all the forces and moments which may act on the quarter-bend:

$$\psi_X = \frac{\partial U}{\partial M_Y} = \frac{R}{4}\left[\frac{2M_X + \pi M_Y - 2ZR}{C} - \frac{2M_X - \pi M_Y + 2ZR}{B}\right]$$

If the rotation due to action of M_X alone is required, this expression becomes

$$\psi_Y = R/4\left[\frac{2M_X}{C} - \frac{2M_X}{B}\right]$$

which may be combined as

$$\psi_Y = \frac{0.150\,M_X R}{EI} \qquad \text{[Case X}(b)\text{]}$$

The rotation of the free end of the quarter-bend around the Z-axis is found from

$$\psi_Z = \frac{\partial U}{\partial M_Z}$$

$$\frac{\partial M_\xi}{\partial M_Z} = 0\,; \quad \frac{\partial M_\eta}{\partial M_Z} = 0\,; \quad \frac{\partial M_\zeta}{\partial M_Z} = 1$$

$$\frac{\partial U}{\partial M_Z} = \int_0^{\pi/2} \frac{M_\zeta R d\varphi}{KB}$$

Substituting value of M_ζ from the general relations

$$\frac{\partial U}{\partial M_Z} = \int_0^{\pi/2} \frac{[M_Z + YR\cos\varphi - XR\,(1 - \sin\varphi)]\,Rd\varphi}{KB}$$

which when integrated gives the expression for rotation about the Z-axis in terms of all possible forces and moments as

$$\psi_Z = \frac{\partial U}{\partial M_Z} = \frac{R}{2KB}\,[\pi M_Z + 2YR - (\pi - 2)XR]$$

In this expression the M_z term does not appear, so it follows that

$$\psi_Z = 0 \text{ due to } M_Z \qquad \text{[Case X]}$$

The above general expressions have been carried through in terms of the moment M_z as that is the case principally in dispute. It is only necessary to drop out the proper terms in the expressions for rotation about each axis developed above to find the effect of any force or moment acting individually and hence check the cases given in the second edition (1931) of the "Piping Handbook."

The deflection of the free end of the bend along the X-axis is found from

$$\Delta_X = \frac{\partial U}{\partial X}$$

$$\frac{\partial M_\xi}{\partial X} = 0; \quad \frac{\partial M_\eta}{\partial X} = 0; \quad \frac{\partial M_\zeta}{\partial X} = -R(1 - \sin\varphi) = R(\sin\varphi - 1)$$

$$\frac{\partial U}{\partial X} = \int_0^{\pi/2} \frac{M_\zeta\,(\sin\varphi - 1)\,R^2 d\varphi}{KB}$$

$$\frac{\partial U}{\partial X} = \int_0^{\pi/2} \frac{[M_Z + YR\cos\varphi - XR(1 - \sin\varphi)]\,[\sin\varphi - 1]\,R^2 d\varphi}{KB}$$

$$\Delta_X = \frac{\partial U}{\partial X} = \frac{R^2}{4KB}\,[(3\pi - 8)XR - 2YR - (2\pi - 4)M_Z]$$

$$\Delta_X = 0 \text{ due to } M_X \text{ alone} \qquad \text{[Case X]}$$

The deflection of the free end of the bend along the Y-axis is given by

$$\Delta_Y = \frac{\partial U}{\partial Y}$$

$$\frac{\partial M_\xi}{\partial Y} = 0; \quad \frac{\partial M_\eta}{\partial Y} = 0; \quad \frac{\partial M_\zeta}{\partial Y} = R\cos\varphi$$

$$\frac{\partial U}{\partial Y} = \int_0^{\pi/2} \frac{M_\zeta R^2 \cos\varphi d\varphi}{KB}$$

$$\frac{\partial U}{\partial Y} = \int_0^{\pi/2} \frac{[M_Z + YR\cos\varphi - XR + XR\sin\varphi]R^2\cos\varphi d\varphi}{KB}$$

$$\Delta_Y = \frac{\partial U}{\partial Y} = \frac{R^2}{4KB}\,[-2XR + \pi YR + 4M_Z]$$

$$\Delta_Y = 0 \text{ due to } M_X \text{ alone} \qquad \text{[Case X]}$$

The deflection of the free end of the bend along the Z-axis is found from

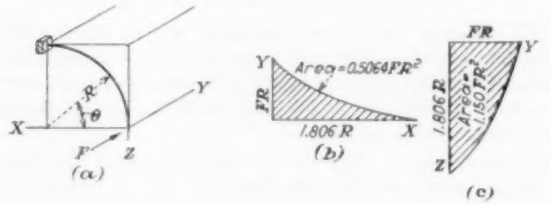


FIG. 32 ILLUSTRATING CASE VIII

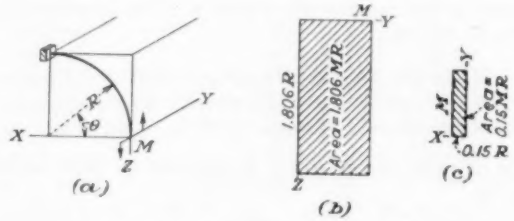


FIG. 33 ILLUSTRATING CASE IX

$$\Delta_Z = \frac{\partial U}{\partial Z}$$

$$\frac{\partial M_\xi}{\partial Z} = -R(1 - \sin\varphi) = R(\sin\varphi - 1); \quad \frac{\partial M_\eta}{\partial Z} = -R\cos\varphi; \quad \frac{\partial M_\zeta}{\partial Z} = 0$$

$$\frac{\partial U}{\partial Z} = \int_0^{\pi/2} \frac{M_\xi\,(\sin\varphi - 1)R^2 d\varphi}{C} + \int_0^{\pi/2} \frac{M_\eta(-R\cos\varphi)Rd\varphi}{B}$$

$$\Delta_Z = \frac{\partial U}{\partial Z} = R^2/4\left[\frac{(\pi - 4)M_X - 2M_Y + (3\pi - 8)ZR}{C} + \frac{\pi M_X - 2M_Y + \pi ZR}{B}\right]$$

$$\Delta_Z = R^2/4\left[\frac{(\pi - 4)M_X}{C} + \frac{\pi M_X}{B}\right] \text{ due to } M_X \text{ only}$$

$$\Delta_Z = \frac{0.5064\,M_X R^2}{EI} \qquad \text{[Case X]}$$

From the general equations for rotation and deflection given in the foregoing, the elementary cases which are in terms of individual forces and moments can be obtained. The cases IV, V, VI, and VII which are involved in this paper are not affected by this rederivation, but cases VIII, IX, and X which are concerned with the solution of piping in space are modified to a greater or lesser degree. The revised cases follow.

CASE VIII. Quarter-Bend Fixed at One End and Acted Upon by a Force F Perpendicular to Plane of Bend (Fig. 32).

(a) $\psi_{XY} = \dfrac{0.5064\,FR^2}{EI}$; $\Delta_{FXY} = \psi_{XY}\,0.754\,R$ = deflection

in direction of force (proportional to rotation in plane perpendicular to free end of bend).

(*b*) $\psi_{YZ} = \dfrac{1.150\ FR^2}{EI}$; $\Delta F_{YZ} = \psi_{YZ}\ 0.754R$ = deflection in direction of force (proportional to rotation in plane tangent to free end of bend).

$\Delta F_Y = \Delta F_{XY} + \Delta F_{YZ} = \dfrac{1.2485\ FR^3}{EI}$ = total deflection in direction of force.

CASE IX. Quarter-Bend Fixed at One End and Acted on by a Moment M Perpendicular to Plane of Bend (Fig. 33).

(*a*) $\psi M_{XY} = \dfrac{0.150\ MR}{EI}$; $\Delta M_{XY} = \psi M_{XY}\ 0.695\ R$ = deflection to *left* along moment arm (proportional to rotation in plane perpendicular to free end of bend).

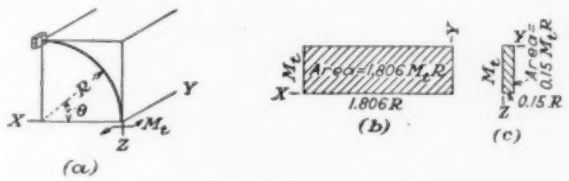


FIG. 34 ILLUSTRATING CASE X

(*b*) $\psi M_{YZ} = \dfrac{1.806\ MR}{EI}$; $\Delta M_{YZ} = \psi M_{YZ}\ 0.695R$ = deflection to *right* along moment arm (proportional to rotation in plane tangent to free end of bend).

$\Delta M_Y = \Delta M_{YZ} - \Delta M_{XY} = \dfrac{1.150\ MR^2}{EI}$ = resultant deflection to right.

(*c*) $\Delta M_Y = \psi M_{YZ}\ 0.637\ R = \dfrac{1.150\ MR^2}{EI}$ = resultant deflection in terms of larger rotation.

CASE X. Quarter Bend Fixed at One End and Acted on by a Torque Moment M_t, in Plane Perpendicular to Free End of Bend (Fig. 34).

(*a*) $\psi_{XY} = \dfrac{1.806\ M_tR}{EI}$; $\Delta M_{tXY} = \psi_{XY}\ 0.306\ R$ = deflection to *right* perpendicular to plane of bend (proportional to rotation in plane perpendicular to free end of bend).

(*b*) $\psi_{YZ} = \dfrac{0.150\ M_tR}{EI}$; $\Delta M_{tYZ} = \psi_{YZ}\ 0.306R$ = deflection to *left* perpendicular to plane of bend (proportional to rotation in plane tangent to free end of bend).

$\Delta M_{tY} = \Delta M_{tXY} - \Delta M_{tYZ} = \dfrac{0.5064\ M_tR^2}{EI}$ = resultant deflection to right.

(*c*) $\Delta M_{tY} = \psi_{XY}\ 0.280\ R = \dfrac{0.5064\ M_tR^2}{EI}$ = resultant deflection in terms of larger rotation.

In applying these elementary cases to the solution of pipe lines in space it is expedient to use the expressions for deflection in terms of the larger rotation and neglect the effect of the smaller rotation. An examination of cases IX and X shows that the smaller rotations are only one-twelfth as large as the principal rotation and in practical problems can be neglected safely.

The changes made necessary by the correction of errors in the original elementary cases given in the "Piping Handbook," while in themselves of considerable magnitude, affect practical problems by only a small percentage. In the particular example of a pipe line in space given in the "Piping Handbook," the maximum stress found for the line is increased by only 1½ per cent by reason of changes in the elementary cases.

With reference to the generalized formulas appended to Mr. de Jonge's discussion, the authors would say that his continued interest in the subject has developed valuable data supplementing the 10 elementary cases presented in the "Piping Handbook." The usual practice in solving expansion problems is to consider pipe lines as consisting of elements built up from quarter-bends and tangents loaded by certain definite applications of forces and moments. The necessity of considering included angles other than 90 deg. in some cases was recognized as early as 1910 by Prof. A. Bantlin[29] in analyzing the characteristics of "lyre shaped" (double offset expansion U-bends), which point also was covered by Crocker and Sanford in their 1922 paper.[30] The general derivation of Professor Timoshenko given in the appendix to this closure works the whole problem out in terms of an indefinite angle to which particular values may be assigned as desired. The final equations are given in terms of a 90-deg. bend as being of the most general use.

Mr. de Jonge has broken up the problem of determining the flexibility of straight pipe and arcs of circles into 24 separate cases. These cases permit the determination of deflections perpendicular to and along a tangent to the arc at the free end as well as along the principal coordinate axes of the system for any given included angle of arc. The case of a straight pipe inclined at an angle other than 90 deg. to the acting force is being developed by Messrs. E. A. Wert and Samuel Smith of The Detroit Edison Company, who hope to present their results in the near future.

Certain of the cases developed by Mr. de Jonge should prove of assistance in calculating the flexibility of arcs of circles other than 90 deg. in cases where need for greater accuracy is apparent than can be obtained by approximating such arcs to straight pipe and quarter-bends. Except when verifying test procedure, it is doubtful if the increased complexity introduced by attempting to use some 24 cases involving generalized angles would be warranted. The ten elementary cases covering 90-deg. bends and tangents given in the Second Edition of the "Piping Handbook," together with the general equations given in the Appendix to this closure, should meet the needs of the practical piping designer.

[29] *Zeitschrift des Vereines deutscher Ingenieure*, Jan. 8, 1910, p. 43.
[30] "The Elasticity of Pipe Bends," A.S.M.E. Trans., 1922, p. 547.

FSP-53-18

Corrosion Tests on Condenser Tubes

Progress Report No. 4 of The A.S.M.E. Special Research Committee on Condenser Tubes[1]

AT the annual meeting in 1929, the report of the committee included evidence that the circulating water in certain surface condensers carried a quantity of air, both in the entrained and dissolved conditions. It was suggested at this meeting that investigations be made on condensers of possibly a dozen selected companies to determine if air is a common cause of trouble. A small budget is necessary to carry out such a course of action. During the year this report was presented to the Prime Movers Committee of the National Electric Light Association with the suggestion that it contribute the necessary funds to carry on this particular feature of the investigation. At the Prime Movers Meeting in Montreal, Oct. 6 and 7, 1930, a resolution was passed endorsing the solicitation of funds either through the N.E.L.A. headquarters or from member companies most vitally interested.

Meanwhile other investigations have been carried on for the committee. A series of tests was made to determine the extent of the effect of air on tube material. The method used was to make comparative corrosion tests with solutions saturated with air and under vacuum.

In consideration of the fact that all chemical reactions such as found in condenser tube corrosion are ionic, an electric current was used in the laboratory tests to accelerate the corrosive action. Cells were made, using Admiralty brass anodes and copper cathodes. These comparative tests were made on a series of solutions ranging in concentration from zero parts per million to 25,000 parts per million of potassium chloride. The arrangement of the apparatus used, both with air and under vacuum, is shown in Figs. 1 and 2, respectively. The data obtained are given in Tables 1 and 2 and also are plotted in curves in Figs. 3 and 4. In Fig. 3 the Admiralty brass anode loss in milligrams per volt drop across the cell is plotted against the concentration of potassium chloride in parts per million. Comparing the two curves—that is, one with air-saturated electrolyte and the other with electrolyte under 27-in. hg. vacuum—it is clear that air decidedly accelerates the rate of corrosion. It is true that its effect is not constant at all concentrations, but its effect is of sufficient magnitude at all concentrations to make its presence undesirable. Referring to Fig. 3, there are peculiarities in the data of the air-saturated solutions at the lower concentrations which are receiving further investigation. Another feature which developed in this investigation is shown in Fig. 4, where the loss of zinc as a percentage of the total anode loss was plotted against concentration. These curves show the tendency of Admiralty brass to dezincify and also the concentration most favorable to dezincification. It appears that a point of maximum is reached at intermediate concentrations. There is a decided break in these curves at from 1000 to 2000 parts per million. It is interesting to note that there was a change in the appearance of the anodes in this range of concentration. At lower concentrations the anode had a darkened

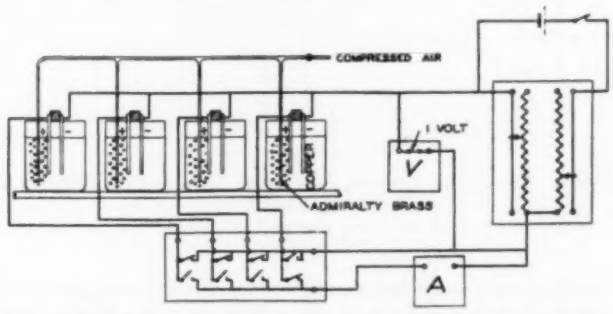


FIG. 1 SET-UP OF CORROSION TEST WITH POTASSIUM-CHLORIDE ELECTROLYTE
(Anode, Admiralty metal; cathode, copper. Electrolyte saturated with air.)

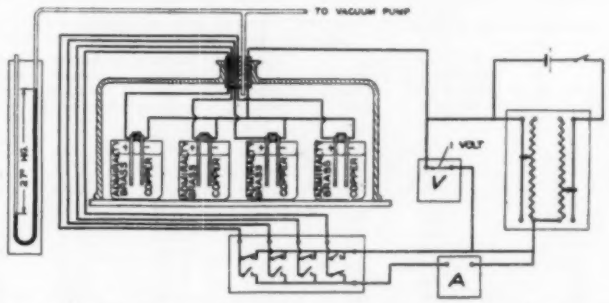


FIG. 2 SET-UP OF CORROSION TEST WITH POTASSIUM-CHLORIDE ELECTROLYTE
(Anode, Admiralty metal; cathode, copper. Electrolyte under 27-in. hg. vacuum.)

[1] The personnel of this committee is as follows:

ALBERT E. WHITE, *Chairman*, Director, Department of Engineering Research, University of Michigan, Ann Arbor, Mich.

BERT HOUGHTON, *Vice-Chairman*, Operating Superintendent, Brooklyn Edison Company, Inc., Brooklyn, N. Y.

PAUL A. BANCEL, Manager, Condenser Department, Ingersoll-Rand Company, New York, N. Y.

R. H. BARBER, International Paper Company, New York, N. Y.

WILLIAM H. BASSETT, Technical Superintendent, American Brass Company, Waterbury, Conn.

GEORGE G. BELL, Manager, Power Development, West Penn Power Company, Pittsburgh, Pa.

DONALD K. CRAMPTON, Metallurgist, Chase Metal Works, Waterbury, Conn.

HARVEY M. CUSHING, Chief Engineer, Buffalo General Electric Company, Buffalo, N. Y.

HAROLD F. EDDY, Mechanical Engineer, Foster Wheeler Corporation, Chicago, Ill.

VINCENT M. FROST, Assistant to General Superintendent of Generation, Public Service Electric & Gas Company, Newark, N. J.

WALTER L. GREEN, JR., Superintending Engineer, Luckenbach Steamship Company, Inc., Brooklyn, N. Y.

CHARLES F. HARWOOD, Manager, Condenser Department, Worthington Pump & Machinery Corporation, New York, N. Y.

C. F. HIRSHFELD, Chief, Research Department, Detroit Edison Company, Detroit, Mich.

HOWARD W. LEITCH, General Superintendent of Power Plants, United Electric Light & Power Company, New York, N. Y.

J. P. LIDIAK, Condenser Department, Westinghouse Electric & Manufacturing Company, Philadelphia, Pa.

ROGER W. PAINE, Lieutenant Commander, Officer in Charge, Specifications Section, Design Division, Bureau of Engineering, Navy Department, Washington, D. C.

WILLIAM B. PRICE, Chief Chemist and Metallurgist, Scovill Manufacturing Company, Waterbury, Conn.

NORMAN G. REINICKER, General Superintendent, Penn Power & Light Company, Allentown, Pa.

HORACE A. STAPLES, Vice-President, British-American Metals Company, Inc., Plainfield, N. J.

WILLIAM REUBEN WEBSTER, Chairman of the Board, Bridgeport Brass Company, Bridgeport, Conn.

Presented before a meeting of the Committee at the Annual Meeting, Dec. 1 to 5, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

oxidized appearance, while at 1000 parts per million and over, the anodes were very coppery in appearance.

The theory that corrosion due to stray electrical currents is the fundamental cause of all condenser tube failures has been suggested from time to time and has been brought forward in recent technical literature. It is argued that, if a condenser is short-circuited properly, the tubes will not corrode, these claims being supported by exhaustive investigation, both in the laboratory and in the plant. As a check of this theory, a large condenser was wired, making eight taps in each water box on the condenser shell and eight taps on each tube sheet. The leads were brought out, and while the condenser was in operation,

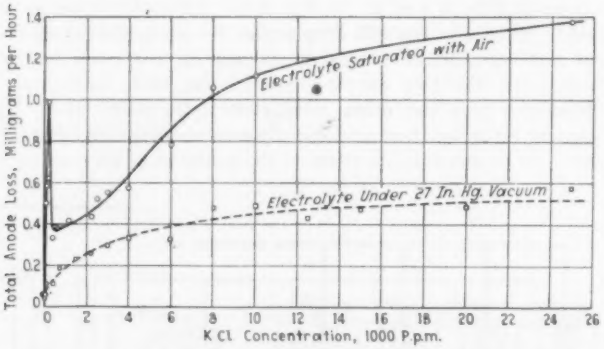


FIG. 3 CORROSION TEST ON ADMIRALTY METAL WITH POTASSIUM-CHLORIDE ELECTROLYTE

(Total anode loss in milligrams per hour, corrected to 1 volt potential difference.)

readings were taken between each pair of leads, from one water box to the other and also across each water box. A very sensitive galvanometer was used for the test, but not the slightest indication was found of voltage drop anywhere in the condenser.

The tubes in this condenser have been in service six years, and in that time 76 tubes have failed out of 13,600 tubes. Inspection of the tubes indicates that the tubes still have several years of life. The corrosion rate on the condenser tubes is therefore slow, and the committee expects to make a similar test on

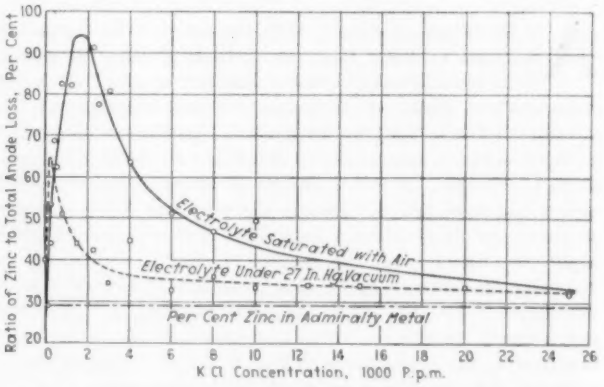


FIG. 4 CORROSION TEST ON ADMIRALTY METAL WITH POTASSIUM-CHLORIDE ELECTROLYTE

(Ratio of zinc to total anode loss.)

condensers having a higher rate of tube failure as a further check on the electrolysis theory.

Investigation of failures of the upper tubes in the second pass of a large condenser was carried on during the year. Tube failures in this condenser were grouped near the top, as shown in the diagram of the tube sheet, Fig. 5. The failed tubes were removed and in many cases were found to be badly dezincified. Further examination of these tubes showed that by far the greater amount of dezincification had taken place on the top of the tube, the bottom of the tube being almost free from dezincification. Figs. 6 and 7 are typical of this condition. The fact that the failures take place near the top of the condenser may be explained by the fact that the upper rows of tubes are not filled with water at all times, and the tubes are exposed to higher concentrations of air in water. That the dezincification took place largely at the top of the tube may be explained by the fact that the air in the circulating water would no doubt follow along the top of the tube, and in line with the committee's investigations concerning the action of air, would speed up dezincification at this point. It was interesting to note that, while in this section dezincification had progressed almost through the tube wall in some cases, in the lower sections of the condenser the tubes showed no attack whatever.

Evidence that the work of this committee is receiving widespread interest is shown by the fact that a number of cases of condenser-tube failure have been sent to the committee for examination. Several of them that are of particular interest are cited.

Several instances of failure of 60/40 brass tubes have been brought to the attention of the committee during the year.

TABLE 1 CORROSION TESTS MADE WITH POTASSIUM CHLORIDE ELECTROLYTE

(Anodes, Admiralty metal; cathodes, copper; area of anodes, 2.4 sq. in.; voltage across cell, 1 volt approx. Electrolyte saturated with air)

Test No.	KCl concentration, parts per million	Duration of test, hours	Average voltage	Average current, milliamperes	Anode loss in weight, milligrams	Anode loss, milligrams per volt per hour	Zinc loss in weight, milligrams	Zinc to total loss, per cent
1.....	0	22	1.114	0.43	1.3	0.053	..	..
2.....	50	22	1.114	0.93	1.4	0.057	..	..
3.....	100	22	1.060	1.19	11.7	0.502	3.4	29.1
4.....	100	23	1.003	0.75	13.5	0.585	2.4	17.8
5.....	200	22	1.060	1.55	14.6	0.626	8.4	52.5
6.....	200	23	1.003	1.42	22.8	0.988	10.0	43.9
7.....	400	22	1.010	1.25	7.4	0.333	5.1	68.9
8.....	500	22	1.110	1.01	7.7	0.389	4.8	62.4
9.....	750	22	1.000	0.80	8.6	0.391	7.1	82.5
10.....	1,200	22	1.050	0.91	9.6	0.416	7.9	82.3
11.....	2,250	22	1.070	0.79	10.3	0.438	9.4	91.2
12.....	2,500	22	1.050	1.02	12.0	0.520	9.3	77.5
13.....	3,000	22	1.070	0.95	13.0	0.552	10.5	80.8
14.....	4,000	22	1.120	1.13	14.2	0.576	9.6	63.6
15.....	6,000	22	1.120	1.12	19.3	0.783	9.9	51.3
16.....	8,000	22	1.080	1.30	25.1	1.056	11.7	46.8
17.....	10,000	22	1.080	1.39	26.4	1.111	13.0	49.3
18.....	25,000	22	1.010	1.32	30.5	1.373	9.4	32.0

TABLE 2 CORROSION TESTS MADE WITH POTASSIUM CHLORIDE ELECTROLYTE

(Electrolyte under 27-in. hg. vacuum)

Test No.	KCl concentration, parts per million	Duration of test, hours	Average voltage	Average current, milliamperes	Anode loss in weight, milligrams	Anode loss, milligrams per volt per hour	Zinc loss in weight, milligrams	Zinc to total loss, per cent
1.....	0	23	1.05	0.042	1.0	0.041	0.4	40.0
2.....	100	22	0.98	0.152	2.4	0.113	1.2	51.7
3.....	400	22	0.98	0.224	2.4	0.113	1.5	62.5
4.....	750	22	0.95	0.184	3.9	0.187	2.0	51.3
5.....	1,500	22	0.95	0.206	4.8	0.230	2.1	43.8
6.....	2,250	22	0.99	0.246	5.6	0.257	2.4	42.3
7.....	3,000	22	0.99	0.292	6.5	0.298	2.2	34.5
8.....	4,000	22	0.95	0.220	7.0	0.335	3.1	44.5
9.....	6,000	22	0.95	0.219	6.8	0.325	2.2	32.7
10.....	8,000	22	0.99	0.297	10.3	0.473	3.7	36.0
11.....	10,000	22	0.99	0.279	10.6	0.487	3.5	33.3
12.....	12,500	22	0.92	0.283	8.6	0.425	2.9	34.0
13.....	15,000	22	0.92	0.232	9.4	0.465	3.2	34.0
14.....	20,000	22	0.98	0.256	10.2	0.478	3.4	33.3
15.....	25,000	22	0.98	0.358	12.1	0.567	4.0	32.7

Each case was a fresh-water installation, and in each case the tubes were badly dezincified after a comparatively short life. One case was that of an inter-and-after cooler. Dezincification had progressed from the outside, or steam side, of the tube. Considered as an electrolyte, the medium of air, condensed steam, and carbon dioxide was extremely weak, but 60/40 metal is so subject to dezincification that even under such apparently favorable conditions the tubes failed. A photomicrograph of this metal is shown in Fig. 8.

Another case that has come to the attention of the committee is the case of a main condenser in the Middle West. This also is a fresh-water installation, using 60/40 brass tubes. The outside, or steam side, of the tube was spotted with a thin film of white deposit about 1 in. in diameter. These spots occurred every few inches along the tube and were generally scattered through the condenser. Upon examination it was found that this was plug-type dezincification which had progressed from the inside entirely through the tube and that the white deposit came from the circulating water and not from the steam. The circulating water was of low concentration and was not contaminated by sewage nor was it acid, having a hydrogen-ion concentration of 7.5. A photomicrograph of this metal is shown in Fig. 9. This case is cited to call further attention to the seriousness of using alloys with high zinc content.

Another recent case of failure occurred in a harbor installation on the Atlantic seaboard. Two large condensers were installed, and presumably were tubed with Admiralty brass. In less than a year of operation 103 tubes have failed in one condenser and 98 tubes in the other. Upon examination it was found that the failed tubes, with one exception, were not Admiralty metal, but what is known in the trade as leaded two-and-one brass. The analysis of one of these tubes is as follows:

	Per cent
Copper	66.88
Tin	Trace
Lead	0.41
Iron	0.06
Zinc (by difference)	32.65

The tubes were pitted in rows of 2 or 3 ft. in extent. These pitted areas did not occur in any particular section of the condenser or in any particular position of the tube. A photograph of a pitted section is shown in Fig. 10. A photomicrograph of the metal, showing a quantity of slag inclusions in which the lead drops may be seen, is shown in Fig. 11. It has been well determined and generally conceded that the alloy leaded two-and-one brass is not suitable for sea-water installations.

Corroded condenser tubes were recently submitted from a plant which was using the chlorine method of tube cleaning. These tubes were protected by Flowrites about 3 in. from the inlet end. Within the Flowrite area the tube was unattacked and in good condition. Then for a distance of several feet the tube was badly pitted, in several places entirely through the tube wall. Chlorine is a very strong corrosive agent, and its action on brass is well known. In order to bring out the rapidity by which brass is corroded in the presence of chlorine and hypochlorous acid, a new condenser tube was exposed to a water-and-chlorine solution for a period of 48 hours. A photograph of this tube is shown in Fig. 12. The vigorous action of the chlorine on the brass is evidenced by the deep corrosion present on the tube. A photograph of the original tube taken out of service is also shown in Fig. 13. While the desirability of preventing algæ growth in the condenser tubes is appreciated, it should be pointed out that the use of so strong a corrosive and oxidizing agent as chlorine should be guarded with great care as it is possible to ruin entirely a condenser-tube installation in a very short time should the chlorine attack the tubes.

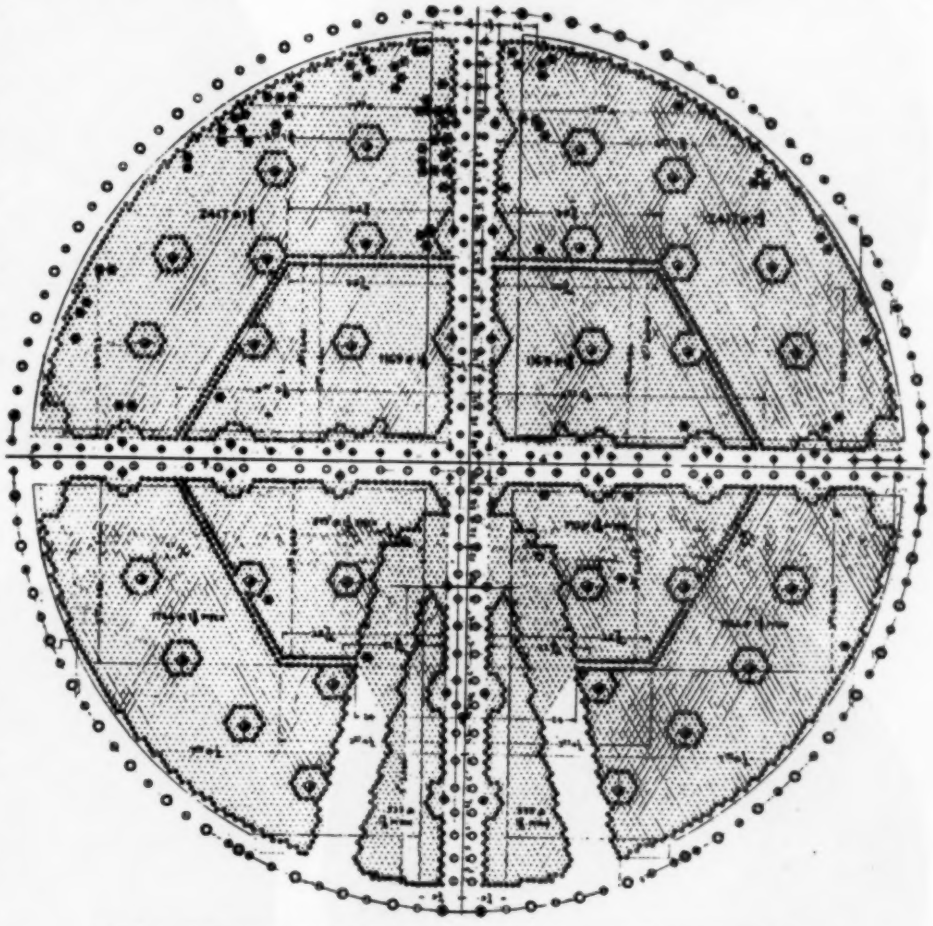


FIG. 5 CONDENSER-TUBE SHEET, SHOWING POSITION OF DEZINCIFIED TUBES

A case of corrosion which, without question, was caused by stray electric currents was submitted from the Middle West. This was a Radojet inter-and-after cooler in a fresh-water installation. The cooler was tubed with Admiralty brass, the tubes being set vertically in groups. The upper half of the tubes in two of these groups was badly corroded on the inside, or water side, of the tube. The lower half of each tube was

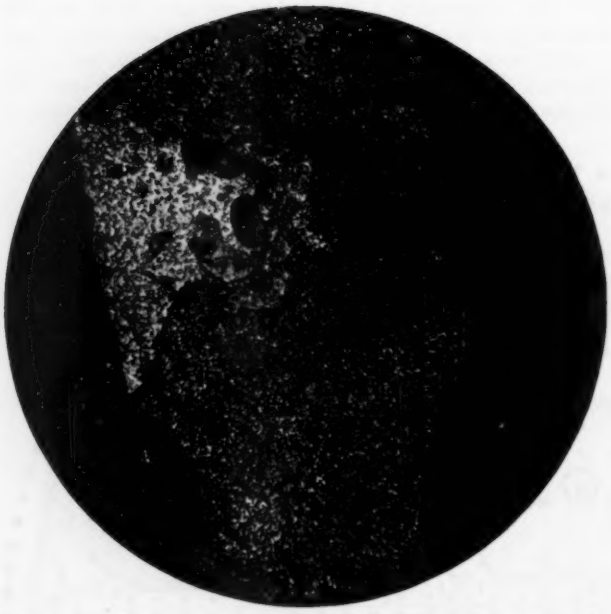


FIG. 6 TOP SECTOR OF SECTION OF ADMIRALTY BRASS TUBE, SHOWING ALMOST COMPLETE DEZINCIFICATION



FIG. 7 BOTTOM SECTOR OF SECTION OF TUBE SHOWN IN FIG. 6, SHOWING SLIGHT DEZINCIFICATION

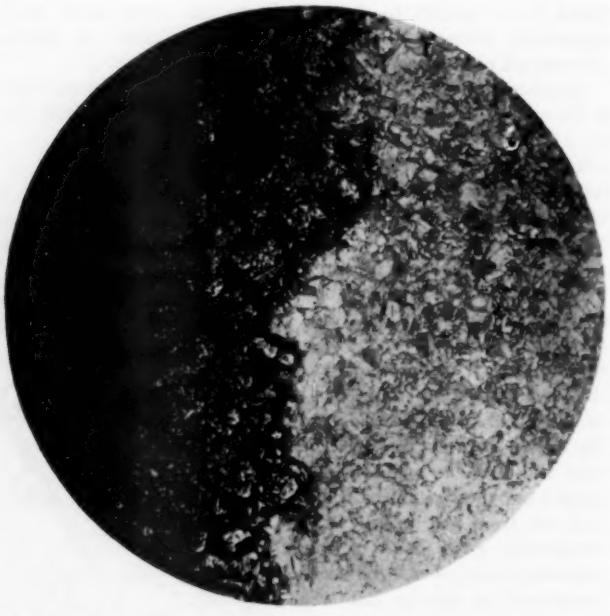


FIG. 8 DEZINCIFICATION OF STEAM SIDE OF 60/40 BRASS TUBE IN FRESH-WATER INSTALLATION

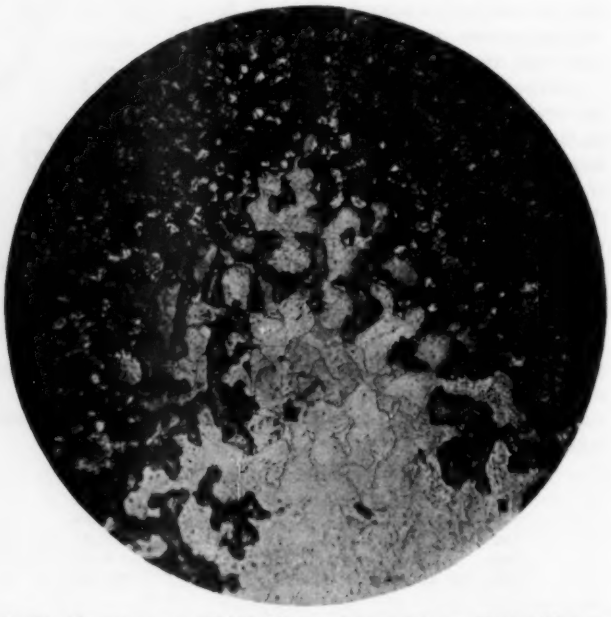


FIG. 9 FAILURE OF 60/40 BRASS TUBE FROM FRESH-WATER CONDENSER

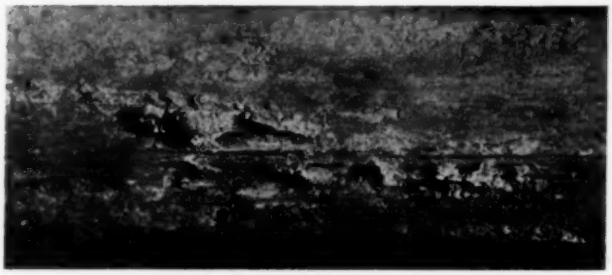


FIG. 10 CORROSION OF LEADED TWO-AND-ONE BRASS TUBE USED IN HARBOR INSTALLATION



FIG. 11 LEADED TWO-AND-ONE BRASS TUBE SHOWING LEAD DROPS

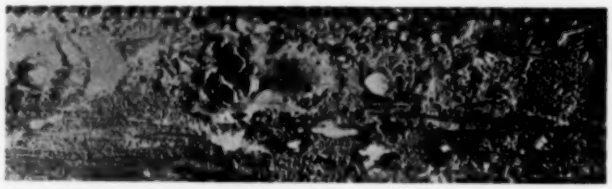


FIG. 12 ADMIRALTY BRASS TUBE TREATED WITH CHLORINE IN LABORATORY

plugged almost solid with the insoluble products of corrosion. The tube wall of the lower portion of the tube, being protected from the electrolyte by the scale formation, had not been affected.

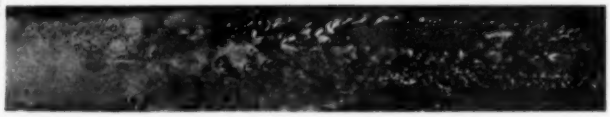


FIG. 13 ADMIRALTY BRASS TUBE FROM CONDENSER WHERE CHLORINE WAS USED IN CLEANING TUBES

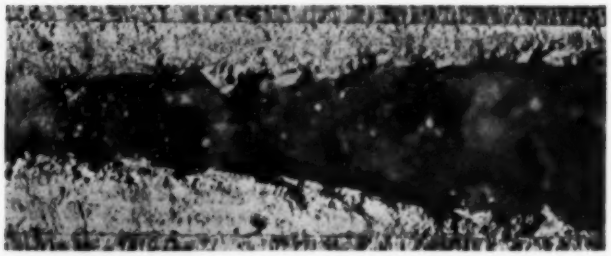


FIG. 14 PRODUCTS OF CORROSION, COLLECTED AS SCALE IN LOWER END, IN ADMIRALTY BRASS TUBES OF VERTICAL INTERCOOLER USING FRESH WATER

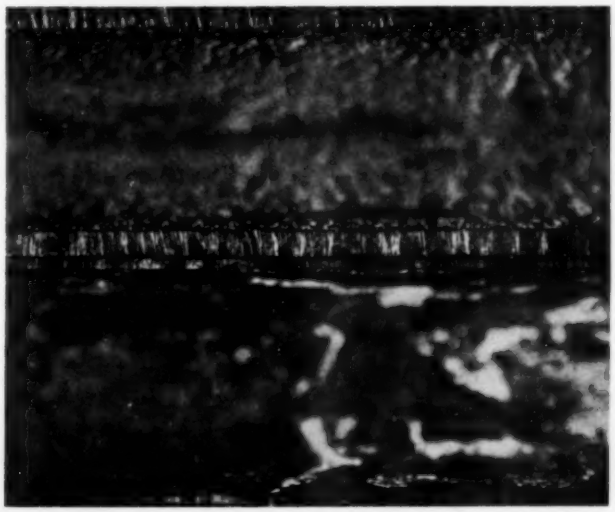


FIG. 15 COMPARISON OF WALL THICKNESS AT UPPER AND LOWER ENDS OF TUBE SHOWN IN FIG. 14

Photographs of the comparative tube-wall thickness, indicative of the extent of corrosion, and the formation of the insoluble products of corrosion as scale in the tube are shown in Figs. 14 and 15.

During the coming year the laboratory and field tests will be continued and their scope broadened as opportunity permits.

FSP-53-19a

Heat Absorption in Water-Cooled Furnaces

By WM. L. DeBAUFRE,[1] NEW YORK, N. Y.

Based on the furnace heat balance, an empirical "effectiveness factor" is derived for water-cooled surfaces therein, relating the heat radiated by the burning fuel and products of combustion to the furnace-outlet temperature. The value of this factor is calculated from certain experimental data for turbulent horizontally fired pulverized-coal furnaces with bare water-cooled tube walls. For these experimental data, the fraction radiated of the heat liberated in the furnace has also been calculated and is compared with Broido's curve and with the value calculated by the Orrok-Hudson formula. The method of applying the effectiveness factor to design problems is illustrated by showing the effect of preheated-air temperature upon the furnace-outlet temperature.



WHEN "water screens" were installed in pulverized-fuel furnaces to prevent slagging of ash in the furnace bottoms, tests were made to determine the rate of heat absorption in order to design properly the downcomers and risers. These tests generally consisted in isolating certain tubes and passing water through them at a sufficient velocity to give a moderate temperature rise. In other cases, the "water screens" were connected to steam drums separate from the main steam drums of the boiler so that the steam generated therein could be measured. The rate of heat absorption so determined per square foot of water-cooled surface exposed to radiation, was compared with the operating capacity of the boiler in per cent of rating rather than attempting to connect the rate of heat absorption with the furnace temperature. This method of handling the problem was satisfactory as long as small portions only of the furnace walls were covered by water-cooled surfaces.

When larger portions of the refractory walls were replaced by water-cooled surfaces, a relation between rate of heat absorption and furnace temperature was based on these data and on an experiment by John Bell made some years previously, in which one of a first row of boiler tubes was replaced by a similar tube through which water was circulated and the temperature rise noted.[2]

With the development of steam generators in which the furnace walls are almost entirely replaced by water-cooled surfaces, it became desirable to have a relation which would more definitely proportion the total heat absorbed by the unit between the heating surface exposed to radiation in the furnace and the heating surface in the boiler tube banks. About that time appeared the timely papers upon the subject by Broido[3] and Wohlenberg,[4] with Orrok's discussions thereon. Broido and Orrok gave empirical relations for calculating the ratio of the heat absorbed by radiation to the heat liberated in the furnace, which relations were assumed to apply to all types of furnaces and conditions of operation; but the actual ratio might differ by 20 per cent from the calculated value. The paper by Wohlenberg showed how complicated the subject becomes when it is attempted to take account of all factors involved in the problem.

The idea was therefore conceived of applying a comparatively simple relation involving a factor which measures the effectiveness of the water-cooled surface and varies with the furnace design and the character of the fuel burned. This relation will be derived and applied to certain experimental data from pulverized-coal-fired furnaces. The method of applying the results to design problems will then be explained.

The above-mentioned relation is based on the furnace heat balance that the heat R radiated from the burning fuel and products of combustion plus the sensible heat P in the products leaving the furnace must equal the net heat B produced by burning the fuel plus the sensible heat A in the preheated air supplied for combustion; that is,

$$R + P = B + A$$

These heat quantities may be total hourly rates for the furnace in question, or may be based on one square foot of water-cooled surface in the furnace or on one pound of fuel fired.

Due to unburnt combustible in the ashes and flue dust, the heat B actually liberated in the furnace is less than the calorimeter heating value of the fuel, which is accordingly reduced by 14,600 times the fraction of a pound of unburnt carbon per pound of fuel fired. A further reduction is made equal to the heat required to evaporate the moisture in the fuel and that formed from the hydrogen burned. The latent heat of this water vapor is taken as approximately 1050 B.t.u. per lb. at room temperature. That is,

$$B = H - 14{,}600 \times C - 1050 \times M$$

where H = gross calorimeter heating value of the fuel fired B.t.u. per lb.
C = lb. unburnt carbon per lb. of fuel fired, and
M = lb. moisture in fuel and formed from hydrogen per lb. of fuel fired.

The latent heat of the water vapor could be added to the sensible heat P in the products of combustion leaving the furnace instead of being subtracted from the gross heating value of the fuel fired. The latter method is preferred, however, because it is considered more logical to base the heat radiated upon the net heat liberated in the furnace than upon the gross heating value of the fuel. Also, it is customary abroad to base the efficiency of

---

[1] Head of Technical Research Department, International Combustion Engineering Corp. Mem. A.S.M.E. Mr. DeBaufre was graduated from the Baltimore Polytechnic Institute in 1903, and acquired the degrees of E.E., M.E., and M.S. from Lehigh University. He taught at Baltimore Polytechnic for several years, and during the latter part of this period was head of the Department of Engineering there. For a number of years thereafter he was mechanical engineer with the U. S. Naval Engineering Experiment Station, leaving this work to become professor of mechanical engineering and later chairman of the Mechanical Engineering Department at the University of Nebraska. He left this position in 1926 to assume his present duties.

[2] "An Outline of the Theory of Heat Transmission and Absorption in Application to Ordinary Engineering Problems." Proceedings of the Engineers' Society of Western Pennsylvania, vol. 32 (1916), pp. 797–845.

Contributed by the Fuels Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[3] "Radiation in Boiler Furnaces," B. N. Broido, Trans. A.S.M.E., vol. 47 (1925), pp. 1123–1177.

[4] "Radiation in the Pulverized-Fuel Furnace," W. J. Wohlenberg, Trans. A.S.M.E., vol. 47 (1925), pp. 127–176.

steam-generating equipment upon the net heating value of the fuel rather than penalize the steam power plant in comparison with the internal-conbustion power plant by using the gross heating value of the fuel, as is done in this country.

The sensible heat A in the preheated air is calculated from the amount of air supplied for combustion, the portion of it which is preheated, and the temperature to which it is preheated. The quantity of air supplied per pound of fuel fired may be calculated from the analysis of the products of combustion leaving the furnace, the composition of the fuel fired, and the loss of combustible in the ashes and flue dust. The sum of A and B is the net heat available for radiation in the furnace and for heating the products of combustion to the furnace-outlet temperature. The sensible heat S in the products of combustion may be calculated for the composition and temperature of the products leaving the furnace. The heat R radiated is then found by difference.

Now, the more effective the water-cooled surfaces in absorbing heat within the furnace, the larger must be R and the lower must be the temperature of the products leaving the furnace. Hence, the ratio of R to some function of the furnace-outlet temperature is an "effectiveness factor" for the water-cooled surfaces within

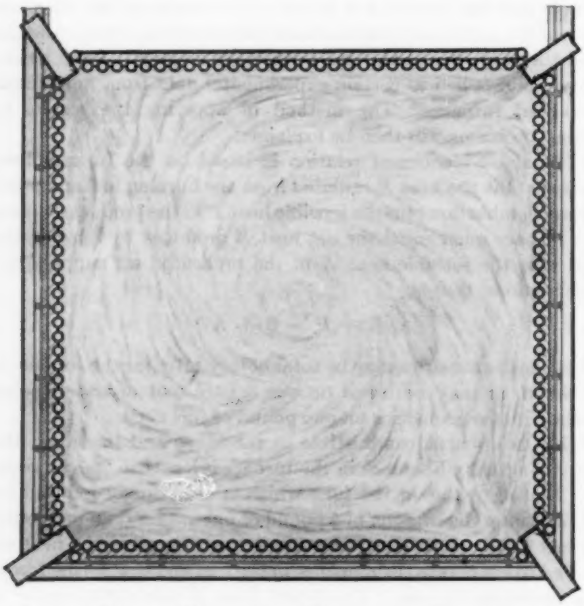


FIG. 1 HORIZONTAL TANGENTIAL FIRING

the furnace. As the effectiveness of these surfaces must vary with their arrangement and condition, the effectiveness factor will show by its variations, the effect of such changes rather than be a universal constant which will apply under all conditions.

The temperature function which has been chosen as a basis for calculating the effectiveness factor of the water-cooled surfaces is the fourth power of the absolute temperature of the products of combustion leaving the furnace minus the fourth power of the absolute temperature of the water evaporating within the water-cooled surfaces. The effectiveness factor E may be employed to calculate the heat R radiated per pound of fuel fired by means of the relation:

$$R = \frac{E}{F}\left[\left(\frac{T_G}{1000}\right)^4 - \left(\frac{T_S}{1000}\right)^4\right]$$

where T_G = absolute temperature of the products of combustion leaving the furnace, deg. fahr.

T_S = absolute temperature of the water evaporating into steam within the water-cooled surfaces, deg. fahr.

F = fuel fired per hour per square foot of water-cooled surface exposed to radiation, lb.

R = heat radiated per lb. of fuel fired in B.t.u.; and

E = effectiveness factor of the water-cooled surfaces.

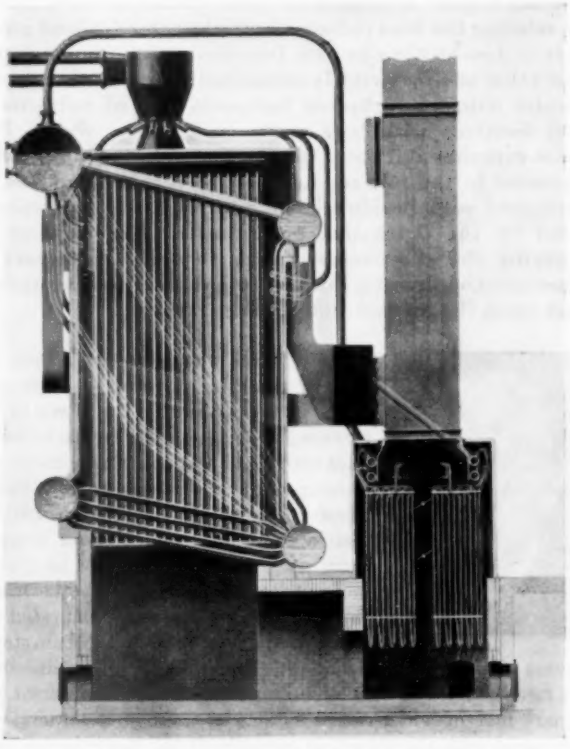


FIG. 2 WATER-COOLED FURNACE AT MANCHESTER, ENGLAND

There are a number of reasons why the heat radiated by the burning fuel and products of combustion to the water-cooled surfaces should not be proportional to the difference of the fourth powers of the furnace-outlet absolute temperature and the absolute steam temperature; thus:

The temperature of the burning fuel and products of combustion within the furnace is not uniform, but varies from a maximum at the place of most intense combustion, to a minimum generally at the furnace outlet.

The water-cooled surfaces absorbing radiant heat are at a higher temperature than the water evaporating into steam within these surfaces.

Some heat may pass by convection and conduction rather than by radiation.

Radiation from non-luminous gases does not follow the fourth-power law, and radiation from burning fuel may not follow this law.

The flames may not entirely fill the furnace volume.

Clouds of cool, unburnt fuel may intervene between the hot flames and part of the water-cooled surface.

Combustion may not be complete within the furnace, but continue into and through the boiler-tube banks.

All of these discrepancies affect the effectiveness factor when the above relation is applied to determine its value for any set of experimental conditions. All of them, however, depend mainly if not entirely upon the characteristics of the fuel, the arrange-

ment and condition of the furnace, and the capacity at which it is operated. Hence, the effectiveness factor determined by the above relation should apply to similar fuels, similar furnace arrangements, and equal capacities.

The effectiveness factor will also vary with the way in which the water-cooled surface exposed to radiation is measured. While the projected area is the simplest way of measuring such surfaces, actual exposed areas have been employed, partly because it has been customary to measure these surfaces in this way, but also because the actual areas were considered to give a better comparative measure of the heat absorbed, for example, by the first rows of boiler tubes and by furnace-wall tubes. Two-thirds of the surface of the first row of boiler tubes and one-third of the surface of the second row have been taken as exposed to radiation. Two-thirds of the surface of the "screen tubes," which are usually widely spaced, has been taken as exposed to radiation. When a wall is made up of plain tubes closely spaced, the exposed area has been taken as one-half of the tube surface. If widely spaced in front of a refractory wall, two-thirds of the surface has been taken. For fin tubes, the actual area of tube and fins on the furnace side has been taken as exposed to radiation.

Table 1 gives experimental data from four pulverized-coal-fired furnaces in which all sides of the furnaces are nearly completely water cooled by bare tubes, and in which tangential firing is used as illustrated in Fig. 1.

The first furnace is that of the Wood steam generator located at Manchester, England. This furnace, as shown in Fig. 2, has the dimensions: width, 12 ft. 6 in.; depth, 12 ft. 0 in.; height, 18 ft. 9 in. It was fired near the top by burners located in the four corners of the furnace. The effectiveness factor is plotted in Fig. 3. In run 1, the furnace was comparatively clean, with a resultant factor of 1600. In run 2, a lower factor was obtained at a reduced rating. Runs 3 and 4 were not completed. In runs 5 and 6, the factor was around 1400. When runs 7, 8, and 9 were made, considerable amounts of slag had collected in the furnace, thereby accounting for the low factors around 800 and 700. The second series of runs with this furnace showed the same drop in the effectiveness factor from run to run as slag accumulated in the furnace.

In the second steam generator, installed at Brighton, England, the furnace walls were kept in a very clean condition by use of steam-jet blowers. Factors of 2000 and more were obtained, as shown by the points plotted in Fig. 3. These factors indicate the effectiveness of unusually clean furnace surfaces. The arrangement of the furnace at Brighton is similar to the one at Manchester shown in Fig. 2.

The data from the third furnace at Hellgate show the effectiveness of water-cooled surfaces in a large furnace operated with good coal at moderate combustion rates. This furnace is 26 ft. 5 in. by 25 ft. 1 in., making a cross-sectional area of 662 sq. ft. The effectiveness factors are plotted in Fig. 4 and indicate a falling off in effectiveness of the surface as the rate of heat liberation is reduced.

The fourth furnace, Fig. 5, has about the same volume as the third furnace, but is higher with a somewhat less cross-sectional area, being 20 ft. 4 in. by 28 ft. 4 in. with a corresponding cross-sectional area of 576 sq. ft. The effectiveness factor is less than for the third furnace, due probably to a thicker layer of slag adhering to the furnace walls.

In view of these and other data, a factor of 1500 has been selected for clean or but slightly dirty furnaces with rates of combustion around 28,000 to 30,000 B.t.u. per hr. per cu. ft. of furnace volume. As the rate of heat liberation is reduced, the factor is taken to drop off in a straight-line relation, as indicated in Figs. 3 and 4, to a value of 500 at zero heat liberation. For a moderately dirty furnace, a factor of 1000 is used at the maximum combustion rate. For a considerable amount of slag adhering to the furnace walls, the factor is taken as 500 for all rates of heat liberation.

Table 2 shows how the heat radiated by the burning fuel and products of combustion varies with the furnace-outlet temperature for several values of the effectiveness factor. These results are based on a steam pressure of 450 lb. gage, with a corresponding saturated steam temperature of 460 deg. fahr. Slightly higher results would be calculated for lower steam pressures and

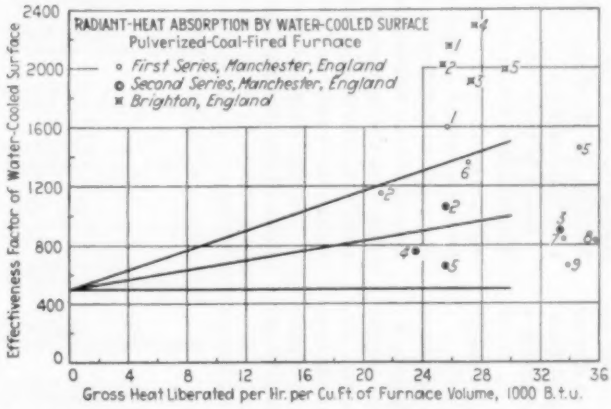


FIG. 3 EFFECTIVENESS FACTORS FOR WATER-COOLED FURNACES AT MANCHESTER AND BRIGHTON, ENGLAND

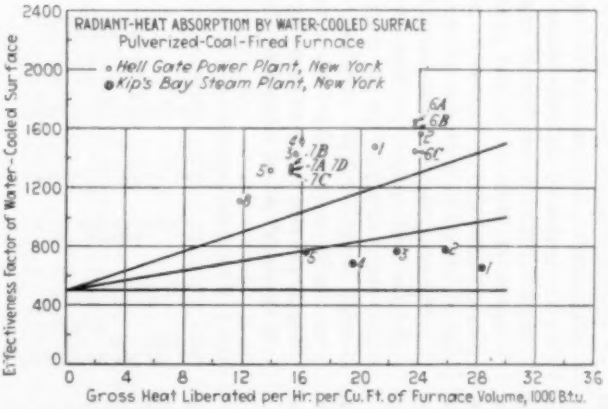


FIG. 4 EFFECTIVENESS FACTORS FOR WATER-COOLED FURNACES AT HELL GATE POWER PLANT AND KIP'S BAY STEAM PLANT

the reverse for higher steam pressures. For example, with a furnace-outlet temperature of 2000 deg. fahr., the calculated heat radiated would be 0.7 per cent higher for 125 lb. steam pressure and 1.4 per cent lower for 1400 lb. steam pressure than tabulated.

In addition to the effectiveness factor, Table 1 also contains the ratios of the heat radiated to the gross heating value of the fuel and to the net heat liberated and in the preheated air. These ratios are plotted in Figs. 6 and 7, respectively, in which Broido's curve has been drawn for comparison.

Table 3 contains the ratios calculated by Hudson's formula as modified by Orrok, and the deviations from the experimental values.

One particularly interesting set of data is found in runs 7-A to 7-D for the third furnace. In these runs, the rate of combustion was maintained constant while the air supply was varied to give from 11.5 to 15 per cent CO_2 in the products of combustion leaving the furnace. The furnace-outlet temperature varied

nearly 100 deg. The fraction of the heat liberated which was absorbed by the furnace walls varied from 45 to 53 per cent. The effectiveness factor averaged 1320, having this value at both the lowest and highest quantities of air supplied for combustion and differing by 20 only above and below the average. As a variation in the effectiveness factor from 1500 to 1000 would change the furnace-outlet temperature by about 200 deg. fahr., a variation of 20 would affect the furnace-outlet temperature by less than 10 deg.

These data therefore indicate that the effectiveness factor is not affected by the CO_2 percentage. It would also seem reasonable to assume that the factor is not affected by the preheated-air temperature.

To illustrate the use of the effectiveness factor, we shall calculate the furnace-outlet temperature as affected by the preheated-air temperature for a typical Pittsburgh bituminous coal burned in a furnace having a volume of 8396 cu. ft. and having 3040 sq. ft. of water-cooled surface exposed to radiation, or 0.3620 sq. ft. per cu. ft. of furnace volume.

The composition of this coal as fired is: Moisture, 3.4 per cent; sulphur, 1.0 per cent; hydrogen, 4.7 per cent; carbon, 74.4 per cent; nitrogen, 1.4 per cent; oxygen, 6.7 per cent; ash, 8.4 per cent. Assuming 80 per cent of the ash to be flue dust containing 20 per cent combustible and practically no carbon in the ashpit refuse comprising the remaining 20 per cent of the ash of the coal, the unburnt carbon will be $0.017\frac{1}{2}$ lb. per lb. of coal fired. The carbon burned will be 0.744 — 0.017 = 0.727 lb. per lb. of coal fired.

Hence, in the furnace exit gases will be 0.727/12 = 0.0606 mole of CO_2 and CO per lb. of coal fired. There will also be 0.010/32 = 0.0003 mole of SO_2 which will be treated as CO_2. With no CO and 15 per cent CO_2, there will be 0.0609/0.15 = 0.4060 mole of dry products, of which 0.4060 — 0.0609 = 0.3451 mole is O_2 and N_2. From the hydrogen in the coal there will be formed 0.047/2 = 0.0235 mole of H_2O. The moisture present in the coal is equal to 0.034/18 = 0.0019 mole of H_2O.

The dry air supplied for combustion equals the moles of dry products plus one-half the moles of H_2O

TABLE 1 EXPERIMENTAL DATA FROM PULVERIZED-COAL FURNACES

Location of furnace	Furnace volume, cu. ft.	Water-cooled surface exposed to radiation, sq. ft.	Date of test	Run number	Coal fired per hour, lb.	Coal fired per hr. per cu. ft. of furnace volume, lb.	Coal fired per hr. per sq. ft. of water-cooled surface, lb.	Gross heating value per lb. of coal, B.t.u.	Gross heat liberation per hr. per cu. ft. of furnace volume, B.t.u.	Gross heat liberation per hr. per sq. ft. of water-cooled surface, B.t.u.	Volatile matter in coal as fired, per cent	Sulphur in coal as fired, per cent	Hydrogen in coal as fired, per cent	Carbon in coal as fired, per cent	Nitrogen in coal as fired, per cent	Oxygen in coal as fired, per cent	Moisture in coal as fired, per cent	Ash in coal as fired, per cent	Fusion temperature of ash, deg. fahr.	Pulverized coal through 40 mesh, per cent	Pulverized coal through 60 mesh, per cent	Pulverized coal through 100 mesh, per cent	Pulverized coal through 200 mesh, per cent	Pulverized coal through 325 mesh, per cent
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)	(20)	(21)	(22)	(23)	(24)	(25)
			1926																					
Manchester, England	2,653	1223	6-21	1	5,706	2.15	4.67	11,947	25,700	55,800	28	1.0	4.0	73.0	1.5	2.5	9.0	9.0	..	99.7	..	88.8	69.4	..
			6-22	2	4,605	1.74	3.77	12,198	21,200	46,000	28	1.0	4.0	73.0	1.5	2.5	9.0	9.0	..	98.8	..	78.2	58.3	..
			6-24	3	7,485	2.82	6.12	12,327	34,700	75,400	28	1.0	4.0	73.0	1.5	2.5	9.0	9.0	..	99.6	..	78.8	53.3	..
			6-26	6	5,808	2.19	4.75	12,391	27,100	58,800	28	1.0	4.0	73.0	1.5	2.5	9.0	9.0	..	99.1	..	77.4	55.0	..
			6-26	7	7,299	2.75	5.97	12,236	33,600	73,000	28	1.0	4.0	73.0	1.5	2.5	9.0	9.0	..	99.3	..	85.0	64.3	..
			6-28	8	7,856	2.96	6.42	12,104	35,800	77,700	28	1.0	4.0	73.0	1.5	2.5	9.0	9.0	..	99.8	..	81.7	51.4	..
			6-29	9	7,304	2.75	5.97	12,308	33,900	73,400	28	1.0	4.0	73.0	1.5	2.5	9.0	9.0	..	99.8	..	83.5	49.7	..
			1927																	30 mesh	50 mesh			
Manchester, England	2,653	1223	3-22	2	5,600	2.11	4.58	11,978	25,600	54,800	34	1.03	4.69	66.37	1.26	9.31	12.14	5.20	2282	99.4	94.0	80.0	61.5	..
			3-23	3	7,724	..91	6.32	11,485	33,400	72,600	34	1.00	4.54	64.18	1.22	9.01	14.19	5.86	2282	99.3	92.3	81.6	58.7	..
			3-24	4	5,302	2.00	4.33	11,808	23,600	51,200	34	1.02	4.65	65.68	1.25	9.22	12.52	5.66	2282	99.4	93.0	76.0	57.4	..
			3-25	5	5,827	2.20	4.77	11,638	25,600	55,500	34	1.01	4.59	64.86	1.24	9.10	13.86	5.34	2282	99.3	92.8	74.0	55.9	..
			1929																					
Brighton, England	3,960	1927	3-27	1	8,260	2.08	4.28	12,375	25,800	53,000	34.7	1.37	4.28	68.97	1.10	8.62	6.20	9.46	..	..	96.9	89.3	67.3	..
			4-3	2	7,820	1.98	4.06	12,880	25,420	52,100	32.8	1.50	4.14	71.73	1.32	7.96	5.75	7.60	2540	..	96.9	89.3	67.3	..
			4-3	3	8,375	2.12	4.35	12,880	27,300	56,100	32.8	1.50	4.14	71.73	1.32	7.96	5.75	7.60	2540	..	96.9	89.3	67.3	..
			4-3	4	8,450	2.14	4.38	12,880	27,500	56,500	32.8	1.50	4.14	71.73	1.32	7.96	5.75	7.60	2540	..	96.9	89.3	67.3	..
			4-12	5	9,100	2.30	4.72	12,880	29,600	60,750	32.8	1.50	4.14	71.73	1.32	7.96	5.75	7.60	2540	..	96.9	89.3	67.3	..
			1929																					
Hell Gate Power Plant, New York	18,800	4827	11-6	1	28,200	1.50	5.84	13,974	21,000	81,600	20.31	0.61	4.20	79.93	1.10	2.95	3.18	8.03	2300	..	..	92	75	26
			11-7	2	32,140	1.71	6.66	14,066	24,100	93,600	20.30	0.62	4.24	80.76	1.16	2.99	4.52	5.71	2290	..	..	86	69	46
			11-8	3	20,910	1.11	4.33	14,070	15,600	60,900	20.98	0.57	4.37	80.05	1.12	3.38	5.10	5.41	2270	..	..	93	78	40
			11-11	4	21,330	1.13	4.42	14,126	16,000	62,400	20.82	0.59	4.34	80.87	1.13	3.03	4.24	5.80	2270	..	..	89	70	32
			11-12	5	18,875	1.00	3.91	13,919	13,900	54,400	19.56	0.65	4.36	79.49	1.13	3.01	4.30	7.06	2320	..	..	90	74	36
			11-13	6-A	31,800	1.69	6.59	14,003	23,700	92,300	20.26	0.57	4.37	79.74	1.14	3.02	5.46	5.70	2300	..	..	91	69	33
			11-13	6-B	31,800	1.69	6.59	14,003	23,700	92,300	20.26	0.57	4.37	79.74	1.14	3.02	5.46	5.70	2300	..	..	79	65	33
			11-14	6-C	31,800	1.69	o.59	14,003	23,700	92,300	20.26	0.57	4.37	79.74	1.14	3.02	5.46	5.70	2300	..	..	88	69	23
			11-14	7-A	20,920	1.11	4.33	13,827	15,300	59,900	20.80	0.61	4.28	70.05	1.13	2.61	5.74	6.58	2280	..	..	92	78	44
			11-14	7-B	20,920	1.11	4.33	13,827	15,300	59,900	20.80	0.61	4.28	70.05	1.13	2.61	5.74	6.58	2280	..	..	92	77	43
			11-14	7-C	20,920	1.11	4.33	13,827	15,300	59,900	20.80	0.61	4.28	70.05	1.13	2.61	5.74	6.58	2280	..	..	84	60	44
			11-14	7-D	20,920	1.11	4.33	13,827	15,300	59,900	20.80	0.61	4.28	70.05	1.13	2.61	5.74	6.58	2280	..	..	92	76	38
			11-15	8	16,000	0.85	3.31	13,827	11,800	45,800	20.80	0.61	4.28	70.05	1.13	2.61	5.74	6.58	2280	..	..	88	66	48
			1930																	40 mesh	60 mesh			
Kip's Bay Steam Plant, New York	18,705	4635	1-22	1	37,620	2.01	8.12	14,090	28,340	114,400	35.03	1.08	4.73	78.59	1.22	7.15	1.02	6.21	2320	86	74	68	30	2
			1-23	2	34,610	1.85	7.47	13,970	25,830	104,400	35.05	1.07	4.71	78.28	1.21	7.11	0.87	6.75	2320	98	92	84	60	32
			1-24	3	30,130	1.61	6.50	14,000	22,550	91,000	34.63	1.07	4.71	78.26	1.21	7.12	1.02	6.61	2320	100	90	82	64	34
			1-25	4	26,190	1.40	5.65	13,960	19,530	78,900	34.15	1.07	4.70	78.01	1.21	7.10	1.08	6.83	2320	99	94	84	56	22
			1-25	5	21,960	1.17	4.74	13,900	16,310	65,900	34.27	1.07	4.69	77.88	1.21	7.09	1.24	6.82	2320	98	88	86	40	16

TABLE 1 (*Continued*)

Unburnt carbon loss per lb. of coal as fired, B.t.u.	Heat to evaporate moisture per lb. of coal as fired, B.t.u.	Net heat liberation per lb. of coal as fired, B.t.u.	CO_2 in products of combustion leaving the furnace, per cent	CO in products of combustion leaving furnace, per cent	O_2 in products of combustion leaving furnace, per cent	N_2 in products of combustion leaving furnace, per cent	Products of combustion leaving furnace per lb. of coal fired, lb.	Air supplied for combustion per lb. of coal fired, lb.	Excess air, per cent	Portion of air preheated, per cent	Preheated air temperature, deg. fahr.	Sensible heat in preheated air per lb. of coal fired, B.t.u.	Heat radiated and in products per lb. of coal fired, B.t.u.	Furnace outlet temperature, deg. fahr.	Sensible heat in products of combustion leaving furnace per lb. of coal fired, B.t.u.	Heat radiated per lb. of coal fired, B.t.u.	Heat radiated per hr. per sq. ft. of water-cooled surface, B.t.u.	Fraction radiated of gross heat liberation	Net heat available per hr. per sq. ft. of water-cooled surface, B.t.u.	Fraction radiated of net heat liberation	Steam pressure, gage, lb.	Saturated steam temperature, deg. fahr.	$\left(\frac{Tg}{1000}\right)^4 - \left(\frac{Ts}{1000}\right)^4$	Effectiveness factor of water-cooled surface
(26)	(27)	(28)	(29)	(30)	(31)	(32)	(33)	(34)	(35)	(36)	(37)	(38)	(39)	(40)	(41)	(42)	(43)	(44)	(45)	(46)	(47)	(48)	(49)	(50)
522	473	10,952	13.5	...	5.5	81.0	13.72	12.84	31.0	90.1	735	1884	12,836	1680	5827	7009	32,730	0.587	59,900	0.546	264	410	20.40	1600
554	473	11,171	12.1	...	6.9	81.0	15.18	14.30	45.8	89.4	746	2116	13,287	1721	6591	6696	25,240	0.549	50,100	0.504	272	413	22.04	1150
616	473	11,238	15.3	...	3.7	81.0	12.24	11.36	15.9	91.0	795	1839	13,077	1888	5953	7124	43,600	0.578	80,000	0.545	280	416	29.81	1460
365	473	11,553	14.3	...	4.7	81.0	13.02	12.14	23.9	89.4	758	1830	13,383	1805	6006	7377	35,040	0.596	63,600	0.551	288	418	25.72	1360
278	473	11,485	14.7	...	4.3	81.0	12.69	11.81	20.5	78.8	860	1804	13,289	2130	7060	6229	37,190	0.509	79,300	0.469	255	408	44.43	840
394	473	11,237	15.1	...	3.9	81.0	12.39	11.51	17.4	79.8	857	1775	13,012	2162	7008	6004	38,550	0.496	83,500	0.462	285	417	46.67	830
329	473	11,506	14.8	...	4.2	81.0	12.62	11.74	19.7	79.2	870	1825	13,331	2250	7462	5869	35,040	0.477	79,600	0.440	308	424	53.33	660
348	570	11,060	16.0	0	3.1	80.9	10.94	10.01	15.2	77.4	515	819	11,879	1860	5301	6578	30,130	0.550	54,400	0.554	215	394	28.44	1060
242	578	10,665	15.5	0	3.7	80.8	11.02	10.10	19.3	84.9	608	1090	11,756	2070	6018	5738	36,260	0.499	74,300	0.488	247	405	40.41	900
235	571	11,002	15.3	0	3.9	80.8	11.40	10.47	20.8	73.6	520	822	11,824	1965	5860	5964	25,820	0.504	51,200	0.504	229	399	34.04	760
315	580	10,743	15.9	0	3.2	80.9	10.81	9.88	16.3	75.8	590	928	11,671	2084	5958	5713	27,250	0.491	55,700	0.489	213	393	41.36	660
220	470	11,685	15.5	0	..	...	11.88	10.89	19.0	95.0	600	1341	13,026	1580	4663	8363	35,790	0.675	55,800	0.641	220	396	16.68	2150
190	452	12,238	15.5	0	..	...	12.37	11.35	19.8	95.1	580	1346	13,584	1600	4911	8673	35,210	0.676	55,200	0.638	228	399	17.46	2020
190	452	12,238	16.3	0	..	...	11.83	10.81	14.1	95.0	580	1281	13,519	1660	4900	8619	37,490	0.668	58,800	0.638	228	399	19.66	1910
190	452	12,238	16.3	0	..	...	11.83	10.81	14.1	94.8	580	1278	13,516	1585	4654	8862	38,820	0.687	59,200	0.656	228	399	16.95	2290
190	452	12,238	16.0	0	..	...	12.02	11.01	16.2	93.6	640	1437	13,675	1680	5046	8629	40,730	0.671	64,500	0.631	235	401	20.42	1990
187	427	13,360	13.57	0	5.63	80.80	15.24	14.34	36.4	76.0	180	291	13,651	1813	7052	6599	38,540	0.472	79,700	0.484	280	416	26.10	1470
213	451	13,402	13.91	0	5.23	80.86	15.02	14.12	33.2	80.5	214	398	13,800	1854	7202	6598	43,940	0.469	91,900	0.478	280	416	28.08	1560
97	468	13,505	13.34	0	5.90	80.76	15.67	14.74	38.5	69.1	185	284	13,789	1705	6770	7019	30,390	0.499	59,700	0.509	280	416	21.38	1420
197	452	13,477	13.34	0	5.78	80.78	15.64	14.65	37.5	69.8	180	273	13,750	1638	6490	7260	32,090	0.514	60,800	0.528	280	416	18.79	1710
213	452	13,254	13.36	0	5.86	80.78	15.38	14.47	38.0	66.9	161	214	13,468	1686	6559	6909	27,010	0.497	52,700	0.513	280	416	20.62	1310
200	476	13,327	15.35	0	3.62	81.03	13.62	12.68	20.5	80.3	185	284	13,611	1861	6531	7080	46,660	0.506	89,700	0.520	280	416	28.43	1640
86	476	13,441	14.02	0	5.10	80.88	14.95	13.99	31.8	80.2	184	311	13,752	1842	7051	6701	44,160	0.478	90,600	0.487	280	416	27.49	1610
222	476	13,305	11.94	0	7.48	80.58	17.14	16.19	54.0	80.1	189	375	13,680	1817	7928	5752	37,910	0.411	90,200	0.420	280	416	26.29	1440
93	465	13,269	15.09	0	3.91	81.00	13.82	12.89	22.5	68.9	214	311	13,580	1768	6259	7321	31,700	0.529	58,800	0.539	280	416	24.05	1320
69	465	13,293	13.74	0	5.42	80.84	15.11	14.17	34.4	69.0	204	319	13,612	1732	6656	6956	30,120	0.503	58,900	0.511	280	416	22.50	1340
75	465	13,287	12.64	0	6.69	80.67	16.31	15.37	45.8	68.9	197	327	13,614	1715	7086	6528	28,270	0.472	58,900	0.480	280	416	21.79	1300
51	465	13,311	11.49	0	7.99	80.52	17.85	16.91	60.1	69.1	211	401	13,712	1676	7539	6173	26,730	0.446	59,400	0.450	280	416	20.23	1320
273	465	13,089	12.47	0	6.88	80.65	16.25	15.31	47.3	..	..	0	13,089	1645	6757	6332	20,960	0.457	43,300	0.484	280	416	19.05	1100
150	458	13,482	13.1	0	5.1	81.8	15.54	14.60	40.3	67.7	609	1306	14,788	2366	9693	5095	41,370	0.362	120,100	0.344	285	417	63.19	655
95	454	13,421	12.9	0	5.0	82.1	15.77	14.84	42.6	59.2	590	1121	14,542	2231	9175	5367	40,090	0.384	108,600	0.369	284	417	51.85	773
62	456	13,482	12.5	0	5.5	82.0	16.28	15.35	47.2	56.3	574	1068	14,550	2156	9112	5438	35,350	0.389	94,600	0.374	283	417	46.24	764
104	455	13,401	12.3	0	5.9	81.8	16.42	15.50	49.5	46.2	516	782	14,183	2113	8980	5203	29,400	0.373	80,100	0.367	275	414	43.25	680
170	456	13,274	15.1	0	2.9	82.0	13.50	12.58	22.2	41.2	474	515	13,789	2072	7301	6488	30,750	0.467	65,400	0.470	275	414	40.52	759

formed from the hydrogen in the fuel minus the number of moles equivalent to the nitrogen and oxygen in the fuel. The last mentioned are equal to 0.014/28 = 0.0005 mole of N_2 and 0.067/32 = 0.0021 mole of O_2, respectively. Hence, 0.4060 + 0.0118 — 0.0005 — 0.0021 = 0.4152 mole of dry air. For a relative humidity of 70 per cent at 70 deg. fahr., each mole of dry air has with it 0.0176 mole of water vapor, or 0.4152 × 0.0176 = 0.0073 mole of water vapor per pound of coal fired.

The air supplied for combustion per pound of coal fired thus consists of 0.4152 mole of dry air and 0.0073 mole of water vapor. The products of combustion leaving the furnace consist of 0.0609 mole of CO_2 (and SO_2), 0.3451 mole of O_2 and N_2, and 0.0235 + 0.0019 + 0.0073 = 0.0327 mole of H_2O.

By means of Table 4, the sensible heat in the preheated air and in the products of combustion may be determined for any temperature desired.

Now, the gross heating value of this Pittsburgh coal is 13,700 B.t.u. per lb. The loss due to unburnt carbon is 0.017 × 14,600 = 248 B.t.u. The heat required to evaporate the moisture in the coal and that formed from the hydrogen is 1050 (0.047 × 9 + 0.034) = 480 B.t.u. Hence, the net heat liberated is 13,700 — 248 — 480 = 12,972 B.t.u. per lb. of coal fired. Assuming 90 per cent of the air supplied for combustion to be preheated to 300 deg. fahr., we find the sensible heat above 70 deg. fahr. to be 610 B.t.u., making 12,973 + 610 = 13,583 B.t.u. for radiating and raising the temperature of the products of combustion.

For a gross heat liberation of 29,500 B.t.u. per hr. per cu. ft. of furnace volume, we have 29,500/13,700 = 2.153 lb. of coal fired per hr. per cu. ft., or 2.153 × 0.3620 = 5.948 lb. per hr. per sq. ft. of the water-cooled surface that is exposed to radiation.

By assuming various temperatures of the products of combustion leaving the furnace, a balance may be found in which the heat radiated per square foot with a factor of, say, 1500, divided by 5.948 lb. of coal fired per sq. ft., plus the sensible heat in the products of 1 lb. of coal, just equals 13,583 B.t.u. For the conditions assumed, this temperature is 1874 deg. fahr.

By similar calculations for other preheated-air temperatures, the following furnace-outlet temperatures are found to correspond to several effectiveness factors:

Effectiveness factor	1500	1000	500
Furnace-outlet temperature, deg. fahr.:			
70-deg. fahr. air	1837	2014	2325
For 300 deg. fahr. preheated air	1874	2057	2379
For 600 deg. fahr. preheated air	1922	2111	2446

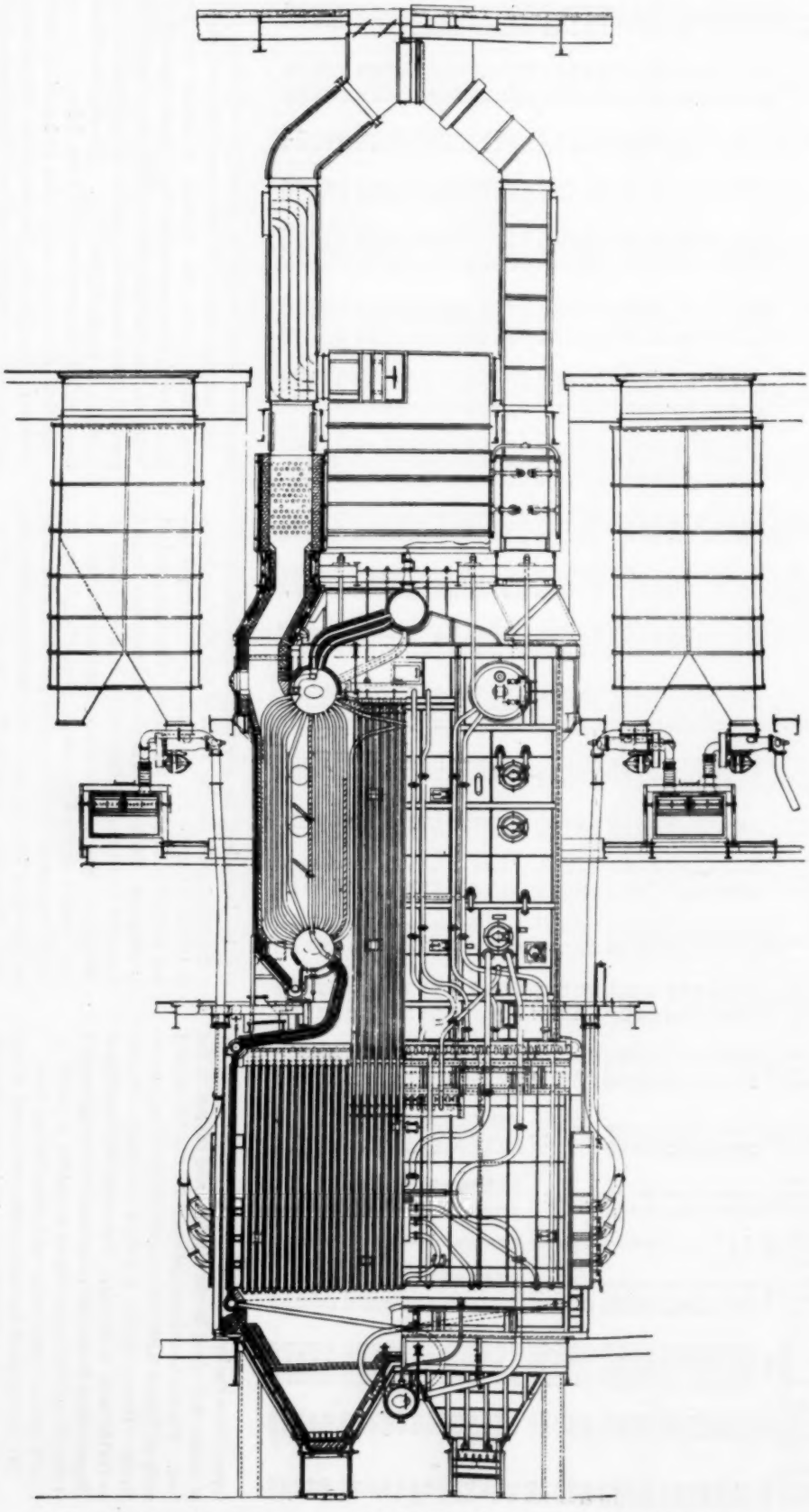


FIG. 5 WATER-COOLED FURNACE AT KIP'S BAY STEAM PLANT, NEW YORK CITY

TABLE 2 HEAT RADIATED PER SQUARE FOOT OF WATER-COOLED SURFACE, B.T.U. PER HR.

(Steam pressure, 450 lb. gage)

Furnace-outlet temperature, deg. fahr.	For effectiveness factor of 1500	1250	1000	750	500
1000	5,742	4,785	3,828	2,871	1,914
1100	7,809	6,508	5,206	3,904	2,603
1200	10,315	8,596	6,877	5,158	3,439
1300	13,318	11,098	8,879	6,659	4,439
1400	16,879	14,066	11,252	8,439	5,626
1500	21,062	17,552	14,042	10,531	7,021
1600	25,938	21,615	17,292	12,969	8,646
1700	31,577	26,314	21,051	15,789	10,526
1800	38,057	31,714	25,371	19,028	12,686
1900	45,456	37,880	30,304	22,728	15,152
2000	53,858	44,882	35,906	26,929	17,953
2100	63,350	52,792	42,233	31,675	21,117
2200	74,022	61,685	49,348	37,011	24,674
2300	85,967	71,639	57,311	42,984	28,656
2400	99,284	82,737	66,190	49,642	33,095
2500	114,074	95,062	76,049	57,037	38,025
2600	130,441	108,701	86,961	65,221	43,480
2700	148,494	123,745	98,996	74,247	49,498
2800	168,345	140,288	112,230	84,172	56,115
2900	190,109	158,424	126,739	95,054	63,370
3000	213,905	178,254	142,603	106,952	71,302

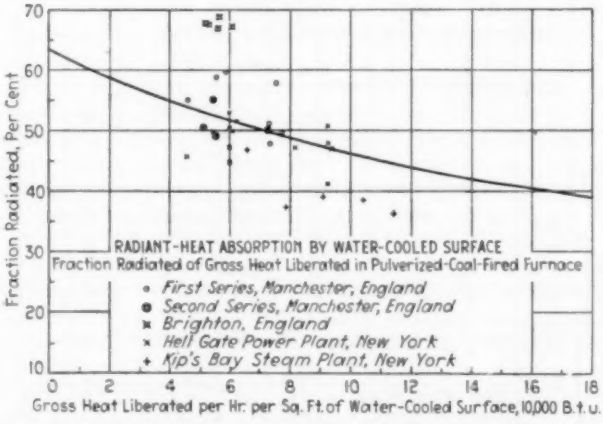


FIG. 6 RATIO OF HEAT RADIATED TO GROSS HEATING VALUE OF FUEL FIRED

The above furnace-outlet temperatures show an increase of somewhat less than 200 deg. fahr. for a change in the factor from 1500 to 1000, and a further increase of about 300 deg. fahr. for a change in the factor from 1000 to 500.

The heat absorbed by the water-cooled surfaces exposed to radiation in a furnace is somewhat less than the heat radiated from the burning fuel and products of combustion as calculated by this method, by reason of loss of heat from the furnace casing to surrounding objects. Also, the "unaccounted for" loss assumed as a margin of safety in a heat balance for a pulverized-coal-fired furnace, may be subtracted from the heat radiated as the most convenient place of including this item in an analysis of a steam-generating unit.

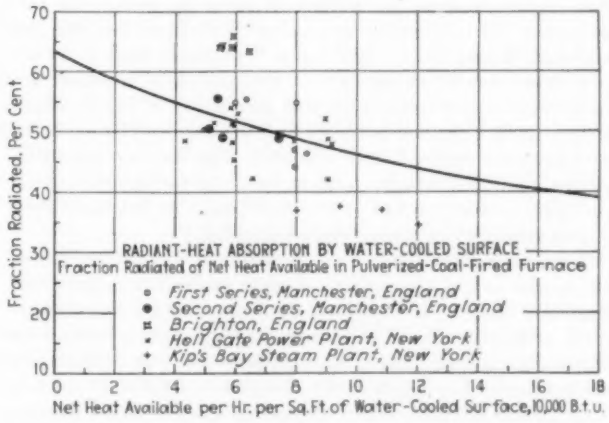


FIG. 7 RATIO OF HEAT RADIATED TO NET HEAT LIBERATED AND IN PREHEATED AIR

The "effectiveness factor" can be applied to stoker-fired furnaces as well as to pulverized-fuel furnaces, and also to oil- and gas-fired furnaces. By experimental and operating experience with furnaces of different types burning various fuels, judgment is developed in selecting the proper factor for determining the heat absorbed by radiation and the furnace-outlet temperature, either by calculation or from curves for typical fuels.

TABLE 3 FRACTION RADIATED FROM ORROK-HUDSON FORMULA

Furnace	Run No.	Fraction radiated by formula	Deviation based on gross heat liberated	Deviation based on net heat available
Manchester, England	1	0.494	−0.093	−0.052
	2	0.494	−0.055	−0.010
	5	0.490	−0.088	−0.055
	6	0.504	−0.092	−0.047
	7	0.484	−0.025	+0.015
	8	0.482	−0.014	+0.020
	9	0.485	+0.008	+0.045
Manchester, England	2	0.558	+0.008	+0.004
	3	0.514	+0.015	+0.026
	4	0.553	+0.049	+0.049
	5	0.557	+0.066	+0.068
Brighton, England	1	0.545	−0.130	−0.096
	2	0.541	−0.135	−0.097
	3	0.544	−0.124	−0.094
	4	0.544	−0.143	−0.112
	5	0.531	−0.140	−0.100
Hell Gate Power Plant, New York	1	0.438	−0.034	−0.046
	2	0.426	−0.043	−0.052
	3	0.468	−0.031	−0.041
	4	0.467	−0.047	−0.061
	5	0.485	−0.012	−0.028
	6-A	0.453	−0.053	−0.067
	6-B	0.429	−0.049	−0.058
	6-C	0.394	−0.017	−0.026
	7-A	0.502	−0.027	−0.037
	7-B	0.478	−0.025	−0.033
	7-C	0.458	−0.014	−0.022
	7-D	0.434	−0.012	−0.016
	8	0.484	+0.027	0
Kip's Bay Steam Plant, New York	1	0.394	+0.032	+0.050
	2	0.400	+0.016	+0.031
	3	0.409	+0.020	+0.035
	4	0.423	+0.050	+0.056
	5	0.496	+0.029	+0.026

TABLE 4 SENSIBLE HEATS OF GASES

Temperature, deg. fahr.	Heat added under constant pressure, B.t.u. per lb-mole CO_2	H_2O	N_2, O_2, CO (dry air)
0	0.0	0.0	0.0
32	278.0	265.4	222.6
60	524.5	497.8	417.5
70	613.3	580.8	487.2
100	882.0	830.0	696.1
200	1,801.6	1,661.9	1,393.6
300	2,756.9	2,496.6	2,092.6
400	3,745.8	3,335.1	2,793.5
500	4,766.4	4,178.0	3,496.5
600	5,816.6	5,026.3	4,201.7
700	6,894.7	5,880.8	4,909.5
800	7,999.0	6,742.3	5,620.1
900	9,127.6	7,611.8	6,333.6
1000	10,279.0	8,489.9	7,050.5
1100	11,451.6	9,377.7	7,770.9
1200	12,643.8	10,275.9	8,495.0
1300	13,854.3	11,185.4	9,223.0
1400	15,081.6	12,107.0	9,955.3
1500	16,324.4	13,041.6	10,692.1
1600	17,581.4	13,990.0	11,433.6
1700	18,851.6	14,953.0	12,180.0
1800	20,133.7	15,931.6	12,931.6
1900	21,426.6	16,926.5	13,688.6
2000	22,729.5	17,938.7	14,451.3
2100	24,041.4	18,968.8	15,219.9
2200	25,361.4	20,017.9	15,994.6
2300	26,688.7	21,086.7	16,775.7
2400	28,022.6	22,176.1	17,563.5
2500	29,362.4	23,287.0	18,358.1
2600	30,707.5	24,420.1	19,159.8
2700	32,057.4	25,576.3	19,968.8
2800	33,411.6	26,756.5	20,785.4
2900	34,769.7	27,961.5	21,609.9
3000	36,131.3	29,192.2	22,442.4

Discussion

A. G. CHRISTIE.[5] This is a subject in which the writer has been interested for many years and which was discussed by him in an earlier paper of a general nature presented before this Society at its Milwaukee meeting in 1925. Since that time such heat transfer has received much attention. However, a study of the now voluminous literature on the subject indicates the need of still further data on the subject of radiation and of the laws governing the dissipation of radiant heat from burning fuels and from flames in a boiler furnace. A further need is the simplification and rationalization of the analytical methods that have been applied in the past, and this paper of Mr. DeBaufre offering such a method is timely and welcome. The method of treatment for radiant heat problems suggested by the author carries additional weight from the fact that he has, through his professional position, had ample opportunity to test the validity and applicability of the solutions suggested.

A paramount factor in this discussion is the determination the proper area of the surfaces of tubes exposed to radiant heat. Professor Ramzin, in a paper before the World Power Conference, analyzed radiant heat absorption in a bank of boiler tubes above a furnace and found that this total area was equivalent to the product of the length of the tubes and the width of the tube bank measured from between the outside tubes. This is less than assumed by the author. Professor Ramzin found that the equivalent area of a Murray fin-tube wall was 1.1 times the total projected area measured over the fins. This also is less than assumed by the author. The writer stresses these differences in order to indicate the need of standardization of the measurement of such heat-absorbing surface and recommends that this question of standardization receive the attention of A.S.M.E. probably through its Power Test Code Committee.

The author suggests an "effectiveness factor" E which will serve as a measure of heat absorption. This can be defined as the B.t.u. absorbed per square foot of surface per hour per unit of difference of the fourth powers of the results of dividing both absolute furnace temperature and absolute tube temperature by 1000. His analysis indicates that this factor is unaffected by variations in CO_2 or by preheated air temperatures. It is well known that both of these materially affect furnace temperature and the total heat absorbed per square foot of heat-absorbing surface. The author assumes that the Stefan and Boltzmann law, based on the difference of the fourth powers of absolute furnace and tube temperatures, holds for furnace conditions. Some have cast doubt on the applicability of this relation. An experimental research to prove or disprove the Stefan and Boltzmann law seems desirable at this time.

A perusal of Figs. 3 and 4 shows many values above the upper line which presumably represent clean surfaces and for which condition an effectiveness factor of 1500 was assumed with a gross heat liberation of 28,000 to 30,000 B.t.u. per hr. per cu. ft. of furnace volume. Why was a higher value not chosen for clean surfaces?

Slag of varying thickness has an appreciable effect on heat transfer. Why was an effectiveness factor of 500 chosen for all rates of heat transfer with considerable slag adhering to the furnace walls?

Evidently the selection of an effectiveness factor for any given case is a matter of judgment based on the condition of the heat-absorbing surface and the presence of slag on such surfaces. This would indicate that this effectiveness factor is still empirical and is not an exact measure of heat transfer.

Hudson's formula as modified by Orrok has been widely quoted. A study of Table 3 indicates divergences from DeBaufre's figure ranging as high as 20 per cent of the observed values. Which is probably in error, the empirical Hudson formula or the empirical effectiveness factor of DeBaufre?

The author deserves the thanks of engineers for developing this interesting solution. Its acceptance will depend to a large extent upon confirmation based on further tests and observations.

[5] Professor of Mechanical Engineering, Johns Hopkins University, Baltimore, Md. Mem. A.S.M.E.

ARNOLD WEISSELBERG.[6] From the valuable experimental data which Mr. DeBaufre has made available through this paper, tests 7-A to 7-D at the Hell Gate Station are singled out to demonstrate that the effectiveness factor is not affected by the CO_2 percentage. This statement should be taken with reservation. If the data of these tests permit the drawing of a conclusion, no doubt tests 6-A to 6-C should enjoy the same privilege. However, these tests show that the effectiveness factor changes considerably with the CO_2 content. If the effectiveness factors of these tests are plotted against the CO_2 values, there is obtained practically a linear relation. Furthermore, with test 3 of the same station carried out at the same firing rate as 7-A to 7-D, the effectiveness factor is a great deal off the average by plus 100. However, this test differs from 7-A to 7-D in that the preheat temperature is 20 deg. fahr. lower, which would also cast doubt on the assumption which the author takes as probable that the effectiveness factor is not affected by preheat. On the other

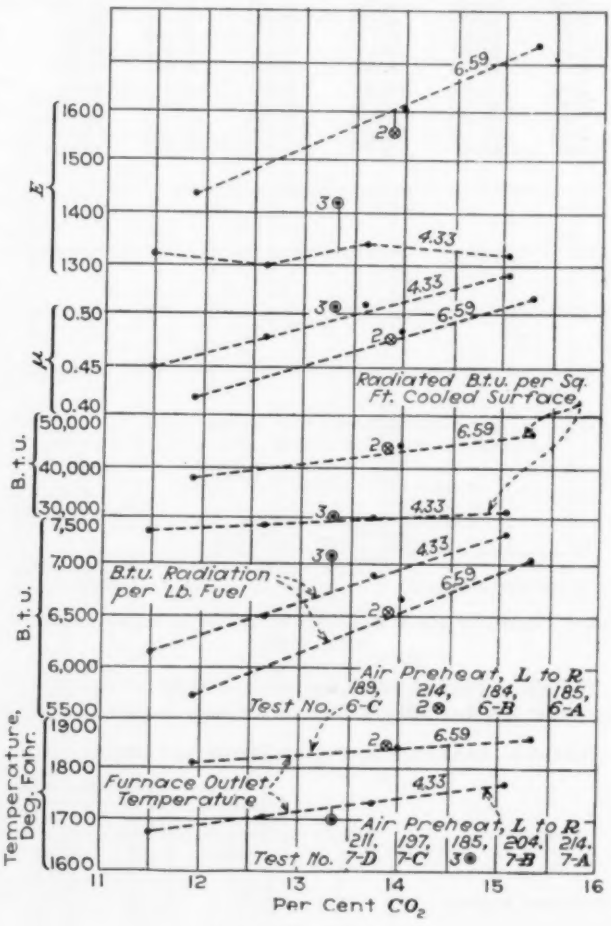


FIG. 8

[6] Mechanical Engineer, Jersey City, N. J.

hand, test 3 at the same station, with a firing rate practically equal to 6-A to 6-C, but with a preheat of about 30 deg. fahr. lower, has an effectiveness factor of 50 points lower than the point corresponding to its CO_2 on the line connecting the effectiveness factors of tests 6-A to 6-C. This proves that the effectiveness factor is manifestly influenced by the preheat. It may be contended that the experimental data of the tests mentioned are not sufficiently reliable to permit of any conclusion regarding the effectiveness factor. But this is not so. If there are plotted the values of furnace-outlet temperature, heat radiated per pound coal, and heat radiated per square foot of cooling surface of all these tests, a decided conformity in the relation of the values will be observed. So far as tests 7-A to 7-D and tests 6-A to 6-C are concerned, the values are practically on straight lines. The values of test 3 in their relation to the lines for tests 7-A to 7-D indicate the effect of the lower air preheat as an increase of heat absorbed by radiation due to the lowering of the furnace thermal level upon which the radiant heat is superimposed. Also, in general, the values check fairly well with Artsay's formula of radiant heat absorption. Therefore, we must conclude that the experimental data are at least fairly accurate.

The author is of the opinion that, with experimental and operating experience, judgment may be developed in the proper selection of the effectiveness factor and thus enable the determination of radiant heat absorption. This may be true, but it also is true that judgment may develop eventually to the point where one may dispense with the use of such a factor entirely.

OLLISON CRAIG.[7] The writer does not believe that it will ever be possible, or even desirable, in every-day practice, to calculate heat absorption by water-cooled walls in boiler furnaces from the empirical formulas based on basic laws of radiation. Mr. Hottel's paper gives a number of the variables that are encountered in problems of this kind and gives a formula for the calculation of heat transmission for each of these variable conditions.

The amount of work that would be involved in calculating total heat absorption in a given furnace, taking into account all of these variable conditions and applying each of these formulas, would be tremendous. Besides, there are other factors involved affecting the amount of heat that would be absorbed which are not taken into account in Mr. Hottel's paper.

It is quite probable that the accuracy obtainable, and even desirable can be obtained by means of empirical formulas and which will reduce the work of calculation enormously. The writer makes use of Orrok's modified Hudson formula and finds that the results check with sufficient accuracy in almost all cases in practice. To carry these calculations still further, the writer has reduced this formula to a series of curves, each curve representing 1 per cent of carbon dioxide and the coordinates the temperature rise of gases and heat release in the furnace per square foot of water-cooled surface.

Mr. DeBaufre has given experimental data for a number of installations and group these into a series of tests. These might be considered as divided into six series, three of the series being at Manchester—one at Brighton, one at Hell Gate, and one at Kip's Bay. In Table 3 of Mr. DeBaufre's paper he has tabulated the fraction of heat radiated for each test as calculated by the Orrok's Hudson formula and has shown the deviation based on both gross heat liberated and on the heat available.

The writer has calculated the percentage variation from the deviation based on the heat available and obtained the average percentage for each of the six series of tests mentioned. In the first series of tests at Manchester, being tests 1, 2, 5, and 6, the average percentage deviation is minus 8.3. The water-cooled surfaces were described in the paper as being comparatively clean. In the second series of tests at Manchester, being tests 7, 8, and 9, the percentage is plus 5.5, the paper stating that considerable amounts of slag had accumulated. The third series of tests, being tests 2, 3, 4, and 5, show a percentage deviation of plus 6.9, and the paper indicates an accumulation of slag as in tests 7, 8, and 9. In the tests at Brighton the percentage deviation was minus 18.5, and the paper states that the walls were kept very clean. In the test at Hell Gate the deviation was minus 8.3, with no comment as to the condition of the walls. For the tests at Kip's Bay the percentage deviation is plus 9.5, and the paper states that the water-cooled surfaces had accumulated a thicker layer of slag than at Hell Gate.

It is quite obvious of course that clean water-tube surfaces in furnace walls will absorb more heat than surfaces which have accumulated ash and slag, as is nicely indicated by the foregoing averages. This is a factor which the engineer cannot evaluate in advance and for which he must of necessity assume an average value.

Taking the average percentage deviation from Table 3, a figure of minus 3.3 per cent is obtained, this being for both clean surfaces and surfaces having more or less slag. The writer believes that it is fair to assume that the Orrok's Hudson formula will give results with sufficient accuracy as to cover the average case of water-cooled furnaces under average operating conditions with an average amount of slag and ash adhesion. If there is any error, it is probable that the actual absorption is somewhat greater than would be indicated by the formula, and for most purposes of calculation the error involved would be on the right side.

It would seem, then, that the use of Orrok's Hudson formula, or curves derived from this formula, would take into account the effectiveness factor as indicated in Figs. 3 and 4, with sufficient accuracy as regards the variation in rating, but would not give consideration to the slag condition of the tubes, the temperature of which is not under the control of the designer.

In this connection it is well known that in starting a new pulverized-coal furnace with clean water surfaces it is more difficult to obtain efficient combustion conditions and smokeless operation than can be obtained after the furnace has been in operation for a considerable period of time and after a certain amount of ash has accumulated on the water-cooled surfaces. Since the normal condition of operation is one in which a certain amount of ash deposit on the wall tubes exists, it might be desirable to arbitrarily produce a comparable condition during the initial stages of operation. The writer has found that this can be produced by spraying a thin film of refractory cement on to the tubes by use of a cement gun. The tubes of course should be clean. The film will gradually crack and fall away, but will be replaced with the ash accumulation when the falling away occurs. This method has been found to make the initial period of operation much easier.

Mr. Hottel has given one extremely interesting set of curves in Fig. 5. From curve 5 in Fig. 5 it is to be noted that if one row of bare tubes is placed around the furnace walls and spaced twice their diameter it might be expected that 90 per cent as much heat will be absorbed by these tubes as if the spacing was equal to the diameter or the tubes touching each other, this of course being on the condition that they are set out from the refractory wall and not imbedded in refractory material.

The writer had occasion to examine a section of a tube cut out from such a wall in which the spacing of the tubes was 1.3 times the diameter of the tube. In the particular installation water conditions were such that there was a considerable deposit of scale inside the tube. The amount of scale adhering to the

[7] Consulting Engineer, Riley Stoker Corporation, Worcester, Mass. Mem. A.S.M.E.

half of the tube facing the furnace was determined as well as the amount of scale in the half of the tube adjacent to the furnace wall. It was found that the outer half of the tube contained 1.6 times as much scale as the rear half of the tube.

It is reasonable to assume that there is some relation between heat absorption in a given surface and rate of scale deposit. The distribution of deposit in the tube referred to shows the effectiveness of heat absorption on the side of the tube adjacent to the furnace wall, which is confirmed by curve 5 in Fig. 5 of Mr. Hottel's paper.

The engineer makes use of heat-radiation and heat-absorption calculations in the case of water-cooled walls, either for the purpose of determining the amount of such surface necessary in order to maintain ash above its fusion temperature or to determine how much steam can be made in the walls and how much will need to be made in the convection portion of the boiler. In either case the absorption of a certain amount of heat is desirable for a particular installation. It is also desirable to obtain this absorption with the least expense.

The writer believes that this can be done by the use of bare tubes spaced apart and set out from the furnace wall. The use of Orrok's formula or curves derived from this formula will give results which are accurate within the limits of variables of operation.

N. C. Artsay.[8] The present trend in steam-generator designs is toward pure radiant-heat boilers with convection zones given to superheaters, reheaters, economizers, and air heaters. In these conditions the correct evaluation of heat absorption by the furnace envelope becomes of acute necessity. Several eminent authors have developed methods of applying the Stefan and Boltzmann formula to the furnace conditions. Wohlenberg and his collaborators, Haslam and Hottel in this country, Rozzak in France, Munzinger in Germany, have tried to embrace a number of factors and have developed cumbersome procedures requiring expert handling. Some doubts as to the validity of the main premises of such methods were already cast by Schack and Damour. The correct determination of furnace-gas temperatures is well-nigh impossible, and only the direct measurement of heat absorption by the total radiant surface is of any value; thus the mentioned methods are hard to verify due to the dearth of proper experimental work and data.

Broido and Orrok started from more general considerations in giving their empirical formulations of the radiant-heat absorption. Their results are sufficiently correct for most of the practical cases and are easy to obtain. Mr. DeBaufre's paper shows a method which is a combination of theory and empiricism and which is introducing a very fortunate term, the efficiency of furnace envelopes. The paper is so valuable that a searching criticism of its main premises is warranted. Let us take, first, its quantitative side.

DeBaufre's quantitative results stand or fall on the correctness of furnace-gas temperature determinations. Unfortunately, this item is the most unreliable of all the boiler test data. Many authors have pointed out the errors usually incurred in most elaborate pyrometric readings. High gas velocity thermocouples (Munzinger's suction pyrometers) have been analyzed in a specific case: gas at 1730 deg. fahr. true temperature, flowing in a bank of tubes, was subject to temperature measurement by different devices, among which a plain thermometer indicated 1290 deg. fahr., while suction pyrometers gave readings spread in between these limits. Temperature measurements in water-cooled furnaces are so sensitive to the uncontrollable conditions existing there that it is quite hazardous to use the readings for any but the roughest qualitative considerations.

The premises from which the formula is derived are open to criticism also.

A. Schack plainly writes that radiation is emitted from sources at higher temperature than that of the exit gas, and he takes the actual flame temperature and its surface. E. Damour investigating industrial ovens prefers the theoretical combustion temperature in the first term of the Stefan and Boltzmann formula, which is already much better. In his latest work on radiation (*Mechanical Engineering*, October, 1930, p. 915), Wohlenberg points out that radiation equilibrium of an oxidizing molecule with its ambient surroundings is reached earlier than the dynamical thermal equilibrium, and this comes to support the view that the initial radiation emission due to combustion is independent of the actual furnace temperature. It is easy to compute the radiation due to chemical processes is by far the greatest part of the total radiation and, hence, the use of exit-gas temperature is illogical.

It should be borne in mind that radiation laws holding for hot permanent surfaces have no connection whatever with radiation emitted by matter undergoing chemical processes. Unfortunately this is too often omitted or forgotten. The familiar sight of bright flames is responsible for the common misconception that the heat of oxidation is conveyed first to the mixture of products of combustion with residual air and then part of the heat is radiated away. Even a cursory glance at the process of oxidation if reduced to a molecular scale would show that this is impossible. The more probable scheme is that a group of molecules at the instantaneous chemical combination emits pulses of radiation until the resultant molecules of the product have time to separate and fly away with the remaining energy. Depending upon the properties and size of the furnace enclosure, a part of this initial radiation will be absorbed, and the remaining radiation will be returned via the CO, CO_2, H_2O constituents of gas and transformed by thermal diffusion into kinetic energy of the hot gas. The ensuing thermal radiation according to some power of absolute temperature will be superimposed on the initial or combustion radiation. The amount of radiant energy in this thermal radiation is small compared with the amount released by the oxidation, and hence we are facing the still more complex question of radiation absorption on the water-cooled walls. It is thus apparent that in any furnace where fuel is burning at a sensible rate there is an excess of radiant heat over the capacity of the envelope to absorb radiant heat, and in such conditions we do not need to study the emission. It is the extent and quality of the water-cooled surface which are of utmost importance, and the society should be indebted to Mr. De Baufre for pointing out the effect of slagging, the first time in such investigations of radiant-heat exchange. It should be remembered that the bulk of radiant energy is of low frequency way down in the infra-red, and for these wave lengths the absorbing capacities of metals are surprisingly low.

Thus the solution of the radiant-heat exchange problem lies, not in temperatures or solid angles, but in the change of absorbing capacity in the layers of slag rust and pure metal on the water-cooled surfaces.

W. G. Wohlenberg.[9] On the first page the author refers to the paper entitled "Radiation in the Pulverized-Fuel Furnace," which was prepared by Mr. Donald Moran and myself and published in Vol. 47 of the Transactions. On page 148 of the latter paper there appears an expression which, for short, is designated by the symbol μ_3. To quote from this paper:

[8] Assistant to Geo. A. Orrok, Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

[9] Professor of Mechanical Engineering, Sheffield Scientific School, Yale University, New Haven, Conn. Mem. A.S.M.E.

"This ratio measures the total heat absorption at the cold surface as compared to that striking this surface by a black fuel bed immediately in front of it. The quantity μ_3 is thus the relative absorption of the furnace as compared to that of a total black furnace with the same amount of cold surface."

Values of this coefficient as computed in the Wohlenberg and Moran paper vary from about 40 to something over 80, depending on the conditions, but in no case does the coefficient value exceed one, and this is as it should be when radiation is the sole mode of heat transfer. The interesting point now is that Mr. DeBaufre's effectiveness factor, except for the units in which it is expressed, is exactly this coefficient. For black-body radiation the relative absorption coefficient μ_3 has a value of unity and the De Baufre effectiveness factor E has a value of 1720. This, then, is the number which relates the two factors.

DeBaufre reports values of E as high as 2300, and in a number of cases the values are above the black-body figure of 1720. It is conceivable that under certain conditions radiation intensities in excess of black-body intensities could exist, but they certainly do not exist in the pulverized-fuel furnace. Then such head-absorption figures either are wrong, the associated temperature measurements are wrong, or heat is absorbed in an appreciable amount by a process other than radiation. No doubt there may be some errors in measurements of conditions, as we all know how difficult it is to measure furnace temperatures. But the divergence from black-body conditions seems too large and this appears so in too many different cases. The only other tenable conclusion is that there must in such cases have been a considerable heat transfer by convection, which, added to the normal radiation transfer, accounts for the very high total heat transfer.

In this connection it is significant to note that such results were obtained in furnaces with tangential firing under which conditions there is at least the possibility for a much stronger convection current of the furnace gases as they sweep directly over the water-cooled furnace walls than for some other types of firing. That point is, I think, worth noting.

The author in setting up charts (Figs. 3 and 4) from which values of the effectiveness factor should be determined relies of course wholly on the method of empiricism. He does not at the outset attempt to expand the factor into an expression containing within it terms which represent each of the conditions a variation of which may influence the value of the factor. This of course is the method which was employed in the radiation papers to which the author refers and which, he says, show how complicated the subject becomes when such an attempt is made. Not to expand such an expression may, however, prove dangerous. Such a factor as E is really but a single symbol for a large number of variable conditions that enter into the heat-transfer process in the furnace. How dangerous a bulk consideration of the phenomena may prove becomes evident from a closer investigation of the data shown in Figs. 3 and 4.

In these curves are shown relating the value of the effectiveness factor to the heat release. In each chart the upper curve, which is for clean surfaces, shows an increasing effectiveness of this surface as the heat-release rate is increased. The lowest curve, which is for dirty surfaces, shows a constant value for the effectiveness factor.

Without going into the mathematics of the situation now, let it suffice to say that it may be shown in a simple manner that it is the clean surface condition, or upper curve in the chart, which should show a constant effectiveness factor value. If the surface is clean and has black-body surface characteristics, then this curve should be represented by a horizontal line $E = 1720$.

The curves representing the dirty surfaces should not show an increasing value of E with increase in heat release, but rather a decrease of this value with increase in heat release. It may now be noted that the experimental point shown may be fitted just as closely by sets of curves with the latter set of specifications as they will the specifications which the author's curves fit. Empiricism might thus lead one to construct either set, of which the author has chosen the wrong set. Other curve sets might likewise be fitted into the points as closely as do the author's, but only one trend condition is admissible from the point of view of the fundamental principles involved, and so this is an example of the slips that may enter into any system based primarily on empirical methods.

The writer does not wish to imply here that as a result of this the author's method is all wrong. The curves shown in Figs. 3 and 4 certainly have wrong trends and should, he thinks, be corrected. When corrected in accordance with the above specifications as to trends, they will fit this specification:

(1) For clean heat-absorbing surfaces, the effectiveness factor of the cold surface in a given furnace is a constant
(2) For dirty or covered surfaces, the effectiveness factor of the surface decreases with increasing heat-release rates.

It will be noted that in each of the foregoing statements the qualifying cause, "for a given furnace," is introduced. There should be other qualifying causes, such as for a given fuel, for a given excess air, for a given air temperature, etc.

Thus the simpler the resulting formulas under such conditions the more must be left to the judgment of the man who attempts to use it. In fact, for a process as complex as is this one, simple formulas probably must leave so much to the judgment of the individual that he is hopelessly lost unless he is an expert in the field.

In closing, the writer wishes to present just one more thought and this is it: A comprehensive analysis of any situation in which account is taken of all of the factors involved results in a theory and in formulas which leave less and less to the judgment of the individual according as the theory approaches perfection. This of course is obvious. If a theory developed on such a basis becomes so formidable in appearance that the results of the theory (such as formulas, etc.) are difficult of application, this in itself is a very good indication that in such a problem a minimum gap should be left to the judgment of the individual. The degree of complexity of the theory is usually a direct measure of the difficulty which would be required in arriving at results based on judgment. The boiler-furnace problem is just of this sort, and those who would use simple relations for its evaluation must not forget the magnitude of errors into which they might fall.

Author's Closure

As stated by Professor Christie, the "effectiveness factor" proposed in this paper is an empirical value to be selected by the furnace designer with judgment based on the condition of the heat-absorbing surface, etc. The value of 1500 was selected for a heat release of 28,000 to 30,000 B.t.u. per hr. per cu. ft. of furnace volume because this value is well within the data obtained for clean or slightly dirty bare surfaces with horizontal turbulently fired pulverized-fuel furnaces, and is therefore a safe value for design purposes under these conditions. The value of 500 was selected to represent a very dirty furnace because this value is below the lowest values obtained with considerable slag adhering to the water-cooled surfaces, and therefore represents conditions worse than actually have been encountered.

Table 3 is a comparison of experimental data with results calculated by Hudson's formula and is not a comparison of the "effectiveness factor" with this formula, as stated by Professor Christie.

Professor Wohlenburg unfortunately confuses the effectiveness factor based on the furnace outlet temperature with his ratio μ_8 based on the furnace temperature immediately in front of the heat-absorbing surface. Experiments not described in the present paper have shown that the heat absorption based upon the temperature immediately in front of the heat-absorbing surface corresponds to about one-half the black-body coefficient. But the furnace temperature varied from about 2500 deg. fahr. at the zone of most intense combustion to about 1800 deg. fahr. at the furnace outlet. Basing the heat absorption on the furnace outlet temperature necessitates the use of an "effectiveness factor" approximately equal to the black-body coefficient, although the actual factor is one-half this coefficient when based on the actual temperature of the flame in front of the surface absorbing heat.

This point is covered in the paper among the reasons given for the fourth-power radiation law not strictly applying in the way used.

When the furnace is operating nearly full of flame, the heat-absorbing surfaces are evidently in a better situation for effective heat absorption than when the flame only partly fills the furnace volume at reduced ratings if the surface of the flame is a factor in the amount of energy radiated. This was taken as an explanation of the falling off in the value of the effectiveness factor based on the area of the absorbing surface as indicated by the experimentally determined points for reduced ratings with comparatively clean surfaces. When the surfaces in the most intense zone of combustion contain more slag than those in cooler parts of the furnace, the flame at reduced ratings apparently does not contract to the same extent as with clean surfaces, so that the effectiveness of the water-cooled surface does not fall off to the same extent. With certain gas-fired furnaces, the effectiveness factor has been found to increase at reduced ratings, but this is attributed to combustion within the furnace being incomplete at the higher ratings so that not as much heat was available for radiation as the heating value of the fuel would indicate. In calculating the performance of steam-generating units at reduced ratings with pulverized fuel, a constant effectiveness factor of 1500 gives temperatures which drop off more rapidly with reduction in rating than experience would indicate to be so. It is therefore believed that the trends of the curves in Figs. 3 and 4 are not as wrong as Professor Wohlenburg concludes.

With regard to the curves plotted by Mr. Weisselberg, it is unfortunate that his point for run 6-A was plotted for an effectiveness factor of 1740 instead of 1640 as given in the paper. If correctly plotted, little difference would be seen between 1640 for 15.35 per cent CO_2 and 1610 for 14.02 per cent of CO_2. There is an appreciable drop, however, between 1610 for run 6-B with 14.02 per cent CO_2 and 1440 for run 6-C with 11.94 per cent CO_2. This may have been due to accumulation of slag in the furnace, which would also account for the smaller difference between runs 6-A and 6-B. It will be noticed that the other values plotted do not line up exactly although plotted to a less open scale than used for the effectiveness factor. However, the lines for runs 6-A, 6-B, and 6-C are nearly parallel to the lines for runs 7-A, 7-B, 7-C, and 7-D, thus indicating that if the temperatures and heats radiated were calculated for a constant value of the effectiveness factor, nearly the same curves would be obtained. One advantage of the use of the effectiveness factor is that a considerable variation in its value affects the quantities determined by means of it to a much smaller extent; hence, a reasonably close choice will give fairly accurate results.

Mr. Craig is correct in concluding that the experimental data in this paper confirms the use of the Hudson formula as representing on an average the heat absorbed by water-cooled bare surfaces in boiler furnaces. For many purposes, an approximate average value meets the requirements, particularly for stoker-fired furnaces. But for more nearly accurate calculations, particularly with pulverized fuel, some differentiation is necessary for various furnace conditions and arrangements. The use of various effectiveness factors provides such a differentiation.

In the present state of the art, more or less empirical methods are necessary to divide the heat absorbed by radiation in the furnace from that remaining in the products of combustion leaving the furnace. Methods having a better theoretical basis may come into use in the future through the efforts of such students of the subject as Mr. Artsay.

While the method of calculation described in this paper may seem rather laborious, it is possible to plot curves for typical fuels which render the determination of the furnace-outlet temperature based on the "effectiveness factor" a very simple matter.

FSP-53-19b

Radiant Heat Transmission Between Surfaces Separated by Non-Absorbing Media

By H. C. HOTTEL,[1] CAMBRIDGE, MASS.

Fundamental laws of radiation from solids have been discussed and applied to individual problems of heat transfer in numerous papers. In this one the author indicates just what principles are involved, to how many problems they have been applied, and gives a number of applications of the principles to problems in industrial-furnace design.

Fundamental principles and definitions discussed include: The definition of normal intensity, Lambert's cosine principle, relation between intensity and emissive power, the square-of-the-distance law, Kirchoff's law, and the Stefan-Boltzmann law. Various surface arrangements are divided into two groups: (1) in which one of the two surfaces is small relative to the distance separating them, and (2) in which both surfaces are of considerable extent relative to their distance apart. Various cases of these and the effect of incomplete absorption are treated mathematically. The task of applying many of the equations given in the paper is simplified by presentation of numerous graphs.



THE fundamental laws of radiation from solids have been treated in many textbooks, and have been applied to individual problems of heat transfer in numerous articles in technical and scientific journals (2–7).[2] No satisfactory presentation has been made, however, of the subject in a form indicating just what principles are involved and to how many problems they have been applied; and a number of applications of the principles to problems of some importance in industrial furnace design have never been made. A summary of the results on the more important problems was presented briefly in a recent paper (11), but no indication was given of the method of obtaining the results. Inasmuch as an intelligent use of the results and the application of the principles to new problems depends on familiarity with the method of obtaining the results, it is felt that they merit a more detailed treatment than it was possible to give them in the paper referred to.

The treatment of problems in radiant heat transmission between surfaces involves the application of certain fundamental principles or definitions which may be grouped into (*a*) principles of photometry and illumination, and (*b*) principles of thermal radiation. The first group includes:

1 *The definition of normal intensity*, as being the energy emitted per unit area of emitting surface per unit solid angle per unit of time, in a direction normal to the surface. Mathematically, this says:

$$dq = i\, dA\, d\omega$$

in which q is energy per unit time, i is normal intensity, A is area, and $d\omega$ is a differential element of the solid angle above dA and normal to dA.

2 *Lambert's cosine principle*, which states that if the cone of radiation dq lying within the solid angle $d\omega$ is not normal to the surface dA but makes an angle θ with the normal to the surface, then the above equation must be modified to

$$dq = i\, dA \cos\theta\, d\omega \qquad [1]$$

The term $dA \cos\theta$ is the projected area of dA on a plane normal to the beam of radiation. The cosine principle, which amounts to the assumption that the intensity i of Equation [1] is independent of the angle θ and is equal to the normal intensity, is not exact except for "black" surfaces, or perfect radiators, but is in small error for most surfaces met in heat-transfer problems (7). The magnitude of the error will be discussed later.

3 *The relation between intensity and emissive power* of a surface, depending on the definition of the latter as being the amount of energy emitted per unit of time per unit of area in all directions throughout the solid hemispherical angle above the surface. It is designated by E, and is sometimes called hemispherical intensity instead of emissive power. Obviously the integration of Equation [1] to include the total hemisphere above dA should lead to the relation between intensity i and emissive power E.

$$E\, dA = i\, dA \int \cos\theta\, d\omega$$

Any textbook on photometry gives the integration, which leads to the relation $E = \pi i$.

4 *The square-of-the-distance law*, which follows directly from Equation [1] and is sometimes more useful. If the beam of radiation from dA, lying within the solid angle $d\omega$, is intercepted by a surface element dA' at a distance r from dA, and the line of length r connecting dA and dA' makes an angle θ' with the normal to dA', then $d\omega$, by definition of a solid angle, may be placed equal to $dA' \cos\theta'/r^2$.

Equation [1] then becomes

$$dq = i\, \frac{(dA \cos\theta).(dA' \cos\theta')}{r^2} \qquad [1a]$$

which states that the radiation from one small surface element to another is proportional to the product of their areas projected on the planes normal to the line connecting them, and is inversely proportional to the square of the distance separating them. This is generally referred to as the square-of-the-distance law.

The principles of thermal radiation necessary to the solution of the problems at hand include:

5 *Kirchoff's law*, which states that the emissive power of a surface divided by its absorptivity is the same for all surfaces at

---

[1] In Charge of Course in Fuel and Gas Engineering, Massachusetts Institute of Technology. Professor Hottel received the degree of A.B. in Chemistry from Indiana University in 1922, and that of M.Sc. in Chemical Engineering in 1924 from the institution with which he is at present connected.

[2] Numbers in parentheses refer to bibliography at end of paper.

Contributed by the Fuels Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

a given temperature, and is equal to the emissive power of a "black" body or perfect radiator. The absorptivity of a surface is defined as the fraction of the radiation incident on it which it absorbs. If we define emissivity as the radiation from a surface divided by the radiation from a black body at the same temperature, we see from Kirchoff's law that emissivity and absorptivity are numerically equal. The law is proved in many textbooks on physics; it is frequently misapplied in radiation problems not involving equilibrium radiation; and the extent to which the result is an approximation is not stated, and in some cases not realized.

6 *The Stefan-Boltzmann law*, which states that the emissive power of a black body is proportional to the fourth power of its absolute temperature. This law was determined by Stefan in 1879 from Tyndall's experiments, and was proved on thermodynamic grounds by Boltzmann in 1884.

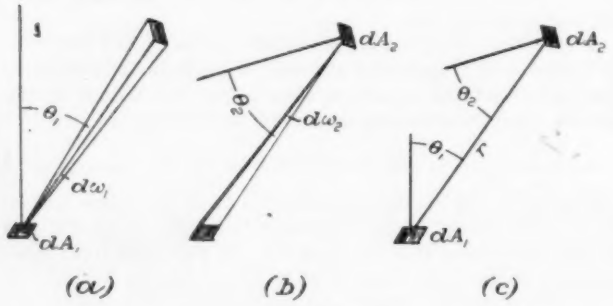


FIG. 1

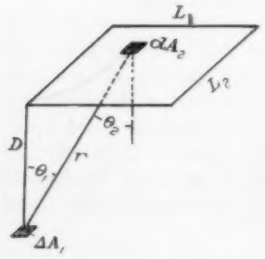


FIG. 2

From Equation [1] the radiation from a surface A_1 to another surface A_2 is

$$q = i_1 \int dA_1 \cos \theta_1 \, d\omega_1$$

with the integration taken over the whole area A_1 and throughout the total angle ω_1 subtended by A_2 [see Fig. (1*a*)]. The reverse radiation from A_2 to A_1 is

$$q = i_2 \int dA_2 \cos \theta_2 \, d\omega_2$$

with the integration taken over the whole area A_2 and throughout the total angle ω_2 subtended by A_1 at A_2 [see Fig. (1*b*)]. But we know from the equivalence of Equations [1] and [1*a*] that either of the above integrals might have been evaluated in the form

$$\int \frac{dA_1 \, dA_2 \cos \theta_1 \cos \theta_2}{r^2} \quad \text{..... [see Fig. (1c)]}$$

and that they are therefore equal. Remembering that $E = \pi i$ and that for black surfaces $E = \sigma T^4$, we may express the net heat interchange between two black surfaces A_1 and A_2 in any of the three forms

$$q = \frac{\sigma}{\pi} (T_1^4 - T_2^4) \int dA_1 \cos \theta_1 \, d\omega_1 \quad \text{.....[2a]}$$

or

$$q = \frac{\sigma}{\pi} (T_1^4 - T_2^4) \int dA_2 \cos \theta_2 \, d\omega_2 \quad \text{.....[2b]}$$

or

$$q = \frac{\sigma}{\pi} (T_1^4 - T_2^4) \iint \frac{dA_1 \, dA_2 \cos \theta_1 \cos \theta_2}{r^2} \quad \text{.....[2c]}$$

The choice among them is determined by the relative ease of evaluation of the different integrals for the problem at hand.

In applying the principles discussed so far to the solution of problems of heat interchange in systems consisting of various surface arrangements, we find it convenient to treat separately the two factors, shape of the system and emissivity of the surfaces composing it, by considering temporarily that the surfaces with which we are concerned are perfect radiators and absorbers, i.e., black bodies, and that therefore only the factor of shape enters.

Effect of Shape of System

We may classify the various surface arrangements into two groups, in the first of which one of the two surfaces is very small relative to the distance separating the two, and in the other of which both surfaces are of considerable extent relative to their distance apart. In the first group, Equation [2] will require integration over but one area. The various cases follow:

Case 1. Interchange between a small surface element ΔA_1 and a parallel plane A_2 of infinite extent. The problem of integrating Equation [2*a*] to include the total hemispherical angle above A_1 is treated in many textbooks, and leads to

$$q = \sigma(T_1^4 - T_2^4) \, \Delta A_1 \quad \text{.....[3]}$$

Case 2. Element ΔA_1 and a circular disk of radius a in a plane parallel to the plane of ΔA_1 and distant from it the distance K. Let x be the distance from ΔA_1 to the normal through the center of the disk. The illumination problem analogous to this has been treated by Foote (3) and leads to

$$q = \sigma(T_1^4 - T_2^4) \, \Delta A_1 \left\{ \frac{1}{2} - \frac{x^2 + K^2 - a^2}{2\sqrt{x^4 + 2(K^2 - a^2)\,x^2 + (K^2 + a^2)^2}} \right\} \quad \text{.....[4]}$$

Case 3. Element ΔA_1 and a rectangle in a plane parallel to the plane of ΔA_1, with a corner of the rectangle contained in the normal to ΔA_1. Equation [2*c*] is of the form most readily applied. Fig. 2 defines the terms appearing in Equation [2*c*].

$$dA_2 = dx \, dy$$

$$r = \sqrt{x^2 + y^2 + D^2}$$

$$\cos \theta_1 = \cos \theta_2 = \frac{D}{\sqrt{x^2 + y^2 + D^2}}.$$

Substituting these terms into [2*c*], we have

$$dq = \frac{\sigma(T_1^4 - T_2^4)}{\pi} \frac{\Delta A_1 \, dx \, dy \, D^2}{(x^2 + y^2 + D^2)^2}$$

If the rectangle has sides L_1 and L_2, the radiant heat interchange is given by

$$q = \sigma(T_1^4 - T_2^4) \, \Delta A_1 \cdot \frac{D^2}{\pi} \int_0^{L_1} \int_0^{L_2} \frac{dx \, dy}{(x^2 + y^2 + D^2)^2}.$$

On integrating this and redefining L_1 and L_2 as the ratio of each side to the distance separating the planes, we have

$$q = \sigma(T_1^4 - T_2^4) \, \Delta A_1 \frac{1}{2\pi} \left[\frac{L_2}{\sqrt{L_2^2 + 1}} \sin^{-1} \frac{L_1}{\sqrt{L_1^2 + L_2^2 + 1}} + \frac{L_1}{\sqrt{L_1^2 + 1}} \sin^{-1} \frac{L_2}{\sqrt{L_1^2 + L_2^2 + 1}} \right] \quad \text{.....[5]}$$

Case 4. Element ΔA_1 and any rectangle in a plane parallel to

ΔA_1. This can be treated as above by dividing the rectangle into four rectangles having a common corner above ΔA_1. If the large rectangle lies completely outside the normal to ΔA_1, the same principle is applicable if we remember to add areas algebraically.

Case 5. Element ΔA_1 and any surface generated by a line of infinite length moving always parallel to the plane of ΔA_1. This case has considerable industrial applicability, and includes radiation to a nest of tubes whose length is great compared to the distance from them to the radiator, radiation from banks of strip and rod heaters in electric furnaces, etc.

In Fig. 3 the surface element ΔA_1 is at the origin, in the XY-plane. Line ABC represents the trace, in the XZ-plane, of the infinite line generating the surface A_2. The figure defines the terms of Equation [2*a*] as follows:

$$d\omega_1 = \frac{\rho^2\, dy\, d\theta}{(\rho^2 + y^2)^{3/2}}$$

$$\cos\theta_1 = \frac{\rho\cos\theta}{\sqrt{\rho^2 + y^2}}$$

$$d\omega_1 \cos\theta = \frac{\rho^3 \cos\theta\, d\theta\, dy}{(\rho^2 + y^2)^2}$$

Substituting into Equation [2*a*], and inserting the limits of $\pm\ \infty$ for y, we have

$$q = \sigma\,(T_1^4 - T_2^4)\ \Delta A_1 \frac{1}{\pi} \int_{-\infty}^{+\infty} \int_{\theta''}^{\theta'} \frac{\rho^3 \cos\theta\, d\theta\, dy}{(\rho^2 + y^2)^2} \quad [6]$$

which on integration becomes

$$q = \sigma(T_1^4 - T_2^4)\,\Delta A_1 \frac{(\sin\theta' - \sin\theta'')}{2} \quad [6a]$$

It is to be noted from Equation [6*a*] that the evaluation of q for this case does not necessitate the use of a three-dimensional still calculations. Fig. 4 shows a cross-section through a bank of infinitely long tubes. It is desired to find the heat interchange between the surface element ΔA_1 of the lower plane and the tubes above. The shaded areas represent intercepted beams. If a circle of unit radius is drawn, inspection of the figure indicates

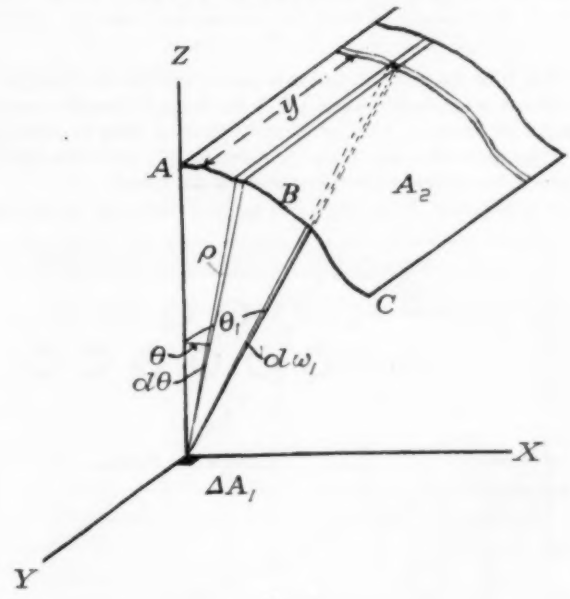


FIG. 3

that for any tube the expression $(\sin\theta' - \sin\theta'')$ is equal to the projection, on the base line, of that arc of the unit circle which lies between the bounding lines of the tube. The sum of the projections, indicated, divided by the diameter of the

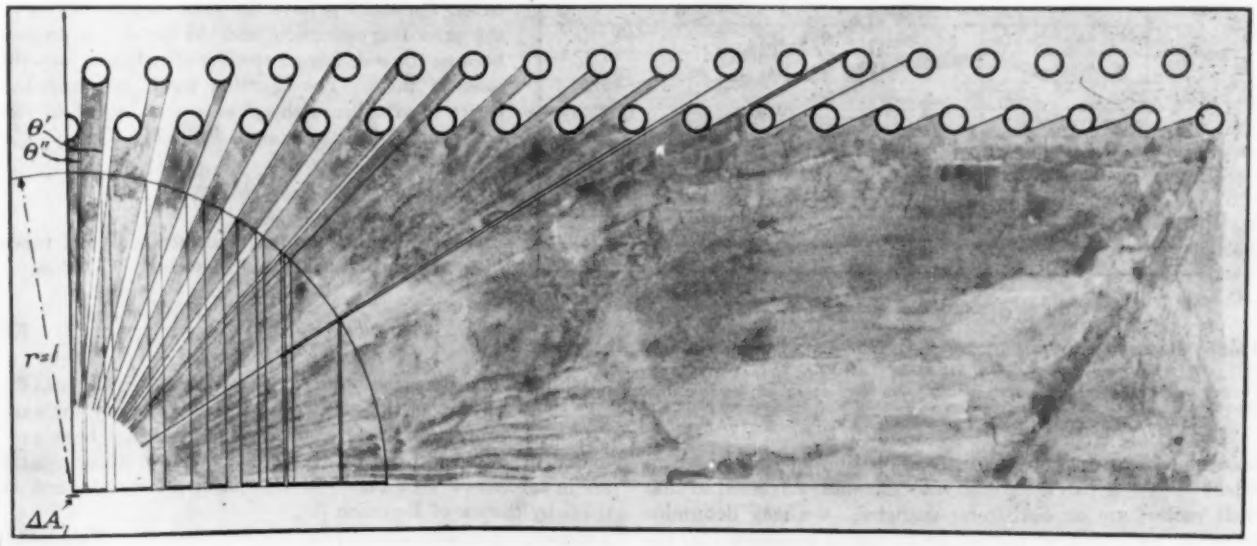


FIG. 4 RADIANT HEAT TRANSFER TO A BANK OF TUBES

picture, but simply of the cross-section through ΔA_1 and surface A_2 taken normal to the generating line of surface A_2. The angles θ' and θ'' are angles in this cross-sectional picture, made by the normal to ΔA_1 with the lines joining ΔA_1 to the edges of A_2.

A graphical solution of problems falling in this class is rapid, and frequently useful in steam-boiler, cracking-coil, and tube-circle, gives a factor F_A such that the heat interchange between the surface element ΔA_1 and the tube bank is

$$q = \sigma(T_1^4 - T_2^4)\,\Delta A_1\, F_A \quad [7]$$

This method may be used to determine quickly the distribution of heat to different rows of tubes in a setting, and to study the effect of spacing on the distribution. When only one row of

equally spaced tubes is involved and the tubes are of infinite extent in both directions (infinite number of tubes of infinite length), the solution of the problem need not be made graphically. The factor F_A for such an arrangement is given by

$$F_A = \frac{K + \tan^{-1}\sqrt{K^2-1} - \sqrt{K^2-1}}{K} \quad \ldots\ldots [7a]$$

in which K is the ratio of center-to-center distance to diameter. This factor is applicable when ΔA_1 is far enough from the tubes to make its location, whether directly under a tube or directly under the space between tubes, inconsequential; or to the whole surface A_1 regardless of its distance from the tubes.

The application of the graphical method discussed above has

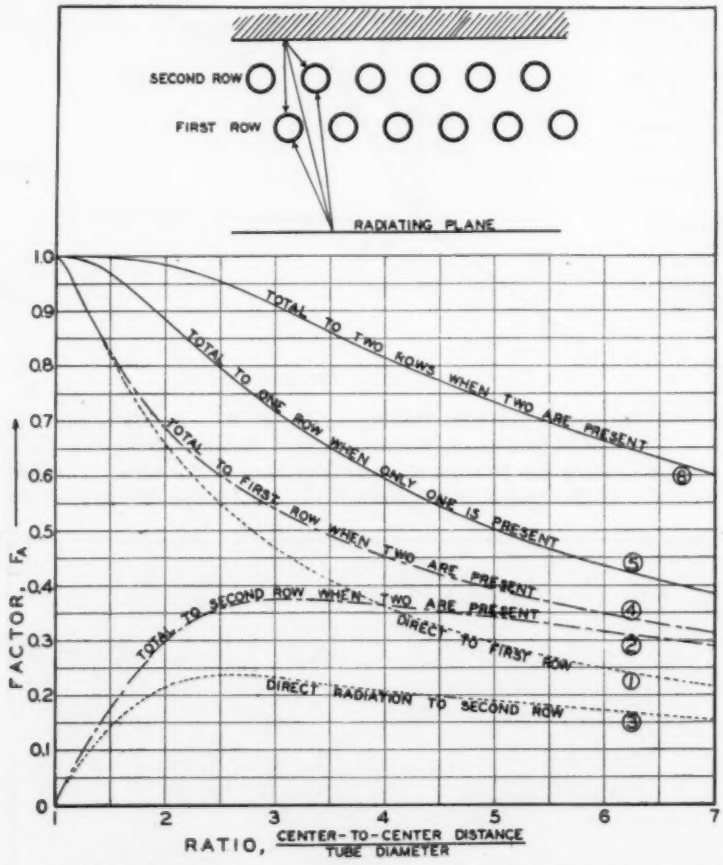


FIG. 5 RADIATION FROM A PLANE TO ONE OR TWO ROWS OF TUBES ABOVE AND PARALLEL TO THE PLANE

(Tubes on equilateral-triangular centers; non-conducting but reradiating refractory surface above tubes. Ordinate expressed on basis of heat transferred from a plane to a plane replacing the tubes, or to an infinite number of rows of tubes.)

been made to two problems in heat distribution. Let us consider the case of two horizontal rows of tubes, staggered so that their centers are on equilateral triangles. We may determine either by the graphical method or by use of Equation [7a] the fraction F_A, of the radiation from a plane below, which is intercepted by the first row of tubes. The graphical method also permits a determination of the fraction intercepted by the second row. These fractions are presented in Fig. 5, lines 3 and 1, as functions of the dimension ratio, center-to-center distance/tube diameter. We have in addition to consider the effect of the presence of a back wall above the tubes. Let F_{A1} and F_{A2} represent the fractions of the direct radiation from below which are intercepted by the first (lower) and second rows, respectively. The fraction $1 - F_{A1} - F_{A2}$ passes through both rows and is intercepted by the back wall, which assumes a temperature intermediate between that of the lower radiator and the two rows of tubes. The back wall, on the assumption of no external loss, reradiates downward an amount of radiation equal to that which it absorbs, i.e., the radiation $1 - F_{A1} - F_{A2}$. The top row, which now is the first row, intercepts the fraction F_{A1} or the amount $F_{A1}(1 - F_{A1} - F_{A2})$. The total fraction F_2 picked up by the top row is therefore $F_{A2} + F_{A1}(1 - F_{A1} - F_{A2})$; similarly, the total fraction picked up by the lower row is $F_{A1} + F_{A2}(1 - F_{A1} - F_{A2})$. These results are included in Fig. 5.

The results on a second problem in heat distribution are presented in Fig. 6. If we change from two rows of tubes, staggered, to two rows in which the tubes of the top row are directly above the lower tubes, we may expect a change in the distribution of heat. Fig. 6 presents the results, not for a variable ratio of center-to-center distance to tube diameter, but for a constant value of 2 (clearance equals diameter), with the variable taken as the distance between rows.

The results presented in Fig. 5 (or 6) are applicable to the solution of an entirely different type of problem. We have been considering radiation from a plane to rows of tubes, basing the heat transfer on the area of the plane. In the design of electric resistor furnaces, rows of rods are frequently used as resistors, and it is desirable to know the radiating power of the rods. If we consider the two rows of rods at temperature T_1 surrounded on both sides by the ultimate heat-receiving surface at temperature T_2, the radiant-heat-transfer equation would be

$$q = \sigma(T_1^4 - T_2^4)2A_p(F_{A1} + F_{A2}) \quad \ldots\ldots [8]$$

in which A_p is the area of a plane replacing the tubes, F_{A1} and F_{A2} have the same significance as in the preceding discussion, and the factor 2 is present because the rods have a receiver of radiation on both sides of them. The equation would be somewhat easier to visualize with an area term equal to the actual area of the rods, A_R. From Fig. 5,

$$A_p = A_R \frac{d}{2\pi D}$$

in which d equals center-to-center distance of tubes in a row and D equals tube diameter. Therefore

$$q = \sigma(T_1^4 - T_2^4)\, A_R \left[\frac{(F_{A1} + F_{A2})d}{\pi D}\right] \ldots\ldots [9]$$

The bracketed term may be called the "surface efficiency" of the rods. The surface efficiency for one- and two-row arrangements of rods has been presented in graphical form by Keene and Luke (5) whose results are in agreement with Fig. 5 of this paper after conversion is made by the use of Equation [9].

Case 6. Element ΔA_1 and any surface whatever. This is the general case covering all those which have preceded. Herman in his textbook on geometrical optics (1) discussed a method of graphical evaluation of the problem, depending on the fact that if a unit hemisphere is constructed above the element ΔA_1, and a straight line with one end fixed at ΔA_1 is made to move around the edge of A_2, then the projection, on the base of the hemisphere, of the area on the hemisphere marked off by the moving line, is equal to the integral $\pi \int_{A_2} d\omega \cos\theta$. Nusselt (8)

later described the same method. There is considerable difficulty, however, in making such a three-dimensional construction; and the author (11) has described a simple mechanical device to be used with a small cardboard model of the surface under consideration. The device, not made with particular care, gives results agreeing within 1 per cent with those obtained for some of those special shapes above which can be solved algebraically.

In the preceding group of surface arrangements the integration of Equation [2] was performed over the area A_2 only. In all that follow both surfaces are of considerable extent, and the integration will therefore be made over both A_1 and A_2. The simplest case is

Case 7. Infinite parallel planes. The integration of Equation [3] over the area A_1 leads to the familiar

$$q = \sigma(T_1^4 - T_2^4)\, A$$

Case 8. Two continuous surfaces, one enclosing the other. If the enclosed surface contains no negative curvature, i.e., if no elements of its surface can "see" anything but the other surface, the integration of [2a] leads simply to

$$q = \sigma(T_1^4 - T_2^4)\, A_1 \qquad \ldots\ldots [8]$$

in which A_1 is the area of the enclosed surface.

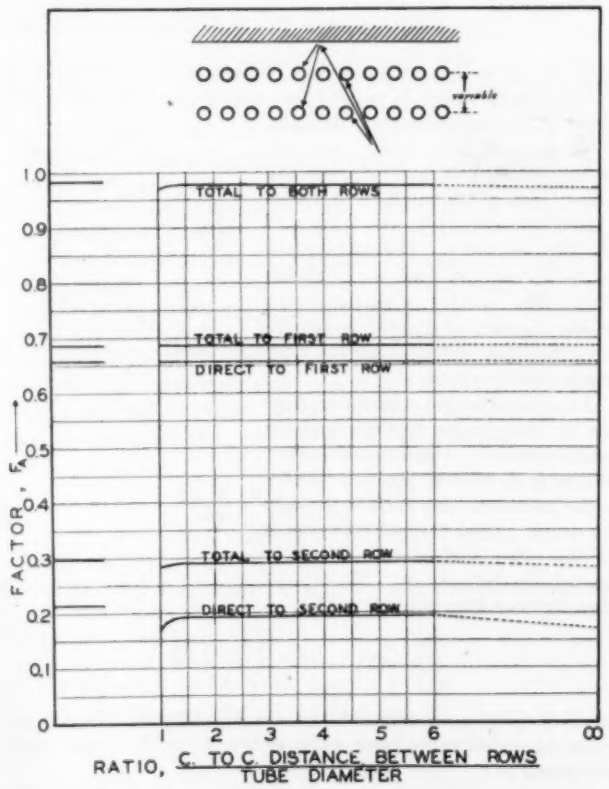


FIG. 6 EFFECT OF DISTANCE BETWEEN TUBE ROWS ON RADIANT HEAT DISTRIBUTION TO TWO ROWS OF TUBES, WHEN TUBES ARE DIRECTLY BEHIND ONE ANOTHER

(Ratio (center-to-center distance of tubes in a row)/(tube diameter) held constant at 2.0. On extreme left are given results, for comparison, on two rows, staggered, spaced with tube centers on equilateral triangles, with ratio just mentioned = 2.0.)

Case 9. Circular disks in parallel planes, with their centers on the same normal to their planes. Case 2, Equation [4], above, gave the radiation from a point to a disk. The integration of this over the circular area A_1 of radius b leads to

$$q = \sigma(T_1^4 - T_2^4) A_1 \frac{1 + \frac{a^2}{b^2} + \frac{K^2}{b^2} - \sqrt{\left(1 + \frac{a^2}{b^2} + \frac{K^2}{b^2}\right)^2 - 4\frac{a^2}{b^2}}}{2} \qquad \ldots\ldots [9]$$

This result has been given in slightly different form by H. B. Keene (2) who also pointed out that when the radii a and b of the disks are small compared to K, their distance apart, Equation [9] becomes

$$q = \sigma(T_1^4 - T_2^4) \frac{A_1 A_2}{\pi K^2} \left\{1 - \frac{a^2 + b^2}{K^2}\right\} \qquad \ldots\ldots [9a]$$

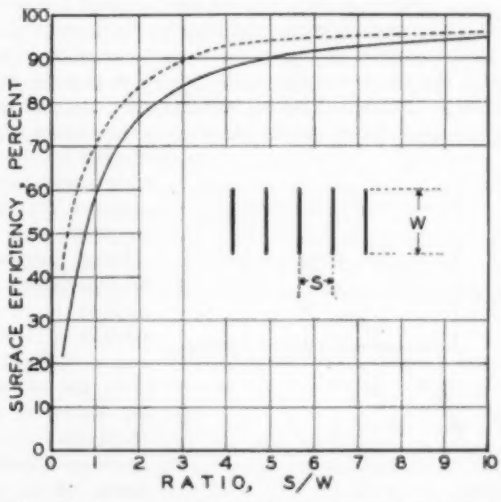


FIG. 7 CURVE OF SURFACE EFFICIENCIES FOR STRIPS

When the disks have the same radius and the ratio of diameter to distance apart is R, Equation [9] becomes

$$q = \sigma(T_1^4 - T_2^4)\, A_1 \left[1 - \frac{2}{R_2}\left(\sqrt{1 + R^2} - 1\right)\right] \qquad \ldots [10]$$

This problem has been attacked by Keller (6) in connection with the evaluation of radiation losses through furnace openings, but the approximation method there suggested for evaluating radiant-heat interchange between parallel disks is more tedious than the use of Equation [10], and less exact.

Case 10. Rectangles of equal size in parallel planes, directly opposed. By making use of Equation [5], case 3, the radiation from an element dA to a parallel rectangle with corner above dA; by applying the principle of case 4; and by integrating over the area A_1, we finally arrive at the interchange between two equal rectangles:

$$q = \sigma(T_1^4 - T_2^4) A \frac{1}{\pi}\left\{\frac{1}{XY}\log\frac{(X^2+1)(Y^2+1)}{(X^2+Y^2+1)} - \frac{2}{Y}\tan^{-1}X - \frac{2}{X}\tan^{-1}Y + 2\sqrt{1+\frac{1}{Y^2}}\tan^{-1}\frac{X}{\sqrt{Y^2+1}} + 2\sqrt{1+\frac{1}{X^2}}\tan^{-1}\frac{Y}{\sqrt{X^2+1}}\right\} \qquad \ldots [11]$$

Keene and Luke (5) in their evaluation of the surface efficiency of different arrangements of heating elements in electric furnaces included the case of parallel strips (see Fig. 7 for cross-section through strips). This case may be treated by the application of Equation [11]. One of the two dimensions is infinite. When we let $Y = \infty$, Equation [11] reduces to

$$q = \sigma(T_1^4 - T_2^4)A\left[\sqrt{1 + \frac{1}{X^2}} - \frac{1}{X}\right] \ldots\ldots [12]$$

in which X is the ratio of strip width to distance between strips and q is the interchange between adjacent strips. The bracketed term represents the fraction of the radiation from one strip intercepted by the next. The fraction not intercepted is the portion in which we are interested. The heat interchange between the strips and their surroundings is then

$$q = \sigma(T_1^4 - T_2^4)A\left[1 - \sqrt{1 + 1/X^2} + 1/X\right] \ldots [13]$$

and the bracketed term is the desired surface efficiency of the strips. It is plotted in Fig. 7 (solid line) against the ratio X. The dotted line represents the results given by Keene and Luke in the paper referred to, bases on their assumption that the fraction of the total radiation from a surface element on one strip which is intercepted by an adjacent strip is equal simply to the plane angle subtended by the strip at the surface element divided by the total plane angle above the element. The extent to which this simplifying assumption is unjustifiable is indicated by a comparison of the two curves of Fig. 7.

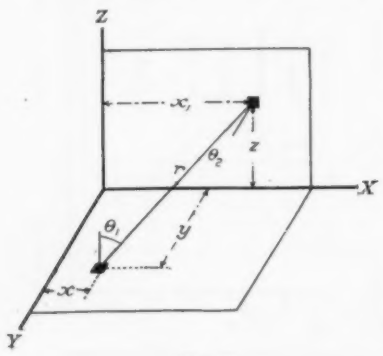


FIG. 8

In the evaluation by Keller of losses through furnace openings already referred to (6), a determination of the direct interchange between rectangles was made by an approximation method similar to that used by him for circular disks, and the conclusion was reached that the factor of interchange is the same, within 1 per cent, as that for disks, when the ratio of side of squares to distance separating them is the same as for diameter of disks to distance separating them. An exact comparison of disks and rectangles by the use of Equations [9] and [11] leads to the conclusion that when the ratio of side or diameter to distance apart is 1, the factor of interchange for squares is 16 per cent greater than that for disks; and that when the ratio is 2, the factor for squares is 8 per cent greater.

Case 11. Rectangles in perpendicular planes, the rectangles having a common side. In the accompanying Fig. 8 the rectangles, between which the heat interchange is to be evaluated are in the XZ- and XY-planes, and of dimensions X by Z and X by Y. The coordinates of the surface element dA_1 in the XY-plane are x and y; those of the surface element dA_2 in the X-Z plane are x_1 and z. Equation [2c] is most readily applicable to this case. In that equation the accompanying Fig. 8 defines the terms as follows.

$$dA_1 = dx\,dy \qquad \cos\theta_1 = z/r$$
$$dA_2 = dx_1\,dz \qquad \cos\theta_2 = y/r$$
$$r^2 = (x_1 - x)^2 + y^2 + z^2$$

Substituting, we have

$$q = \sigma(T_1^4 - T_2^4)\frac{1}{\pi}\int_0^Y\int_0^X\int_0^X\int_0^Z yz\,dz\,dx_1\,dx\,dy \ldots [14]$$

The integration, performed with respect to z, x_1, x, and y in order, leads to a solution which is slightly simplified by redefining Y and Z to refer to Y/X and Z/X, respectively, of Fig. 8. We then have

$$q = \sigma(T_1^4 - T_2^4)A_{xy}\frac{1}{\pi} \times$$

$$\left\{\frac{1}{4}\ln\frac{(1 + Y^2 + Z^2)^Y - \frac{1}{Y} + \frac{Z^2}{Y}\,(Y^2)^Y(Z^2)^{\frac{Z^2}{Y}}}{(1 + Y^2)^Y - \frac{1}{Y}\,(1 + Z^2)^{\frac{Z^2}{Y}} - \frac{1}{Y}\,(Y^2 + Z^2)^Y + \frac{Z^2}{Y}}\right.$$

$$\left. + \tan^{-1}\frac{1}{Y} + \frac{Z}{Y}\tan^{-1}\frac{1}{Z} - \sqrt{1 + \frac{Z^2}{Y^2}}\,\tan^{-1}\frac{1}{\sqrt{Y^2 + Z^2}}\right\} \ldots [15]$$

Case 12 (General). Interchange between any two surfaces of however irregular extent. If one of the two surfaces is quite small compared to the other, the mechanical device described under case 6 may be used, with its pivot point at the center of the small area. One would thus obtain a factor F_A to use in the equation

$$q = \sigma(T_1^4 - T_2^4)\,A_1F_A \ldots\ldots [16]$$

in which A_1 is the smaller area. The extension of the method to the case in which A_1 is not small compared to the mean distance to A_2 consists simply of dividing A_1 into a number of equal areas and of determining the factor F_A with the pivot point of the mechanical device placed successively at the centers of the different areas. The arithmetical mean F_A is then used in Equation [16]. The number of segments into which A_1 need be divided is usually small to obtain sufficient accuracy for engineering purposes.

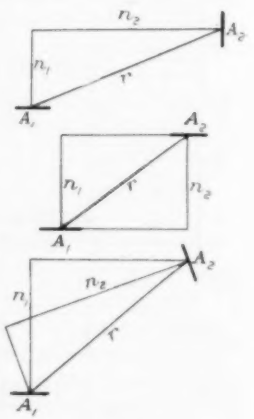


FIG. 9

An approximation method applicable to interchange between surfaces small relative to their distance apart and in such relative positions that a plane normal to one surface through its center passes through the center of the other surface has been presented by Schack (9). It follows directly from Equation [2c] and is

$$q = \sigma(T_1^4 - T_2^4)\,\frac{n_1\,n_2A_1\,A_2}{r^4}$$

in which n_1 and n_2 are defined in the sketches of Fig. 9. The method leads to considerable error when either A_1 or A_2 is not small compared to r.

EFFECT OF EMISSIVITY OF SURFACES ON RADIANT-HEAT INTERCHANGE

At the beginning of the preceding discussion of the effect of the shape of a system on radiant heat transmission in it, the assumption was made that all beams intercepted by a surface are absorbed by it. We shall now consider the effect of incomplete absorption on the equations presented.

Cases 1–6, Inclusive. In all these cases, one surface is very small compared to the distance to the other. Surface ΔA_1 will emit p_1 times as much as a black body, and surface A_2 will absorb the fraction p_2 of what ΔA_1 emits (absorptivity and emissivity are equal). The reflected portion $(1 - p_2)$ will be reflected in all directions from A_2, and since ΔA_1 is small, a negligible fraction of the reflected beam will undergo a second reflection from

ΔA_1 back to A_2. The factor p_1p_2 therefore enters directly into all the equations for cases 1 to 6, Equations [3] to [7].

Case 7—Infinite Parallel Planes. A beam leaving A_1 undergoes an infinite number of partial absorptions alternately at A_2 and A_1. The summation of the alternate members of the infinite series so formed leads to the familiar conclusion that the factor to allow for emissivity of the two surfaces is $\dfrac{1}{1/p_1 + 1/p_2 - 1}$. As long as p_1 and p_2 are not far from unity, this factor is approximately equal to p_1p_2.

Case 8—Enclosed and Enclosing Surface. Let us first consider the symmetrical cases of concentric spheres, or concentric infinite cylinders. Because the heat-transfer equations given are based on the area of the enclosed body of area A_1, we follow the history of a beam starting at A_1. If the reflection is specular,[3] every beam leaving A_1, in no matter what direction, will impinge on A_2 and A_1 alternately until completely absorbed; and the summation of terms representing absorption at surface A_2 leads to the same result as for infinite parallel planes; i.e., Equation [8] should be multiplied by $1/(1/p_1 + 1/p_2 - 1)$. This is an unexpected result and is due to our assumption of specular reflection of the unabsorbed portion of the radiation. An assumption much nearer the truth is that the reflection is completely diffuse instead of completely specular. Again we start with a beam leaving A_1; the fraction p_2 is absorbed by A_2, and $1 - p_2$ is reflected. Only a portion of the reflected beam from A_2, however, is intercepted by A_1, the other portion going directly to other parts of A_2. The calculation of the portion intercepted (not all absorbed) by A_1 may be made by proper application of the formula under case 2 for spheres, or case 5 for cylinders. In each case we find that the fraction of reflected radiation from a spot on A_2 which is intercepted by A_1 is A_1/A_2. By following the beam back and forth we form a family of infinite series, much more complicated than for the case of infinite parallel planes. Summation of terms representing absorption at A_2 leads to a fairly simple factor F_E by which [8] should be multiplied to make allowance for the emissivities of the surfaces. It is

$$F_E = \frac{1}{\dfrac{1}{p_1} + \dfrac{A_1}{A_2}\left(\dfrac{1}{p_2} - 1\right)}$$

Nusselt has given this result, but with no indication of his method of obtaining it, or of the assumption involved in deriving it. Inasmuch as actual reflection is neither completely diffuse nor completely specular, the true factor of multiplication of Equation [8] will be an intermediate value, such that

$$\frac{1}{\dfrac{1}{p_1} + \dfrac{1}{p_2} - 1} < F_E < \frac{1}{\dfrac{1}{p_1} + \dfrac{A_1}{A_2}\left(\dfrac{1}{p_2} - 1\right)}$$

The factor will be much nearer the right- than the left-hand expression.

It will be noted that the expression for F_E in terms of A_1, A_2, p_1, and p_2 reduces simply to p_1 when A_1/A_2 approaches zero, i.e., for the case of a small body enclosed in a large one.

Cases 9-12—Pairs of Surfaces Not Forming an Enclosure. Inasmuch as the whole system is not completely specified when we are considering heat interchange between two bounded surfaces, A_1 and A_2, we are in no position to consider what happens to those radiant beams from one surface not directed toward the other. They may undergo partial reflection at the unspecified surfaces and return to A_1 or A_2 for further absorption. We can only say, therefore, that the factor p_1p_2 introduced into the equations covering cases 9 to 12, inclusive, will give a result not far wrong, but always more or less low. An exact consideration of the emissivity effect in a system of many surfaces of different temperatures and emissivities would be extremely complicated.

Inasmuch as Lambert's cosine principle, which was assumed valid in deriving the equations of Section I, is no longer exact when the heat-interchanging surfaces are not black bodies, the discussion of the effect of the emissivity of surfaces on heat transmission may properly include a consideration of errors introduced by assuming the cosine principle valid. Probably the best work on the subject is that of Schmidt (7), who found that the deviation from the cosine principle was large for metallic surfaces, and that the average intensity was about $^4/_5$ the intensity measured normal to the surface. The error in using the principle is not serious, however, for oxide and other nonmetallic surfaces, and for rough surfaces. The modification of the equations here given, to make allowance for the dependence of intensity on angle of emission of radiation, must wait until we have more detailed knowledge of the departure of industrial surfaces from the cosine principle.

The engineer's task of applying many of the equations given in this paper may be greatly simplified by making use of the graphs presented in the summary, to which reference has been made (11).

[3] The angle of incidence equals the angle of reflection.

BIBLIOGRAPHY

1 R. A. Herman, "Treatise on Geometrical Optics," Camb. Univ. Press (1900).
2 H. B. Keene, Proc. Roy. Soc., vol. 88A (1913), p. 60.
3 P. D. Foote, B. of S.Sc. Pap. No. 263 (1916).
4 W. Nusselt, *Gesundheits-Ing.*, vol. 41 (1918), p. 171.
5 A. D. Keene and G. E. Luke, *Jl. A. I. Elec. Eng.* (1926), pp. 222–226.
6 J. D. Keller, *Fuels and Furnaces*, vol. 5 (1927), pp. 1591–1598.
7 E. Schmidt, *Beiheft zum Ges. Ing.*, vol. I, 20 (1927), Verlag R. Oldenbourg, Munich.
8 W. Nusselt, *Z.V.D.I.*, vol. 72 (1928), p. 673.
9 A. Schack, "Industrielle Wärmeübergang," (1929), Verlag Stahleisen m.b.H.
10 O. Siebert, Forsch.-Arb. no. 324 (1930), V.D.I.-Verlag.
11 H. C. Hottel, 2nd World Power Conf., vol. 18, sect. 32, no. 243 (1930), and *Mech. Eng.*, vol. 52 (1930), p. 699.

Discussion

W. J. KING.[4] The paper is a valuable contribution to the theory and practice of heat transmission. It will be welcomed by designers of the many types of apparatus in which radiant heat is a factor as providing a clear qualitative statement of the general problem, as well as giving a number of very useful and convenient formulas.

It has been the writer's experience that much confusion exists in the minds of many engineers with regard to this subject of radiant heat transmission. For example, the present paper is limited to "non-absorbing media." Of course it is generally recognized that a vacuum is non-absorbing, and for most practical purposes so is ordinary air, but what about the passage of radiation through the earth's atmosphere to the cosmic space beyond? This is an important question in connection with the cooling of outdoor apparatus exposed to the sky. If the temperature of astronomical space is somewhere near absolute zero, the radiation from a surface exposed to the sky on a clear night should be sufficient to cool it far below the ambient. As a matter of fact most terrestrial radiations are completely absorbed within a few miles of atmospheric air, so that the effective temperature T_2 in Equation [3] is never much lower than about −45 deg.

[4] Engineering General Department, General Electric Company, Schenectady, N. Y.

fahr., or 415 deg. absolute.[5] In most practical cases this temperature may be taken as that of the ambient.

Carbon dioxide and water vapor absorb considerable amounts of radiant heat. An instance of this recently came to the writer's notice, in connection with some tests on a large electric furnace for annealing copper wire. The heat was transmitted to the charge mainly by radiation from resistance strip heaters. To reduce oxidation, the air in the furnace was displaced by steam, whereupon the temperature drop from the heaters to the charge was greatly increased.

There seems to be a tendency among engineers in general to underestimate the relative importance of radiation as compared with other forms of heat transmission. Some time ago the writer was asked to design the heat-transfer features of a large sulphur-dioxide converter, in which the hot gases within a 10-ft. steel cylinder were to be cooled by a current of cool gas in the annular space between the catalyst cylinder and the outer shell. The instructions were to "neglect radiation." The temperature of the inner cylinder was about 250 deg. cent., and the amount of surface required to dissipate the heat by convection alone made the apparatus too large. It was finally found, however, that the radiation through the gas to the outer jacket amounted to about 20 per cent of the total heat transfer, which was sufficient to bring the machine down to the desired dimensions.

It might be thought that in an airplane traveling at 100 miles per hour the radiation from the engine cylinders would be negligible compared with the forced convection. However, it was noticed recently that in a certain plane the indicated engine temperature rose continually as the black paint peeled off of the cylinders, exposing the bright metal. It was thought that the low emissivity of the bright metal might account for this. Calculation showed that for a constant amount of heat dissipated, the temperature should rise from 450 to 510 deg. fahr. when the paint all came off. This was confirmed by the fact that the temperature at once dropped back to normal when the cylinders were repainted.

Even at lower temperatures, radiation may be an important factor, particularly in still air. For example, the dissipation from a plain self-cooled transformer tank is about 55 per cent radiation and 45 per cent convection.

Referring to the formula for F_E on the seventh page, the author says, "Nusselt has given this result, but with no indication of his method of obtaining it or of the assumption involved in deriving it." Apparently he has overlooked the excellent treatment of this problem by A. D. Moore in the chapter on Heat Radiation in his book on "Fundamentals of Electrical Design" (McGraw-Hill, 1927). Professor Moore also presented a paper on "Dissipation of Heat by Radiation" at the Great Lakes District Meeting of the A.I.E.E. in Chicago, in December, 1929. This included an interesting study of radiation from a slot or from a corrugated or finned surface. In all such cases the effective emissivity of the enveloping surface is higher than that of a smooth surface of the same material.

Some of the results of the present paper were given last year by O. A. Saunders in his "Notes on Some Radiation Heat Transfer Formulas," published in Vol. 41 of the Proceedings of the Physical Society (London). The case of non-concentric spheres also was treated. Some interesting conclusions are drawn from the formulas. For example, it is pointed out that "whereas coating the surface of a relatively small hot body with aluminum paint reduces the radiation heat transfer from the body to its surroundings in direct proportion to the change of surface emissivity, a similar treatment of the walls of the room in which the body is situated has no effect upon the heat loss."

A few special cases also are discussed by ten Bosch in his text, "Die Wärmeübertragung," published by Julius Springer, Berlin, 1927.

N. C. Artsay.[6] The paper is most timely, and it gives a clear exposé of the principles so vaguely understood by the vast majority of engineers. The treatment of the problem is especially valuable for the design of gas-fired furnaces where the materials are not exposed to the direct radiation of flames. The presentation of this paper seems to be an appropriate moment to discuss certain ideas on radiation. The phenomena of heat release during combustion are so complicated and, to our shame, we must say, so poorly known, that a general overhauling of our conceptions of what happens in a boiler furnace where fuel is burning is necessary. The fascination of the Stefan and Boltzmann fourth-power law begins to fade, especially when we realize that the source of radiant energy is the chemical process itself and not the high temperature of the product. The factor controlling heat exchange by radiation in cases of combustion in cavities is not on the emission side but on the absorption side. This apparent heresy may be illustrated by the following example.

Let us light a bonfire and keep it going at a steady rate in a wide open space on a clear cold night. This will be a sufficiently "black" and cold envelope for the flame. In such conditions the bonfire should relate at least 80 per cent of the total sensible heat liberated. Now let us screen the fire from all sides and gradually move the walls converging them toward the center of the fire. It is obvious that the rate of radiation emission would not decrease; it might even increase if we still entertain the idea that emission during combustion depends upon flame temperature. The temperature of gases leaving the bonfire will rise due to lowering the rate of radiation absorption. Let us cool the screens by evaporating water in them at constant temperature; the second term of the Stephan and Boltzmann formula will increase slightly due to higher metal temperature, but by no means to such an extent that the reduction of radiation absorption during the convergent approach of the walls could be explained by its new values. There is the increased flame temperature, and, if we believe that radiation emission from burning fuel depends upon the temperature then the emission and hence the absorption also should increase. All our experience with boilers tells us a different story, and the application of any formula for radiation emission during combustion is futile.

The laboratory experiments on the reflectivity of metal surfaces are usually conducted with radiation intensities far below those obtained in furnaces and mostly on wave lengths in the visible part of the spectrum. The International Critical Tables gives data on reflectivity of steel which for the infra-red range show a substantial reduction. But this is not all; determinations of reflectivity at low intensities of incidence are like nozzle tests with insufficient fluid to fill the orifice. It is high time that proper attention should be called to the absorption; under high-intensity incidence of radiation the absorbing surfaces perform as atomic mechanisms of variable efficiency, and hence a constant factor of "blackness" or "grayness" in such conditions is nonsense.

In a short discussion it is impossible to give all the consideration to support the new viewpoint, but the idea can be defined in the following simple terms:

(1) During the process of combustion the heat of chemical reaction is liberated to a great extent as radiation, and only a

[5] See "Heat Transmission," by Margaret Fishenden and A. F. Dufton, Special Report No. 11, (British) Department of Scientific and Industrial Research, H. M. Stationery Office, London, 1929.

[6] Assistant to Geo. A. Orrok, Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

small fraction, roughly 20 per cent, of it remains with the initial products of combustion.

(2) Depending upon surroundings, a part of the radiant heat will return in the product of combustion via unburned fuel, CO_2, CO, H_2O, to be further diffused into N_2.

(3) The total radiant heat absorption by the furnace envelope is fixed by the rate of heat liberation times the efficiency of absorption, the latter being a function of the load or total radiant heat per square unit of surface per unit of time.

Broido and Orrok in their empirical formulations of radiation absorption in boiler furnaces were closer to the true aspect than the majority of authors who tried to apply the fourth power law to the indefinable flame or furnace temperatures or surfaces. As Wohlenberg points out in his recent work, the radiation equilibrium of an oxidizing molecule is reached earlier than the dynamical equilibrium with the surrounding gas, and with this kept in mind we should regard the molecular chemical processes of oxidation as a source of radiant energy at a temperature (if such is needed) which is constant and dependent only upon the kind of combining molecules.

So much for the case of combustion; but there are many industrial furnaces and ovens where the treated materials are not subject to combustion radiation, and for these cases the author's work will be invaluable.

NOTE: Further discussion of this paper, together with the paper of William L. DeBaufre, by Ollison Craig, will be found following the DeBaufre paper.

AUTHOR'S CLOSURE

Some of the discussion of the paper would seem to indicate a mistaken idea of its object, which was to collect the old and to present some new material on one phase of the broad subject of radiant heat transmission. No pretense is made that the results are applicable, without modification, to problems in which the phenomena attending combustion are superimposed on the simple radiation from one surface to another. The complete treatment of radiant heat transfer under such conditions involves the use of other material not the subject matter of the present paper. The writer is in agreement with Mr. Craig that the complete analytical treatment of such a problem as the calculation of total heat absorption in a given boiler furnace would be inordinately tedious. It should be remembered, however, that although the Fuels and Steam Power Divisions are accustomed to think "boiler furnace" whenever radiant heat transmission is mentioned, there are many other types of industrial furnaces in existence, some of which are by no means incapable of analytical treatment. The author's experience with calculations on steel reheating furnaces, oil-cracking coils, and tube stills has indicated that much can be learned about probable furnace performance by the application of principles such as those discussed here, augmented of course by the use of other material included in the subject of radiant heat transmission. Even in the case of boiler furnaces certain fundamental questions of local overheating, losses through openings, distribution of heat to tube rows, etc., may be answered by a consideration of the geometrical problem involved, although the overall problem of furnace problems is at present best treated empirically.

Mr. King calls attention to the fact that Case 8, that of one body completely enclosing another, has been treated by A. D. Moore ("Fundamentals of Electrical Design," McGraw-Hill, 1927), by O. A. Saunders (Proc. Phys. Soc., vol. 41, p. 569, 1929) and by M. ten Bosch ("Die Wärmeübertragung," published by Julius Springer, 1927), in addition to the author's own treatment and that of Nusselt, to which reference has already been made. Of these treatments that of Mr. Saunders, with which the author was not familiar, is to be preferred for clarity and completeness. Mr. Saunders, considering the case of nonconcentric spheres, reaches the interesting conclusion that the maximum heat interchange occurs when the bodies are concentric, but that the decrease below this maximum as the inner sphere is moved away from the center is not of sufficient magnitude to be of much practical importance.

Mr. King further calls attention to a paper of A. D. Moore (A.I.E.E., vol. 49, p. 22, 1930) on the study of radiation from a slot or from a finned surface, in which the results of a tedious stepwise numerical calculation were presented for the special case of a slot of infinite length, with depth equal to four times its width. The author and Mr. P. V. Keyser about a year ago considered the general problem of finned-surface radiation and obtained an algebraic solution applicable to any ratio of slot depth to width and to any emissivity of the surface. The results have not been published, pending a solution of an allied problem.

Reference is made by Mr. King to the excessively high value obtained for the necessary heat-transfer area of a sulphur-dioxide converter, due to the fact that the radiation from hot sulphur dioxide was neglected. Mr. S. A. Guerrieri and the author, making use of available infra-red spectroscopic data on sulphur dioxide, have recently calculated a sulphur-dioxide radiation plot similar to those published for water vapor and carbon dioxide; and it should be possible with such a chart to make allowance for SO_2 radiation in the design of such equipment as the coolers following sulphur burners, etc.

The author cannot agree with Mr. Artsay's statement that "the factor controlling heat exchange by radiation in cases of combustion is not on the emission side, but on the absorption side," nor with his statement that "the bonfire should radiate at least 80 per cent of the total sensible heat liberated," if he means net heat of combustion by the term, "total sensible heat liberated," as the succeeding text would seem to indicate that he means. All the experimental evidence with which the author is familiar indicates that the fraction of heat radiated will be far less than 80 per cent; less, in fact, than half that figure. Experimental evidence for Mr. Artsay's unusual picture of the absorptivity of surfaces being dependent on the intensity of the radiation striking them would be welcome. Certainly there is a possibility of some such effect, but experimental evidence so far tends to deny its practical importance.

Operation of the Holland Station[1]

By E. M. GILBERT,[2] READING, PA.

The Holland Station, being among the first plants designed exclusively for high-pressure steam, is now in operation, and the experiences obtained during operation will be of interest. The main features of the plant are a 55,000-kw. cross-compound turbo-generator unit designed for variable-load operation; two similar boilers, each with its reheater built integral, designed for generating 250,000 lb. of steam per hr. each; elimination of the wall between turbine and boiler room, and control centralized on the operating floor. The operating experiences with the different elements of the plant are given, together with data showing the operating results for the six-month period. Operating results, while of short duration, show justification of the use of higher pressures from an economic standpoint.



THE Holland Station of the New Jersey Power & Light Company, a subsidiary of the Associated Gas & Electric Company, was designed and constructed by W. S. Barstow & Co., Inc. It is located on the Delaware River in New Jersey, about 13 miles below Easton, Pa. A description of the design features of the plant appeared in the February, 1929, issue of the *General Electric Review* and the March 1, 1929, issue of *Power Plant Engineering*.

The ultimate capacity of the Holland plant will be approximately 220,000 kw., the first unit of 55,000 kw. capacity having been put into operation the latter part of February, 1930.

The turbo-unit consists of a high- and a low-pressure cross-compound turbine, the low-pressure section working at variable pressure. The high-pressure turbine has a multi-valve arrangement in order to obtain high efficiency over a large load range. It receives steam of 1250 lb. gage, 750 deg. fahr., and exhausts through steam and gas reheaters at a pressure varying from 400 lb. gage to 85 lb. gage into the low-pressure turbine, the steam being reheated to 750 deg. fahr. before entering the low-pressure turbine. The high-pressure turbine operates at a speed of 3600 r.p.m. and the low-pressure at 1800 r.p.m., their maximum capacities being 11,800 kw. and 43,200 kw., respectively.

The condenser is of single-pass type having a surface of 31,600 sq. ft. The tubes are rolled at both ends, and rubber expansion joints are provided between water boxes and the shell to allow for longitudinal expansion, as the shell is hung from the turbine and the water boxes are supported from the basement floor, one box being bolted rigidly down and the other box being free to slide longitudinally.

The two boiler units, each designed for 250,000 lb. of steam per hour, are of the cross-drum type, 45 sections wide and 8 tubes high. The drums are made from solid forged billets and are 52 in. inside diameter, 4 in. thick, and 46 ft. long.

The furnaces are water-cooled except around the circular burners where refractories are used. The side walls of both furnaces and the front and rear walls of No. 1 furnace are of fin-tube construction. On No. 2 boiler the front and rear walls are of the bare-tube type, backed up by refractories. The furnace volume is 18,000 cu. ft. and the B.t.u. release about 18,000 B.t.u. per cu. ft. maximum.

Arrangement and Control

The general arrangement of equipment is shown on the plan (Fig. 1) and the cross-section (Fig. 2). All equipment operating continuously is controlled from a central station (with two turbo-units installed) located on the turbine floor, elevation 154.0, as shown in Fig. 1. The omission of the wall between the boiler and turbine rooms enables the station operator to see and control all principal equipment. With the exception of the coal-handling and milling machinery all apparatus is under the control of the station operator and can be governed automatically or by push-button devices. All equipment is regulated to maintain a constant steam pressure at the inlet of the high-pressure turbine. The water level in the boilers is automatically controlled by Smoot regulators, and feed-line pressure is regulated by combination Bailey pressure and electric control on the primary boiler-feed pump. A miniature board located at the centralized station controls all high-voltage outdoor switching equipment.

A photograph of the turbine room is shown in Fig. 3, the high-pressure unit being on the left and the low-pressure machine on the right. A photograph of the boiler room is shown in Fig. 4.

Operating Personnel

The control and operation of the plant on one floor without any wall between turbine and boiler room has made the Holland plant simpler to operate than existing low-pressure plants, and it has been found that no added personnel or higher class of labor than used in low-pressure plants is needed. The operators have all been drawn from existing plants of the Associated Gas & Electric Company's system and had no experience with plants of pressures higher than 200 to 400 lb. prior to their connection with this plant.

---

[1] The station was renamed "Gilbert" since the paper was written.

[2] President, W. S. Barstow & Co., Inc. Mem. A.S.M.E. Mr. Gilbert was born in Wilton, Conn., and after receiving his technical training in Cornell University he entered the employ of the American Stoker Company, Dayton, Ohio. In 1899 he was sent to London to install stokers in England and France, remaining abroad until 1907 as general superintendent of a parent organization that sponsored the sale and manufacture of stokers in Europe. After his return to the United States he became chief engineer for the Leadville (Colo.) Light & Power Company, and from 1909 to 1914 he was active in the operation and construction of steam- and hydroelectric-generating plants for the Colorado Power Company. He was also connected with the construction of the Boulder and Shoshone dams and power houses in Colorado. He sponsored the development of oxyacetylene welding for a steel-pipe line over 8000 ft. long used in connection with the Boulder dam. In 1916 he became general manager of the Metropolitan Edison Company and chief engineer of the properties of the General Gas & Electric Corporation. When the Associated Gas & Electric System acquired these properties, he was a director of the Metropolitan and vice-president and director of four of its subsidiaries. At present Mr. Gilbert is President of W. S. Barstow & Co., Inc., and the Utilities Purchasing & Supply Corporation. He is also director and an executive officer in several of the operating companies in the Associated Gas & Electric System.

Contributed by the Power Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

TABLE 1

Week ending	Gross generated	Net output	Maximum hour	Plant service hours	Station load factor	Ratio of station use to net output	Net coal rate	B.t.u. per net kw-hr.	B.t.u. coal	Thermal efficiency
Mar. 1	203,000	122,610	7,000	56	2.20	..	1.127*	15,553	13,800	21.94
8	42,000	9,990	5,000	13	.45	..	1.127*	15,553	13,800	21.94
15	186,000	127,700	11,000	25	2.01	..	1.127*	15,553	13,800	21.94
22	1,833,000	1,650,680	27,000	93	19.84	11.05	1.127*	15,553	13,800	21.94
29	2,920,000	2,660,070	28,000	126	31.60	9.77	1.127*	15,553	13,800	21.94
Apr. 5	1,510,000	1,342,520	23,000	70½	16.34	12.48	0.949	13,091	13,800	26.06
12	2,351,000	2,119,460	23,000	113	25.44	10.92	0.933	12,872	13,800	26.51
19	2,782,000	2,540,400	35,000	119½	30.11	9.51	0.928	13,008	14,024	26.23
26	3,819,000	3,520,800	51,000	137½	41.33	8.47	0.901	12,637	14,024	27.00
May 3	5,095,000	4,763,890	56,000	138	55.14	6.95	0.890	12,476	14,024	27.35
10	3,710,000	3,396,630	30,000	145½	40.15	9.23	0.880	12,344	14,024	27.64
17	3,118,000	2,862,270	26,000	125	33.74	8.93	0.987	13,844	14,024	24.65
24	3,308,000	3,040,750	30,000	148	35.80	8.79	0.928	13,014	14,024	26.22
31	3,926,000	3,581,940	30,000	168	42.49	9.61	0.952	13,345	14,024	25.57
June 7	4,187,000	3,823,540	33,000	168	45.31	9.51	0.959	13,454	14,024	25.36
14	4,360,000	3,952,000	34,000	168	47.19	10.32	0.960	13,463	14,024	25.34
21	3,946,000	3,591,210	32,000	155	42.71	9.88	0.919	12,894	14,024	26.46
28 / July 5	277,000	32,480	Out of service for repair on turbines, condenser, reheaters, and economizers							
July 12	4,223,000	3,872,530	33,000	143	45.70	9.05	0.907	12,716	14,024	26.83
19	4,899,000	4,519,360	42,000	149	53.01	8.41	0.983	13,786	14,024	24.75
26	6,319,000	5,878,270	53,000	144	68.39	7.50	0.846	11,866	14,024	28.75
Aug. 2	6,337,000	5,904,260	54,000	126	68.58	8.17	0.902	12,652	14,024	26.97
9	7,344,000	6,853,870	56,000	149	63.35	7.15	0.913	12,798	14,024	26.66
16	8,030,000	7,467,490	54,000	168	86.90	7.53	0.918	12,977	14,139	26.29
23	7,342,000	6,783,220	54,000	168	79.46	8.24	0.878	12,420	14,139	27.47
30	3,429,000	3,068,620	37,000	144	37.11	11.74	0.961	13,594	14,139	25.10

* Estimated.

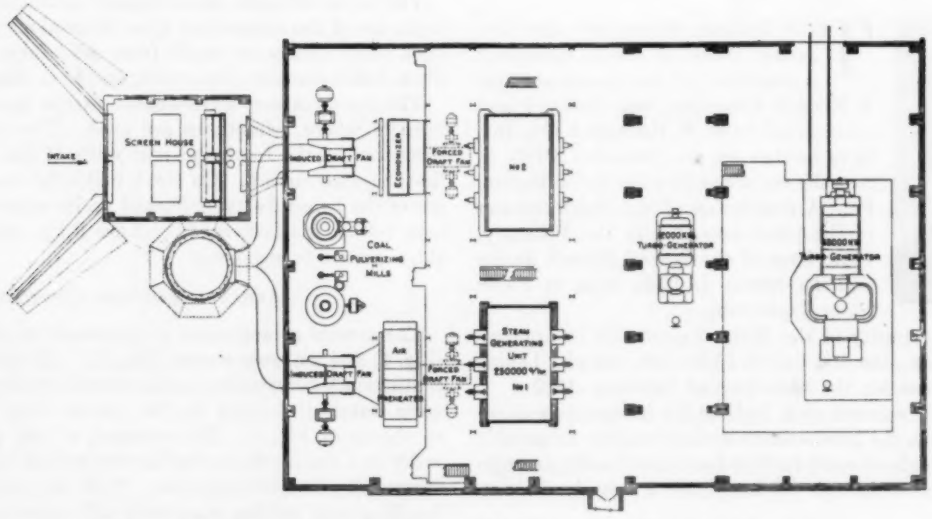


FIG. 1 GENERAL ARRANGEMENT OF EQUIPMENT AT HOLLAND STATION

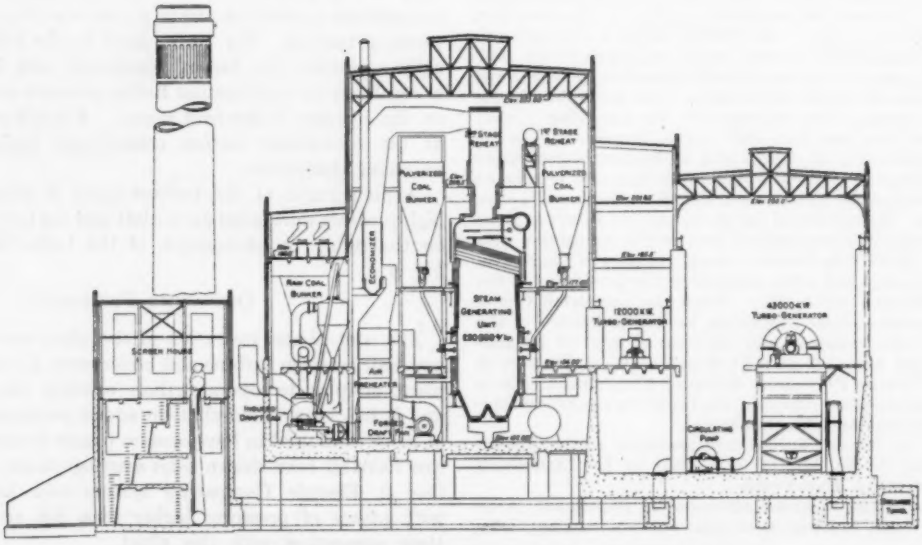


FIG. 2 CROSS-SECTION OF EQUIPMENT AT HOLLAND STATION

TABLE 2 TYPICAL AVERAGE OPERATING DATA

Date and time	August 19, 1930, 2:30 p.m.	
Station load:		
Kilowatt gross	52,000	
Boiler No.	1	2
1000 lb. steam per hour	218	218
Steam pressure	1265	1265
Steam temperature out of superheater	770.0	754.7
Steam temperature in reheater	543.5	543.5
Steam temperature out of reheater	808.6	818.8
Steam pressure in reheater	415.0	415.0
Steam pressure out of reheater	410.0	410.0
Economizer:		
Gas temperature in economizer	935.0	937.2
Gas temperature out of economizer	550.0	557.6
Water temperature in economizer	363.0	365.0
Water temperature out of economizer	505.0	494.4
Air preheater:		
Gas temperature in preheater	550.0	557.6
Gas temperature out of preheater	312.0	308.3
Air temperature in preheater	85.0	85.0
Air temperature out of preheater (left)	410.0	405.0
Air temperature out of preheater (right)	420.0	412.6
Flue-gas analysis:		
CO_2 in economizer	13.13	13.53
Condenser:		
Barometer	29.86	....
Vacuum	28.20	....
Circulating-water temperature, in	73	....
Evaporator:		
Make-up water, lb. per hour	7200	
Make-up water temperature	70	

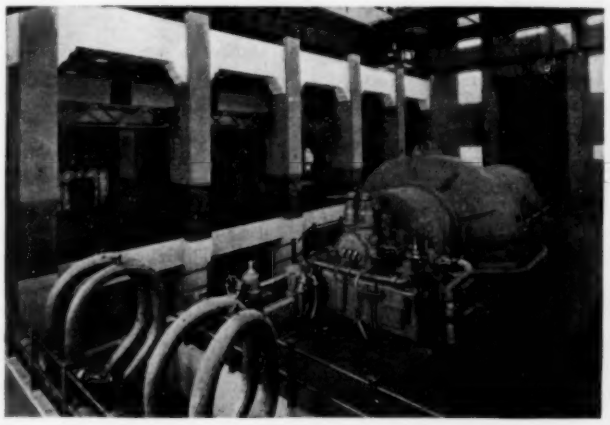


FIG. 3 TURBINE ROOM AT HOLLAND STATION

The following list of the personnel shows that the station can be operated satisfactorily with a minimum number of men:

1 superintendent
1 assistant superintendent
1 boiler foreman
1 maintenance foreman
1 chemist
1 instrument man
1 station clerk
4 switchboard operators
3 turbine operators
6 condenser and pump operators
3 firemen
3 mill operators
3 water tenders
13 maintenance men
5 coal handlers
2 janitors
6 laborers

55 total

OPERATING RESULTS

Table 1 shows operating results from February 23 to September 1. The coal consumed includes all fuel purchased for the drying out and used for all preliminary operation of the plant.

Table 2 shows average readings throughout the plant under date of August 19, 1930.

OPERATING EXPERIENCES

General. With a plant radically new in design and with many untried features it is evident that some difficulties would be encountered and would have to be rectified.

It is impossible at this time, due to the short period of operation, to report all operating problems as completely solved. Any operating difficulties stated below should not be considered as a reflection on the effort or service of the manufacturers connected with this station. Where difficulties have been encountered, the manufacturers have cooperated to the fullest extent, and if a complete remedy of any trouble cannot be given it is due to lack of time.

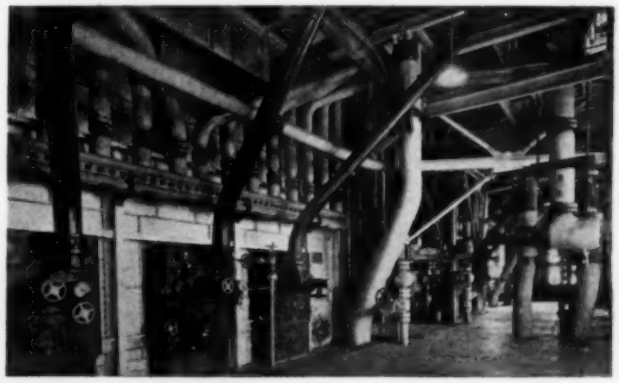


FIG. 4 BOILER ROOM OF HOLLAND STATION



FIG. 5 CENTRALIZED CONTROL

While the plant has not attained the normal expected output, sufficient information is available regarding operating costs and fixed charges to indicate the economical justification of the design and construction of the plant.

Piping. All piping joints are of the gasketed type—metal gaskets for high pressure and composition gaskets for low pressure. Flanges for the high-pressure joints were made 35 to 50 per cent stronger than the 1350-lb. standard, and special

threaded studs were used. All other flanges and boltings are of standard construction.

During the early period of operation composition gaskets used on the high-pressure lines in sizes of 2 in. and under gave trouble and were replaced by metal gaskets, which have been entirely satisfactory.

It was found that when the flanges on the high-pressure superheated lines were covered leaks developed during the change of temperature periods when a line was being brought up to

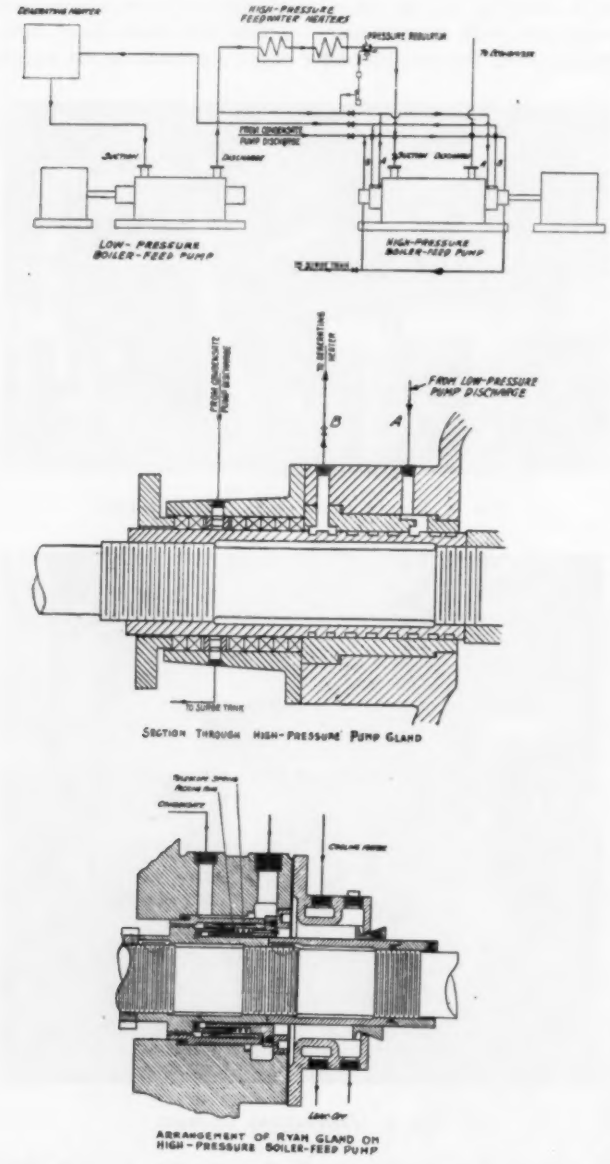


FIG. 6 HIGH-PRESSURE BOILER-FEED PUMP, ARRANGEMENT OF GLAND SEALS

pressure or the pressure was being lowered. The removal of the flange covering overcame this trouble.

Turbines. Minor trouble was experienced in the beginning on the high-pressure turbine due to condensed steam from the glands getting into the oil system. There has been considerable trouble due to sticking of the valve stems in their bushings and in the cam mechanism of the valve control, and alterations have to be made to obtain even changes of load. Leakage in the horizontal joints of the high-pressure turbine was stopped by tightening the holding-down bolts. On the low-pressure unit trouble developed with the vacuum tripper, which was due to faulty adjustment. No difficulties have been experienced with the intercepting valve, but changes are being made to give a more rigid construction.

Condensers. Leakage between the rubber expansion joint and the water box, causing the loss of the circulating system, has been overcome by a change made in the bolting arrangement. Some trouble was experienced with the reciprocating air pump during starting due to pulling water from the water seal between the first stage jet and the condenser into the cylinder, which caused cracking of the head. This has been corrected by the installation of a bypass around the first stage jets, which is used during starting up.

Boiler Feed. The secondary feed pumps were originally provided with Ryan seals. These were replaced by a combination labyrinth and stuffing box, which has proved very satisfactory. Some trouble was encountered with the oiling system, but was overcome by rearrangement of piping. A Mercon regulator is being installed between the primary and secondary feed pumps to insure bottling up of high-temperature water in the secondary pump during light load operation. This arrangement is shown in Fig. 6.

Fourth Stage Drip Pump. The plant has not operated regularly with No. 4 high-pressure heater in service. Ryan glands were originally installed on the drip pump, but these had to be replaced with labyrinth-type glands.

Superheater. The superheat obtained at the start was low, but this has been adjusted by the installation of a kicker baffle.

Gas Reheater. Reheat temperature was entirely too high during the first period of operation and required the use of a desuperheater. To remedy this trouble reheat surface has been removed from both boiler units.

Steam Reheater. Trouble has occurred in the steam reheater due to faulty welding of the high-pressure elements. This reheater has not been used regularly as the gas reheater has given more than enough temperature, and the use of the steam reheater has not been required.

Economizers. The first trouble on the economizers was due to leaky handhole caps, this having been remedied by using softer steel gaskets. Later considerable trouble developed in leaks between the tubes and the headers. Plans are in progress to correct the difficulty, but at this writing the final cure cannot be given. However, it is felt that this problem will be satisfactorily overcome.

Induced Draft Fan. At the starting period, difficulties were experienced, caused by erosion of the fan-inlet vanes, due to the high velocity when the vanes were almost entirely closed at light ratings. This trouble has been overcome by the installation of leak-in dampers which are used at light loads during starting.

Burners. With the original installation of the burners the flame went below the slag screen and changes were made to remedy this. Trouble has been experienced with the burning of the cone, and alterations are being made to overcome this difficulty. It has been impossible to operate continuously burning the mill vents in the boiler, due to the fact that the vent fan does not have sufficient excess pressure to operate satisfactorily with the high secondary air pressure required by the first design of the burners. It is expected that the changes in the burners will make the present vent fan satisfactory. If not, a vent fan of greater pressure must be installed.

Slag Trouble. In the beginning slag trouble was experienced due to faulty burner operation and the impossibility of removing

excess slag during operation where the slag screens cross. Changes in burner design and the installation of additional doors in the ashpit have overcome this trouble.

Emergency Boiler-Feed Pump. A turbine-driven emergency boiler-feed pump was installed; all other auxiliaries being motor-driven. This pump has functioned when trouble had developed with the main turbine, and its use has been felt to be justified. This emergency pump gave some trouble due to a piece of concrete being left in the suction, which threw it out of balance and caused the freezing of the runner and wearing ring.

Feedwater. The quality of the feedwater at the Holland Station was expected to be a very uncertain factor due to the fact that it is made up of water from the Delaware River, which is comparatively clean, and water from the Lehigh River, which flows into the Delaware at Easton. The pollution of the water is increased by water dumped from factories and sewerage systems between Easton and the plant. During the first months of operation the water did not cause any difficulties, but during the extreme dry season the amount of solids in the water increased very rapidly. The evaporator had to work at its full capacity, due to excessive leakage of the economizer, which also aggravated the feedwater difficulties and raised the concentration in the evaporator and boilers, causing solids to be carried over with the steam into the turbine, where they deposited and reduced the efficiency and capacity of the unit. It was decided to wash out the turbine with saturated steam, and this method has been considered satisfactory, although at this writing only the first washing has been accomplished and the final work has not been completed. Greater care is now being taken to keep down the concentration in the evaporator, and with this precaution it is expected that no further difficulties with the turbine being filled with scale will be experienced.

Discussion

SHEPPARD POWELL.[3] To evaluate adequately the various factors contributing to the success or failure of the operation of the boiler-feedwater system at this station, it will be necessary to review briefly the type of the existing system. The raw-water supply is obtained directly from the Delaware River a few miles below where the Lehigh River enters the Delaware. The Lehigh River is contaminated by mine drainage, and also receives considerable trade-waste pollution from the industries. Prior to the design of the feedwater system, a comprehensive study was made of the Delaware River as a source of supply in comparison with other available water sources. As a result of this investigation it was indicated that the Delaware River would be relatively satisfactory for the initial operation of the station provided sufficient pretreatment of the water was provided to insure a fairly good feed to the evaporator. The treatment recommended and installed consists of coagulation and subsidence, followed by filtration and final softening by the zeolite system. As a result of this treatment all water fed to the evaporator is filtered and softened. The pretreatment has undoubtedly prevented scale formation in the boilers, especially in the water walls, since without softening ahead of the evaporator the high carry-over from the evaporator would have resulted in some scale formation.

A critical study has been made of the boiler-feedwater conditions, in order to allocate the causes responsible for the carry-over from the boilers which resulted in turbine deposits. The investigation has not been completed at this time, but a study of the operating data indicates that the difficulty has been due in part to poor evaporator performance, resulting in heavy contamination of the evaporator condensate. The evaporator was guaranteed to deliver a condensate containing a maximum total solid content of 13 parts per million (0.75 grain per gallon), when the evaporator was operated at a net maximum of 15,000 lb. of distillate per hour. Actually the total solids determined have been as high as 78 parts per million (4.5 grains per gallon). It is quite probable that even a greater amount of solids was entrained at critical periods of operation, but was not recorded, since no recording devices to determine the purity of the condensate have yet been installed. In order to determine the relative extent of the contamination of the evaporator distillate, the actual concentration of salines in the evaporator shell as measured by analysis was compared with the theoretical saline concentration which should have occurred under normal operation of the equipment and under a given set of conditions. This study, based upon the actual operating conditions, indicated that about 10 lb. of solids per day, or more, were introduced into the boilers from the make-up evaporator. As a result of these conditions the concentrated boiler salines determined were as high as 2000 parts per million (117 grains per gallon). It has been clearly demonstrated that concentration of solids approaching these quantities cannot be carried without serious contamination of the steam leaving the boilers.

A number of factors have been responsible for the unsatisfactory evaporator performance, although the importance of each factor has not been allocated. The following conditions have probably contributed to failure to produce a good evaporator distillate at all times:

(1) Probably high water in the evaporator shell
(2) Rapid fluctuation in evaporator output, due to the necessity of hand regulation of the drips, which in turn was due to failure of the existing traps on the drip line
(3) Relatively high concentrations of the salines in the evaporator shell
(4) Possible clogging of the vanes in the vapor purifiers, and clogging of the drip line from the vapor-purifier section
(5) No baffle plate ahead of the vapor-purifier section.

Based upon the foregoing analysis of these operating problems a number of changes in the evaporator construction and operation are indicated. A summary of these proposed improvements is listed as follows:

(1) The installation of a baffle plate to prevent surging of water into the vapor-purifier elements, and the relocation of the drip from the vapor purifiers so as to discharge the water removed by the vanes outside of the shell
(2) Continuous blowing down of the concentrated salines in the evaporator shell, limiting the maximum total solids of the salines to 1200 parts per million, or lower if found necessary to produce a satisfactory distillate
(3) Operation of the evaporator at ratings sufficiently high so that it is not necessary to flood the coils; effective automatic operation of the drip trap will be provided as a means of accomplishing this end, since past operation of the trap has not been satisfactory
(4) The installation of recording conductivity apparatus to determine the purity of the distillate and the concentration of salines in the evaporator shell; this will insure a continuous record to indicate desired changes in the control of the evaporator, thus providing a distillate of high purity.

A condition of zero carry-over is not likely to be met in practice, so the following study of boiler concentrations has been based upon the assumption that the carry-over from the evaporator

[3] Baltimore, Md.

can be limited to 13 p.p.m. of total solids in the distillate (the evaporator guarantee). The further assumption has been made that these various solids are present in the evaporator distillate in the same proportion as exists between these various solids in the evaporator salines.

When the boilers are operating at full load (500,000 lb. of steam per hour), with 1.5 per cent evaporated make-up and no condenser leakage, there will be introduced into the boilers 2.34 lb. of solids per day as carry-over from the evaporator under these conditions. Since these solids are alkaline in character, their content of sodium sulphate is added to give the desired ratio, and the total introduction of solids per day, counting both the carry-over and the added sodium sulphate, would be 3.25 lb. Assuming a saline content of 60,000 lb. per boiler with no carry-over therefrom, the total solids in the boiler salines would be increased about 27 p.p.m. per day by the evaporator carry-over plus the added sodium sulphate. It can thus be seen that if the boilers be blown enough to decrease the concentration 100 p.p.m., about four days would elapse before they would have to be so blown again. This schedule of blowdown is in marked contrast to existing conditions where it is often necessary to blow the boilers every six hours. It can be seen that, even with the limit of 13 p.p.m. total solids in the evaporator distillate, satisfactory blowdown conditions of the boilers could be maintained, and that with purer distillate the boiler concentration would be even less of an operating detail. It should be remarked here that an evaporator distillate containing 13 p.p.m. total solids is unsatisfactory, and conditions should be established whereby a much purer distillate can be obtained.

G. C. Derry.[4] Erosion is a rather difficult problem to deal with and has become increasingly so the last few years. During this time the designers of power stations have been very materially increasing the capacities of their plants, and with the insertion of reheaters, economizers, air heaters, and cinder catchers, the draft loss has mounted steadily, so that it is now quite common to receive requests for draft loss varying from 12 in. up to 20 in. This draft loss necessitates a much higher speed of the fan than heretofore, accompanied by a more rapid velocity of the cinder-laden gases across the face of the fan blades. We have kept pace with the designers so far as strength of the material is concerned, although the stresses are becoming very high.

These high stresses require the use of heat-treated alloy steels. It was thought at one time that through the use of these steels the problem of erosion might be solved. However, we have in the course of the last 10 years experimented with practically all of the alloy steels and have been greatly disappointed to have been forced to the conclusion that none of the alloys at present on the market has any virtue whatever in resisting the erosion encountered in induced-draft service.

We have been agreeably surprised at a change in the character of the erosion accomplished by our vane-control, spiral-inlet, induced-draft fans. This design was worked out to secure an even distribution of the gases to the inlet of the fan, and we found when the fans were put in service that there was also an even distribution of the dust. The result is that such erosion as occurs on the wheel is distributed across the whole face of the blade and at the outer periphery, so that the resulting erosion of metal does not seriously impair the strength of the wheel.

In some cases the wear has been almost exactly uniform across the width of the blade, and in no case has there been the sharp file-like cutting that we used to run into with the old square inlet box. It is possible to operate these fans at high tip speeds with the delivery edge, or portions of the delivery edge, worn feather thin. Where the fans are operated at high speeds and low capacities with the adjustable inlet vanes very nearly closed, wear results on these inlet vanes. This wear can be minimized by the use of Stellite or by a leak-in damper, in some such manner as the author suggests.

Wear on these stationary inlet vanes is a much less serious matter than wear on the wheel. They are much less expensive to replace, and the danger of wheel failure during operation is avoided. To sum up the matter of the new type of fan briefly, it may be said that the wear has been transferred from the wheel to the inlet vanes, which we consider is a very desirable transfer.

We have noticed in many plants a tendency to weld material to the wheel blades at the first sign of erosion. This practice cannot be recommended. The duty required from this modern induced-draft fan requires the use of alloy steels, and alloy steels cannot be subjected to the acetylene torch or the electric arc without seriously impairing their structure. The best procedure is to operate the fan with the maximum allowable wear on the blade, and then to reblade. The plant operator is too apt to consider that the slightest bit of erosion requires immediate attention. Such is not the case. Where erosion is occurring only at the outer periphery of the wheel, the impairment of the strength of the wheel is very slight, even for a large degree of erosion. The inner portion being of the original thickness and strength, together with the reasonable allowance for safety made in the design of modern fans, eliminates entirely any necessity for rapid and quick repairs.

We have also found in the old type of rectangular inlet box that the way in which the gases come into these boxes made a great deal of difference in performance and fan guarantees. We have proved through tests that with these rectangular inlet boxes the performance can be changed as much as 50 per cent, depending on whether the gas comes in on one side or the other of the fan inlet box. This is one of the reasons that caused us to design our spiral-inlet boxes, and we have found that it



Fig. 7 View of the Gilbert Station

makes no difference which way the designer leads his air into this particular type of fan, for he can be assured of his expected performance from the beginning of the inlet box to the outlet flange of the fan.

[Note: Further discussion of this paper, together with the paper of K. M. Irwin, by H. J. Kerr and W. S. Johnston will be found following the Irwin paper.]

Author's Closure

It may be of interest to hear what troubles have been encountered in the first six or seven months of operation of this new

[4] Manager, Power Apparatus Department, B. F. Sturtevant Co., Inc., Hyde Park, Boston, Mass.

1250-lb. pressure steam plant. Some months of continuous operation have passed since the data originally submitted in the paper were printed, and some changes have taken place and some additional experience has been gained. A view of the plant is shown in Fig. 7.

Piping. All piping joints are of the gasket type, special metal gaskets being used on the high-pressure and composition on the low-pressure joints. All flanges for the high-pressure joints are made with from 35 to 50 per cent added thickness above the 1350-lb. standard, this being done to prevent distortion of the flanges. Threads on the studs are considerably finer than those used under the standard. The high-pressure joints have all been satisfactory, with the exception of two which have given first. These bushings were replaced with nitralloy, and present operation shows this to be a decided improvement, but it is too early to state whether this is a complete cure. Very poor regulation was obtained at first with loads from 28,000 to 34,000 kw. This trouble was thought to be due to the springing of the camshaft under load, all operating arms being keyed to a steel shaft. This arrangement was changed so as to apply the load directly to new steel cam arms, merely using the shaft as a bearing and thus eliminating any strain on it. This appears to be satisfactory to date.

Considerable trouble has been experienced with oil leakage from No. 1 bearing of the high-pressure turbine. The loss from this bearing has been as much as 100 gal. per day in its worst

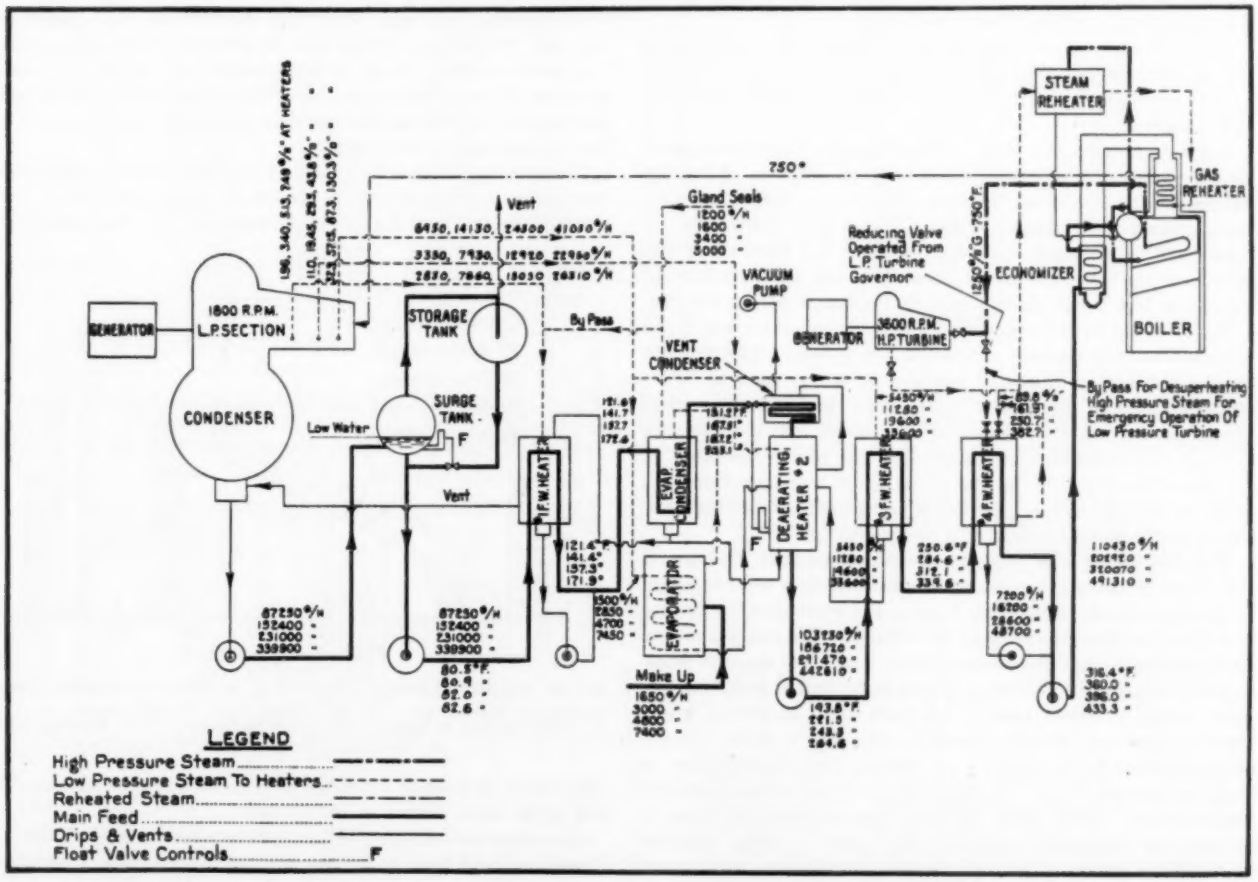


FIG. 8 GILBERT STATION HEAT-CYCLE DIAGRAM

trouble several times. It was decided that these two leaks were due to lack of flexibility in the piping, and added flexibility is being arranged for at these points.

On high-pressure piping less than 2 in. in diameter, where the joints are subjected alternately to high and low temperatures, leakage has been encountered even with the metal gaskets. Experience in the plant with small forged fittings, which are screwed on to the pipe and then welded, has been highly satisfactory, and where necessary on this smaller piping this method will be used.

On low-pressure joints composition gaskets are being used and standard flanges and bolting. Fig. 8 is a diagram of the heat cycle.

Turbine. Sticking of the valve stems in their cast-iron bushings on the high-pressure turbine occurred frequently at condition. A new bearing has just been installed, but results are not yet known.

A great deal of care must be exercised in starting the machine to prevent the bursting of the lead diaphragm on the turbine used in connection with the vacuum tripper. If too much non-condensible gas or air remains in the shell at the time of opening of the throttle, the pressure in the turbine may increase enough to burst the diaphragm. The removal of this air before starting is difficult to accomplish as the turbine gland seals are practically inoperative until the turbine is moving.

Trouble was experienced during the lightning season by the low-pressure turbine tripping off with a sudden large change of load. This was caused by the governor controlling the intercepting valve not acting quickly enough to prevent the steam contained in the piping, reheaters, etc., from entering the low-

pressure turbine and causing overspeeding. This governor was set to act more quickly.

It was noted that the capacity of the turbine, when operating under full steam flow, gradually dropped from 56,000 kw., which had been the high point of operation, to 46,000 kw. Investigation, by the comparison of pressures under the turbine shell, etc., led to the belief that the turbine blading was choking with foreign material. The turbine was not opened, so that the exact point of clogging was not determined. Washing of the turbine with saturated steam was resorted to in an endeavor to see whether this scale formation could be dissolved and removed this way. The result was satisfactory, and the capacity of the machine was gradually brought back. This scaling of the turbine has been traced to the evaporator, and changes in this piece of apparatus are being made.

Boiler Feed. As stated originally, there was considerable trouble encountered with Ryan seals, which were later replaced by labyrinth gland and stuffing box. This has greatly improved this condition.

Some time after starting, leakage in the boiler-feed pump casings developed, requiring that they be welded. After the welding, leaking occurred between diaphragms, this apparently being caused by a warped casing. These have since been rebored, and it is hoped that this condition has been corrected, as no similar trouble was experienced in the pumps where no distortion has taken place nor in these pumps before welding.

It was noted in the paper that a Mercon regulator was being installed between primary and secondary pumps. This has not been a satisfactory piece of equipment, so no data as to its operation can be given, but it has been determined by hand operation of a valve between the two pumps that maintenance of constant differential will accomplish the results desired.

Induced-Draft Fans. In the paper there was mentioned the wear on the inlet vanes. Stellite is being used in repairing these vanes, and it seems to materially increase their life.

The fan casing has been lined with removable steel plates which can be easily taken out and replaced as they wear through.

The floats on the fan, which wear at the outer tips, are having new metal welded into place as this occurs. The operators have been cautioned that the present floats, which are made of heat-treated steel, can be welded only to a very limited extent on the outer tips. It seems that a fan built from materials which prevent rapid and simple repairs is not what is desired in the average power plant, and we are asking the manufacturer to look into the question of building these wheels so as to simplify maintenance. Their great problem apparently is to effect a satisfactory compromise between the class of metal required to stand the strain of the high tip velocity and that which most satisfactorily resists erosion.

Burners. All experiments to reduce secondary air pressure to allow use of present vent fans have been without success, and new vent fans of higher pressure are now being installed. Some maintenance still remains on the burner cones, and there is some slag trouble in the ashpit and on the boiler tubes when the boilers are operating continuously at the higher ratings.

Evaporators. During the first months of operation a great amount of leakage occurred in the economizers, the result being that the make-up for the system increased very materially and the evaporator was working at unusual capacities. It was also found that the method of operating the evaporator was irregular, which still further increased its overload at times. Under normal conditions, however, the evaporator was not meeting its guarantees, too great a quantity of solids being carried over into the distilled water. Various changes were decided upon covering operating procedure, alterations of steam baffle, limit to steam supply lines, etc., and these were explained in the discussion by Mr. Sheppard Powell. As these changes are not all completed it is too early to state definitely whether the solids carried over will be reduced to the required point, but operation has already been materially improved.

Feedwater Treatment. The feedwater treatment which is being used was discussed by Mr. Powell, who is responsible for the layout.

As to heaters, reheaters, economizers, etc., nothing need be added to the text originally prepared. Such troubles as existed have been overcome except for leaking in the rolled joint of the economizer, and that is in the hands of the manufacturers.

No trouble with the boilers, water walls, or brickwork has been encountered; not a brick has been replaced in the furnaces and no boiler or water-wall leaks have occurred.

Conclusions. It may be safely stated that the operators feel that very little trouble can be ascribed to pressures and temperature used. Most of the troubles are similar to those occurring in any new plant, especially where newly developed machines or apparatus arrangements are being tried out, and a cure is apparent in each case.

The quality and quantity of labor required for plant operation has not been increased due to the use of higher pressures. The chart shown in Fig. 9 is a sort of yardstick used to compare plant

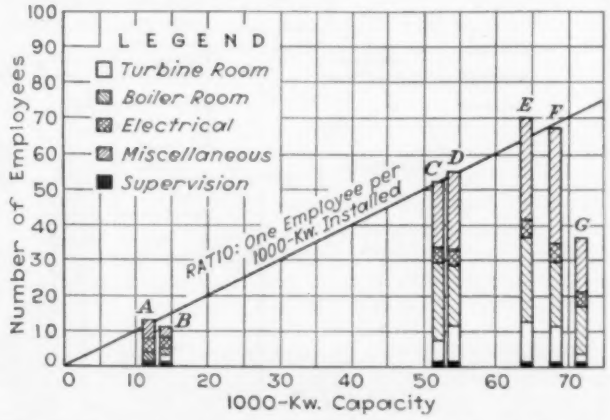


FIG. 9 CLASSIFICATION OF EMPLOYEES AT STEAM POWER STATIONS (*A*, Benson Springs, Fla.; *B*, Avon Park, Fla.; *C*, Binghamton, N. Y.; *D*, Gilbert Station at Holland, N. J.; *E*, Middletown, Pa.; *F*, Reading, Pa.; *G*, Parr, S. C., standby for hydro.)

labor costs, and while not considered as an accurate measure, it still gives some degree of comparison.

Operating statistics for the plant to date are shown in Table 3. These coal and heat quantities are based on shipper's weights and include all coal used for heating, testing, numerous shutdowns and restarting, and also without any benefit from No. 4 heater. The B.t.u. per kw-hr. will be reduced somewhat when all unusual troubles are removed and steady operation is possible.

It is interesting to note that this plant was built within the estimate, and cost about 7 per cent more than the 350-lb. Middletown plant.

Fig. 10 gives the relative costs between various stations which have lately been built by the Barstow Company, these costs being placed so as to compare the different stations under each construction heading. It will be noted that in several of the divisions the 1200-lb. plant was not by any means the most expensive.

Before installing the bin system at the Gilbert Station a careful analysis of costs for both bin and unit systems was made. The figures for these comparisons were furnished by the manufacturers, and the result was that there was practically no difference in investment cost of the two systems. This analysis

TABLE 3 OPERATING FIGURES FOR GILBERT STATION

Week ending		Gross generated	Net output	Maximum hours	Plant service hours	Station load factor	Ratio of station use to new output	Net coal rate	B.t.u. per net kw-hr. [b]	B.t.u. coal	Thermal efficiency
Mar.	1	203,000	122,610	7,000	56	2.20	...	1.127[a]	15,553	13,800	21.94
	8	42,000	9,990	5,000	13	0.45	...	1.127[a]	15,553	13,800	21.94
	15	186,000	127,700	11,000	25	2.01	...	1.127[a]	15,553	13,800	21.94
	22	1,833,000	1,650,680	27,000	93	19.84	11.05	1.127[a]	15,553	13,800	21.94
	29	2,920,000	2,660,070	28,000	126	31.60	9.77	1.127[a]	15,553	13,800	21.94
Apr.	5	1,510,000	1,342,520	23,000	70½	16.34	12.48	0.949	13,091	13,800	26.06
	12	2,351,000	2,119,460	23,000	113	25.44	10.92	0.933	12,872	13,800	26.51
	19	2,782,000	2,540,400	35,000	119½	30.11	9.51	0.928	13,008	14,024	26.23
	26	3,819,000	3,520,800	51,000	137½	41.33	8.47	0.901	12,637	14,024	27.00
May	3	5,095,000	4,763,890	56,000	138	55.14	6.95	0.890	12,476	14,024	27.35
	10	3,710,000	3,396,630	30,000	145½	40.15	9.23	0.880	12,344	14,024	27.64
	17	3,118,000	2,862,270	26,000	125	33.74	8.93	0.987	13,844	14,024	24.65
	24	3,308,000	3,040,750	30,000	148	35.80	8.79	0.928	13,014	14,024	26.22
	31	3,926,000	3,581,940	30,000	168	42.49	9.61	0.952	13,345	14,024	25.57
June	7	4,187,000	3,823,540	33,000	168	45.31	9.51	0.959	13,454	14,024	25.36
	14	4,360,000	3,952,000	34,000	168	47.19	10.32	0.960	13,463	14,024	25.34
	21	3,946,000	3,591,210	32,000	155	42.71	9.88	0.919	12,894	14,024	26.46
	28 }										
July	5 }	277,000	32,480	(Out of service for repair on turbines, condenser, reheaters, and economizers.)							
	12	4,223,000	3,872,530	33,000	143	45.70	9.05	0.907	12,716	14,024	26.83
	19	4,899,000	4,519,360	42,000	149	53.01	8.41	0.983	13,786	14,024	24.75
	26	6,319,000	5,878,270	53,000	144	68.39	7.50	0.846	11,866	14,024	28.75
Aug.	2	6,337,000	5,904,260	54,000	126	68.58	8.17	0.902	12,652	14,024	26.97
	9	7,344,000	6,853,870	56,000	149	63.35	7.15	0.913	12,798	14,024	26.66
	16	8,030,000	7,467,490	54,000	168	86.90	7.53	0.918	12,977	14,139	26.29
	23	7,342,000	6,783,220	54,000	168	79.46	8.24	0.878	12,420	14,139	27.47
	30	3,429,000	3,068,620	37,000	144	37.11	11.74	0.961	13,594	14,139	25.10
Sept.	6	6,214,000	5,725,690	54,000	168	67.25	8.53	0.949	13,421	14,139	25.42
	13	5,359,000	4,942,900	52,000	168	57.80	8.42	0.893	12,624	14,139	27.03
	20	6,017,000	5,557,280	52,000	138	65.10	8.28	0.964	13,633	14,139	25.03
	27	5,791,000	5,323,190	54,000	144	62.67	8.79	0.889	12,572	14,139	27.14
Oct.	4	7,072,000	6,610,900	55,000	135½	76.54	6.97	0.892	12,610	14,139	27.06
	11	6,912,000	6,432,290	54,000	140	74.81	7.46	0.900	12,731	14,139	26.80
	18	7,140,000	6,642,570	54,000	139½	77.27	7.49	0.924	12,825	13,883	26.60
	25	7,690,000	7,162,170	56,000	144½	83.23	7.37	0.889	12,343	13,883	27.64
Nov.	1	7,723,000	7,199,870	56,000	156	83.58	7.27	0.870	12,071	13,879	28.27
	8	7,093,000	6,616,470	55,000	146½	76.76	7.20	0.916	12,708	13,879	26.85
	15	5,596,000	5,155,320	54,000	156	60.56	8.55	0.892	12,381	13,879	27.56

[a] Estimated. [b] Based on mine weights.

included the cost of and space required for the pulverized-coal bunkers. Of course it may make some difference on what basis these comparisons are made, and ample mill capacity is one point that should be insisted upon in making comparisons The reason for this may be illustrated as follows:

At the Middletown plant pulverizing mills were installed

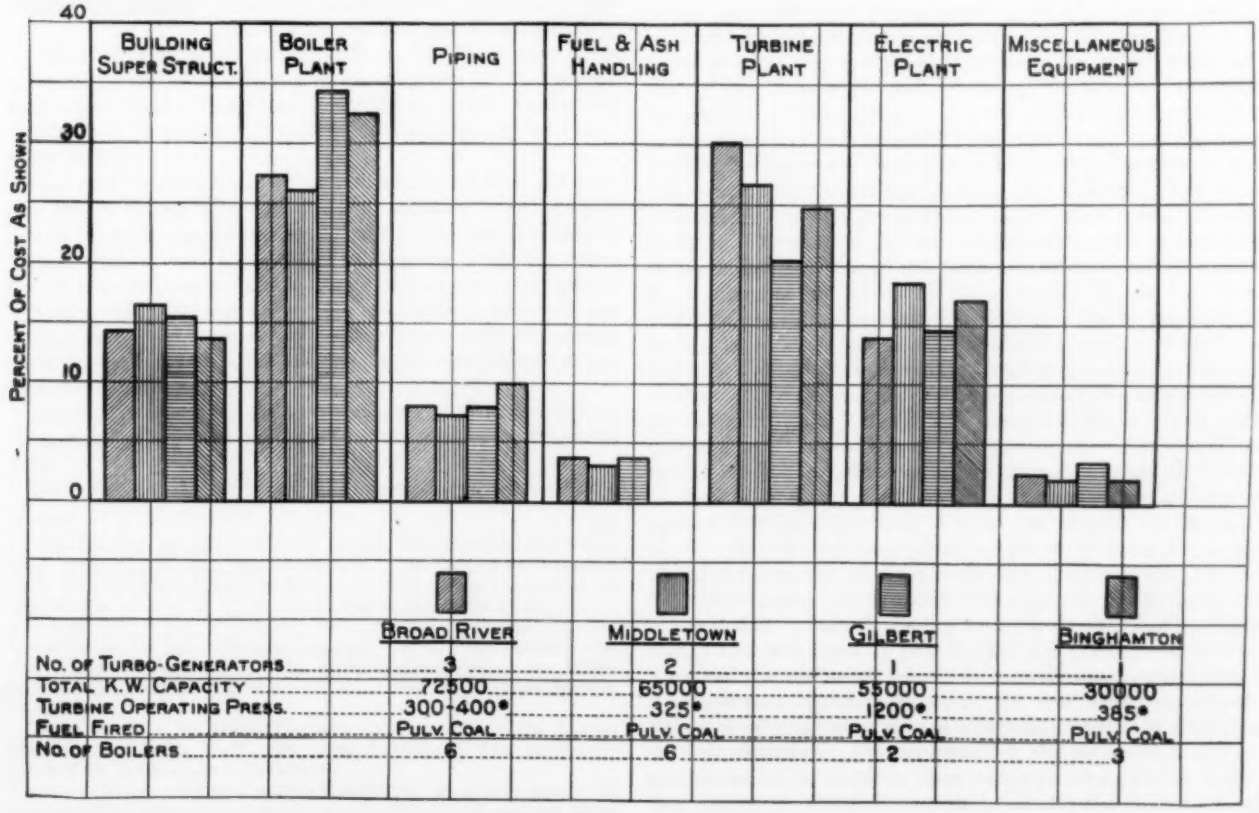


FIG. 10 COMPARATIVE COSTS OF FOUR STEAM-ELECTRIC STATIONS

which gave sufficient capacity, milling 16 hours a day, to run the station at its maximum load. During a strike which occurred about two years ago it was necessary for several months to obtain coal other than that ordinarily used, the result being that owing to the reduction in friability of this fuel the capacity of the mills was reduced by about 40 per cent, and it was necessary to operate these mills 24 hours a day, including Sundays, to keep the plant running at all. If this had been a unit system, with the same capacity in mills, nearly half the capacity of the plant would have been out of service.

Therefore when a unit system is installed it is felt that the mills used should be of greater capacity than would be necessary with a bin system in order to care for emergencies which may occur at any time due to strikes or failures, from any cause, of the mines or railways supplying the fuel.

There is a preference for the flexibility obtained by a limited bin system built in such a way that the benefit of mill drying and burning of vent dust is accomplished as it is in the Gilbert Station. There is also the ability to shut down the mills on the peak hours or in cases of emergency.

As to the matter of turbine scaling, this is in no way peculiar to high-pressure stations, but occurs in low-pressure plants as well.

As to the availability of the feed pumps there are as yet no definite figures on this. Generally speaking, the author may say that it never has been necessary to shut down the plant or to reduce the capacity because of trouble with the pumps. Of course there is a spare pump which always can be cut in. Availability factors may be obtained after all changes are made and the pumps are definitely on a permanent operating basis.

FSP-53-20b

Operating Experiences, Deepwater Station

By K. M. IRWIN,[1] PHILADELPHIA, PA.

The physical characteristics of the Deepwater Plant are described, emphasizing only those details which are relatively new or those which have had a distinct bearing upon operating experiences. The paper gives a history of the operations from the time the plant was first started in March, 1930, up until the present, summarizing the difficulties that have been encountered and attempting to segregate them between those that are directly chargeable to the pressure or temperature and those which have been encountered due to other features of design. Due to the fact that a large portion of the plant has been in what might be termed commercial operating service only since June 1, or for three months at the time this paper was prepared, very little authentic data on operating efficiencies can be given.



PART I GENERAL

THE Deepwater Plant was built by the American Gas and Electric Company and The United Gas Improvement Company for their respective subsidiaries, and is centrally located for these properties on the Delaware River in the southern part of New Jersey. The initial installation consists of two General Electric Company compound turbines, which have a nominal manufacturer's rating of 53,000 kw., but have been operated satisfactorily up to 58,000 kw., and one high-pressure turbine, which is an exact duplicate of the high-pressure elements of the beforementioned compound machines, but which is used for the benefit of the du Pont company, whose plant is adjacent to the Deepwater Station.

The boiler room has six boilers, which are so arranged that two are to supply each high-pressure machine. Out of each pair of boilers supplying the compound turbines, one is equipped with a reheater and a small-size economizer, while the other has no reheater, but a large-surface steaming-type economizer. The two boilers opposite the du Pont turbine, however, are equipped only with economizers. All six boilers have air heaters, those on the reheater boilers having 14 per cent larger surface than those on the boilers with steaming economizers.

The fuel is fired in pulverized form, the direct-fired system being installed on three boilers and a bin system on the other three.

The boilers are designed for 1350 lb. operating pressure and the turbines for 1200 lb. The initial temperature admitted to the high-pressure cylinder is 725 deg. fahr., and at the exhaust to the high-pressure cylinder this steam is reheated in the flue-gas reheaters to 750 deg.

Arrangement of du Pont Apparatus

It is necessary to describe briefly the arrangement of the equipment which was installed for the benefit of the du Pont company before going into a description of the plant heat cycle, as the design is contingent upon the working in of the separate cycle which was necessary in order to supply the industrial demand.

The du Pont turbine is a duplicate of the high-pressure elements of the compound machines, and if in the future it is found desirable by either party to discontinue the arrangement with du Pont, a low-pressure element may be added, so that this portion of the plant becomes a duplicate of the other two portions devoted to the power companies.

The turbine exhausts into evaporators in order to condense the plant steam and permit this to be saved and pumped back into the boiler plant. Raw water at 180 deg. is delivered to the evaporators by the du Pont company, and is converted into steam at 180 lb. gage pressure and superheated 40 deg. in the steam superheaters. The condensed plant steam or evaporator drips are picked up in individual hot wells on each evaporator and pumped forward by drip pumps into the suction of the boiler-feed pumps, which are also duplicates in size and characteristics of the boiler-feed pumps on the power companies' units. All boiler-feed pumps discharge into a common header. A considerable problem has presented itself in the operation of the evaporator drip pumps, as the pressure at full load is approximately 400 lb. and the temperature of the drips at saturated temperature of this pressure. A great deal of the pump trouble in the plant has been caused by the difficulty of maintaining a level in the hot wells of the evaporators sufficiently high to give the pumps their proper submergence and sufficiently low to be able to properly vent the entrained oxygen from these hot wells. Flashing difficulties in the drip pumps are quickly transmitted to the boiler-feed pumps and have caused considerable trouble in maintaining the boiler-feed pump seals. It has only been with considerable experience and skill that the operators have been able to maintain an equilibrium in this portion of the cycle and at the same time keep the discharge of the boiler-feed pumps at the same pressure as the discharge pressure of the feed pumps for the other units. If the feed pumps on the du Pont cycle deliver a higher pressure in the boiler-feed mains than the pumps on the compound-turbine systems, this would cause the boilers to draw the majority of the water from the du Pont boiler-feed pumps, draining this system of water and backing up the water in the other two systems, while, on the other hand, if the discharge head in the du Pont boiler-feed pumps is too low, the boiler-feed pumps of the other two systems which supply all of the feedwater shut off the flow from the du Pont boiler feeders, and in that way back up into the hot wells of the evaporators. As all of the pumps of the du Pont cycle are constant speed, it is necessary therefore to adjust the final discharge pressure either by varying the number of drip pumps used or by throttling on their discharge. As later described, the pumps on the power companies' units, however, have variable-speed primary boiler-feed pumps, so that it is possible to control the discharge head on these pumps.

---

[1] Mechanical Engineer of The United Gas Improvement Company, Philadelphia, Pa. Mem. A.S.M.E. Mr. Irwin was graduated from Yale Sheffield Scientific School in 1915. The same year he entered the employ of B. F. Sturtevant Co., Hyde Park, Mass.; later was with Stone & Webster, Boston, Mass. He served in the Navy during the war. He has been with The United Gas Improvement Company since 1919. Mr. Irwin is Chairman of the Philadelphia Local Section, A.S.M.E., and a member of the Prime Movers Committee, N.E.L.A.

Contributed by the Power Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

At the present time, the regulating is done by using water-level regulating valves on the evaporator hot wells, which allow a small portion of the water from the discharge of the boiler-feed pump to be bypassed back into the evaporator hot well. The designers now are considering installing considerably larger hot wells on the drips from the evaporators, in order to supply greater storage at a given head elevation over the drip pumps.

Heat Cycle, Power Companies' Units

The low-pressure cylinders are bled at three points, and a fourth bleed point is provided at the exhaust of the high-pressure unit. The condensate pumps discharge through the first stage of bleeder heating into a deaerator. This deaerator also receives the various plant drips and the make-up water from the plant evaporators. Connections are provided to the deaerator from the second and third bleed points, which are controlled by thermostat-operated valves in order to maintain 214 deg. fahr. to insure complete deaeration. Immediately below the deaerator, a large storage tank is provided, and from this tank the primary boiler-feed pumps or tank pumps take their suction and discharge through the second, third, and fourth bleeder heaters into the suction of the boiler-feed pumps. These tank pumps are variable speed, and all of the boiler-feed pumps are constant speed, so that the pressure in the boiler-feed main is maintained by varying the speed of the tank pumps, and in that way varying the suction head of the boiler-feed pumps.

The first-stage bleeder heater is provided with a drip pump which discharges into the deaerator. The fourth-stage bleeder heater, which takes its steam from the exhaust of the high-pressure turbine, is also equipped with a drip pump which discharges into the line leaving the fourth-stage heater. The drips from the second- and third-stage bleeder heaters are cascaded back into the deaerator.

The only modification which has been made to this system since the plant has been put into operation has been to supply additional cold-water storage by installing a storage tank on the discharge of the condenser condensate pumps. These storage tanks float on the line, and admit water to the system when the float valve opens up on the tank below the deaerator and receive water from the system as this float valve closes.

Boiler and Furnaces

The boilers are all of the Babcock & Wilcox Company cross-drum type, nine tubes high, without baffles, the gas passing straight up through the boiler directly into the economizers on four of the boilers and into the reheaters on the other two boilers. The boilers which are equipped with reheaters also have a small economizer below the reheater in order to reduce the temperature of the gases before entering the air heaters.

All four sides of the furnaces are equipped with water walls, consisting of bare tubes on 6 in. centers backed up with refractory walls. The bottoms of the furnaces are flat, and are designed for slag tapping. Originally, two of the furnace bottoms were lined with dolomite, two with mica schist, and two with cans filled with coke.

Pulverized-Fuel System

The owners felt at the time the plant was designed that the advisability of utilizing either the bin or the direct-fired system had not been definitely determined, and it was therefore thought advisable to equip half of the plant with a bin system and the other half of the plant with a direct-fired system. It seemed that each system had very distinct advantages and that this would offer an opportunity for the two owning companies to gain experience which would prove of value to them in their other stations.

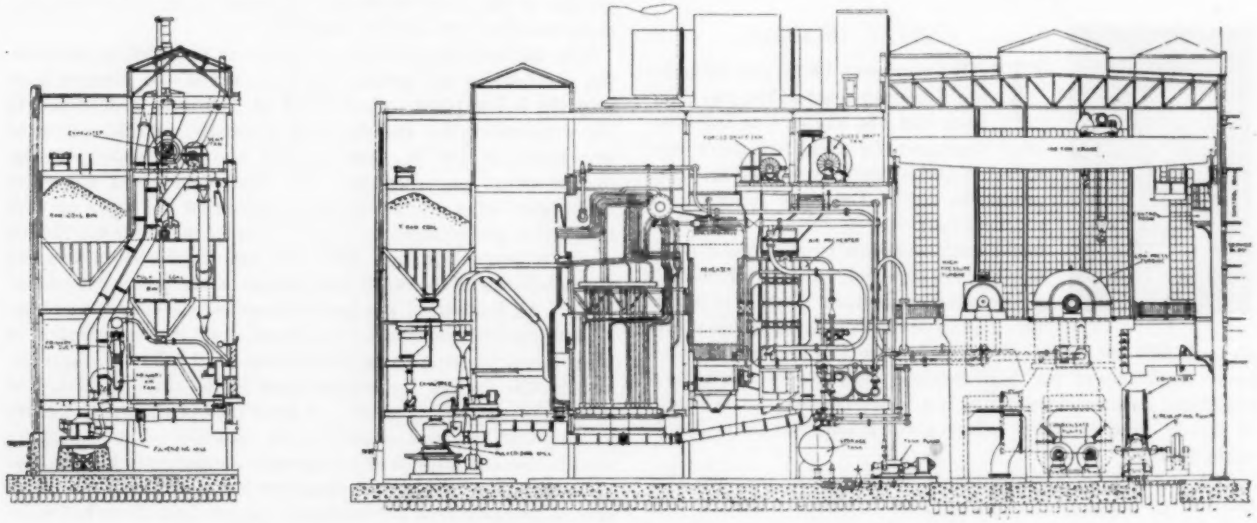


Fig. 1 Elevation of Boiler and Turbine Room in the Deepwater Power Station
(At left, elevation through bin system; at right, elevation through direct-fired system.)

This plan was adopted, the three boilers on the bin system having one mill each of 20-ton capacity. The mill exhausters take their suction out of the mill cyclones in place of discharging into them, as was the more general practice at the time the plant was designed. Mill drying is done by taking a portion of the air out of the preheated-air ducts into the mills, and providing a vent or leak off fan to take a portion of the air out of the mill system beyond the discharge of the cyclone exhauster, and discharge this air into the secondary-air chamber around the burners. It is felt that, with this arrangement, two of the most serious objections to the bin system had been overcome—namely, the elimination of driers and the elimination of cyclone vents, which have in the past proved a dust nuisance. The cyclones on each mill discharge directly into the powdered-coal bins in front of the boiler. A screw conveyor has been provided so that it is possible to utilize any mill on any boiler. Each of the

bin-system boilers is equipped with six burners, which are fed by the feeders directly below the powdered-coal bunkers. The burners are of the Combustion Engineering Couch type.

On the direct-fired system, two mills of 10 tons capacity are provided on each boiler, each mill discharging up to three burners. The burners are the Fuller-Lehigh Calumet cross-tube type.

The designers laid out the building and required the manufacturers of the two respective systems to work their equipment into exactly the same space requirements, so that there can be charged no additional investment for building space against either system. The final installed cost of the two systems, including all necessary wiring and control equipment, was very close, so that the ultimate choice between these two systems must lie in features of flexibility, reliability, efficiency, and general appearance, including such items as light and ventilation.

Ash-Handling System

The slag is tapped from the boilers into a disintegrating chamber, with high-pressure water jets to break up the slag stream and quench it. A Coghlan ejector is provided to draw the water and slag out of the disintegrating box and discharge into a 10-in. pipe line, which carries the slag out to the yard.

placed by either soft-steel or corrugated Monel-metal gaskets. Difficulty was also encountered with leaks on the bonnets of some of the valves, and again it was found due to the disintegration of composition gaskets, which have been replaced.

The economizers, which have been subjected to as high as 1800 lb. pressure, are the B. & W. return-bend type with copper gasket joints. In general, it may be stated that these have been satisfactory, in that only one of the boilers has been troublesome with leaks, and on this boiler it was found that most of the leaks have been due to the gaskets being improperly installed or that the faces have not been properly prepared to receive the gasket. It has not been necessary to repair any one joint more than once. Generally, the same observation has been made of the water-wall tubes. Some of the boilers have had a number of leaks, but in nearly all cases after these had been rerolled, no further difficulty has been experienced. In a few cases where there has been a repetition of the difficulty, it was found to be traceable to the fact that the water from the quenching on the slag-tap system has slopped over on these tubes before the covering was completed.

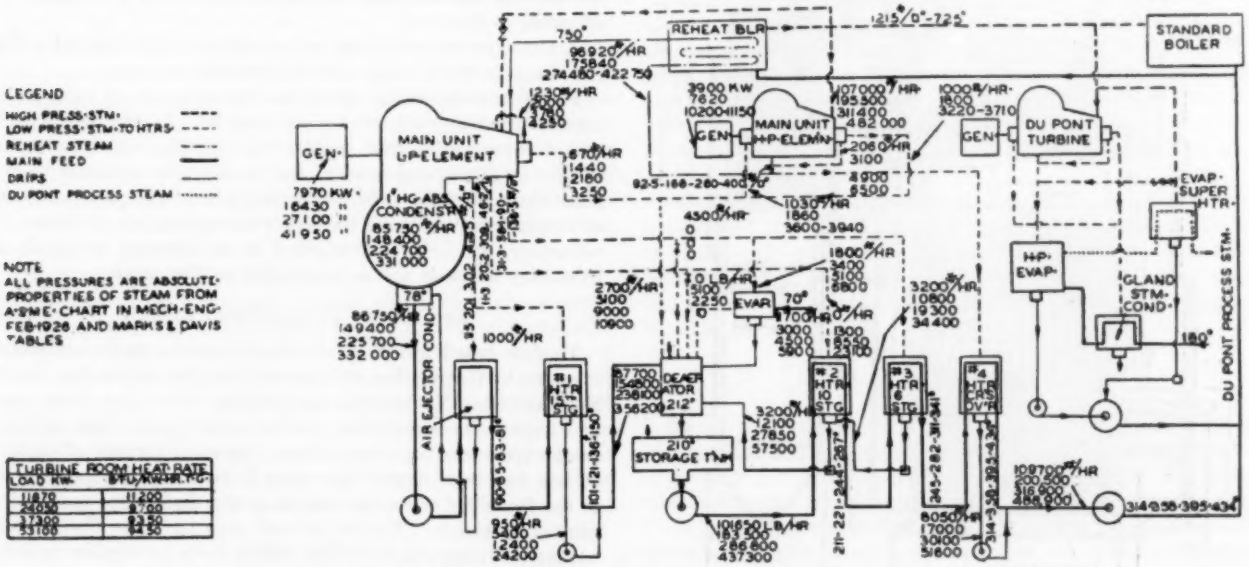


Fig. 2 Heat Balance Diagram of Deepwater Power Station

PART II OPERATING EXPERIENCES

The author wishes to again emphasize the fact that at the time this paper was prepared the plant had been in commercial operation only a little over three months, and therefore had not had a long enough running period to give definite figures on the economies obtained or on the operating efficiencies of the various parts of equipment.

Many difficulties, such as are encountered in new plants and especially those with novel features of design, have come up, for some of which remedies have been provided, and others are still being treated.

Many of the fears of 1200 lb. have been centered around the piping, and it can be stated that, to date, there have been no leaks on the larger sizes of high-pressure piping, which is equipped with the Sarlun joints. Some difficulties have been encountered with the smaller size pipe, which has been used for drips and instrument connections. Most of this is chargeable to the use of composition gaskets, nearly all of which have now been re-

The reheaters and the superheaters have been entirely satisfactory, both as to their operating characteristics and types. The temperatures at the outlets of both the superheaters and reheaters have been very close to the guaranteed condition. Summing up, it might be stated that there had been no difficulties in the boiler room traceable to 1200 lb. pressure selected.

Operating Difficulties Not Traceable to Pressure

Other difficulties encountered in the boiler room which are entirely foreign to the pressure or temperature have been with the slag-tap furnace bottoms, boiler side and rear walls, and the pulverized-fuel system.

Since this plant was designed, a great deal of development work and knowledge have been gained on slag-tap furnace floors at other stations, and difficulties which are gradually growing, but of which the operators of other plants were not cognizant, have come to light. The first of these was that boilers which are taken off of the line and put back at fairly frequent intervals had their furnace floors grow and push the side walls out. It was found on inspection that this was due to having the bottoms crack in cooling down, and the cracks and interstices fill

with molten slag when the boilers are again heated up. The repetition of this operation gradually pushes the walls out, so that it is necessary to cut back the furnace bottoms adjacent to the side walls and to allow the walls to spring back or to be pushed back into shape, and to start over again.

This was brought to the attention of the owners of the Deepwater plant after two of the boilers had been installed with dolomite bottoms similar to other installations, so that the next two boilers were changed, and mica schist was used in place of dolomite. At one other plant with a slight amount of experience with mica schist, it appeared that the slag did not penetrate in the same manner that it did in the dolomite. Later operating experience at Deepwater, however, has proved that dolomite bottom is much superior to the mica schist. To date there have been only two cases of leakage of slag from the furnace bottoms covered with dolomite. Neither one of these

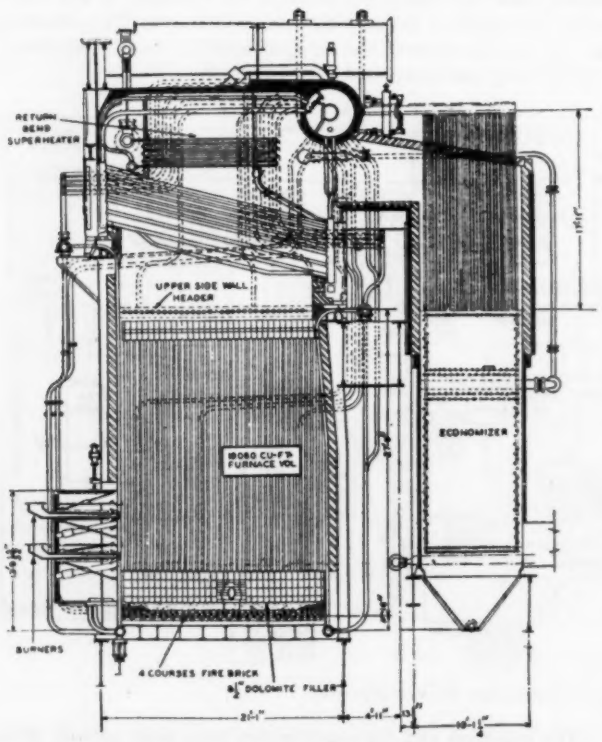


FIG. 3 SECTION THROUGH ONE OF THE STANDARD BOILERS IN THE DEEPWATER POWER STATION
(There are four standard and two reheat boilers installed in the plant.)

was serious, and both were immediately stopped by chilling the slag leak with water, and after the furnace was shut down, cutting back the floor approximately 6 in. along the two sides, allowing the walls to assume their normal position, filling again with silica sand (a sand used in glass making) and dolomite. Summing up, it might be stated that bottoms of this type can be operated without a great amount of trouble if the side walls are carefully watched for deflection and if the boilers are not taken off and on the line too frequently, and operating periods of a year or more may be obtained without any serious maintenance work to the bottoms.

About the time the first four boilers mentioned went into operation, the owners were advised by the manufacturers that, on the basis of laboratory tests where they had used slag tapped from furnaces of other companies, they had found that coke or carbon was a material which the slag would not penetrate. This was also confirmed by investigation of practice in the metallurgical furnaces. Therefore, the last two boilers were designed with bottoms of radically different design, and coke cans were placed on top of 5 in. of silica sand, and above this additional coke and coking coal were spread, in order to form a solid coke bottom. In other plants carbon bricks are used. It is now found that these last two boilers with their coke bottoms have been the least satisfactory of any of those installed. The slag contains a large amount of iron sulphide, which, due to its weight, sinks to the bottom of the slag bath and is therefore never drawn out through the slag tap. This material is very heavy, penetrating down through the coke and entirely consuming it, so that on one boiler with this design of bottom, which has been taken off the line and cut into, it was found that all remnants of the cans of coke had entirely disappeared, and the only thing that was left was approximately 22 in. of iron sulphide and approximately $^3/_4$ in. of the silica sand over the steel-plate bottoms.

The different performance of these bottoms in practice over the laboratory tests is accounted for by the fact that the tests were run with slag taken out of other boilers, and therefore did not contain the iron sulphide which is actually obtained in an operating boiler.

It has now been decided to reconstruct with magnesite the two furnaces which originally had the coke bottoms.

The side walls earlier described are made up of bare plain tubes backed up with refractory material. It has been found that the rear wall, that is, the wall opposite the burners, is subject to severe slag erosion, and it has been necessary to replace three of these walls. Combinations of the particular ash of the fuel that is being burned, with various compositions of refractory, are being investigated in an attempt to obtain a refractory which is not so susceptible to slag erosion.

BIN VERSUS DIRECT-FIRED SYSTEM

To date it is impossible from the records to make any statement as to the relative efficiency of the bin versus the direct-fired system. No careful evaporating tests have been run, and from normal operating data it would appear that the systems were operating practically at the same furnace efficiency. It has also been found that there is no marked improvement in the flexibility of the bin system over a direct-fired system for normal operation. The bin system does have the distinct advantage of being able to cut the boilers down to very low ratings. This feature was taken advantage of at the time the plant was originally put into operation, and only small quantities of steam were required for turning turbines over for drying out, operating auxiliaries, etc. The direct-fired system has a distinct low-rating limitation.

It is well to mention here that, due to not being able to run the furnaces at extremely low ratings, it was necessary to utilize oil in drying out the boilers after erection. This difficulty which was experienced with this oil has received considerable publicity, and for that reason it seems advisable to describe the accidents which occurred.

Apparently the oil fire was so small for such a large black surface furnace that the oil was not entirely consumed, and unburned oil vapors passed on up through the boiler and condensed in the breechings and stacks, forming a heavy, greasy soot. On two occasions the soot in the stacks and breeching caught fire, and on one of these occasions two men were fatally burned.

The direct-fired end of the plant presents a very much more desirable appearance, in that the pulverized bins of the bin system are bulky and cut off considerable light and air, so that a visitor to the plant is much more favorably impressed by the general arrangement at the direct-fired system end. The one counterbalancing feature, however, has been the more reliable operating

condition existing at the bin end of the plant. On a number of occasions it has been necessary to cut back the load which the plant was carrying, due to failure of coal supply to furnaces from the direct-fired system. A number of these interruptions were caused by small adjustments which were needed on the mills themselves. These have been made, and on one of the boilers which caused a drop in load four times during the month of April. due to a faulty lubrication system on the mill and which was corrected at that time, no interruptions have occurred since. On three of the mills which are of the Fuller-Lehigh table B type, trouble is still being experienced with the loss of fuel. It is felt that one of the difficulties with these three mills is that in normal operation they are extremely quiet and do not become noisy even when the coal feed fails, as is the case with the other mills. Therefore the operators are not cognizant of this condition as quickly, and a tie-up in the fuel actually causes loss of steam pressure at times before the condition can be corrected. It should be emphasized, however, that this again is a matter of operating experience, and it is believed in time that a great deal better record will be obtained.

Difficulty has been experienced with the coal hanging up in the raw-coal bunkers and sticking in the connections between the raw-coal bunkers and the scales which are installed between the raw-coal bunkers and the mills, jamming in the scales or in the magnetic separators and catching small particles of wood or foreign material on the Star feeders of the mills themselves. Two cases occurred where a failure in the mill of wearing parts was not caught in sufficient time to prevent the mill from failing while in service. The records show that difficulties have been steadily decreasing, however, and it is hoped that very soon they will be entirely eliminated.

As the author is a representative of the company which advocated putting in the bin system, any prejudice can be excused, for he now is of the opinion that it is safer and more comfortable to have a small supply of powdered coal on hand in order to take care of unforeseen difficulties which may happen in the fuel-supply system which will be absolutely no fault of the particular mill which is being used, and that in the Deepwater plant ten hours' supply of powdered coal is probably far more than is necessary. The recommendation would be that a two to three hours' supply would make an ideal practice.

Turbine Room

Difficulty has been experienced in the turbine room with silica scale deposited on the last rows of blading and nozzles in the high-pressure turbine which is used in the du Pont system. This difficulty has been entirely peculiar to this one machine out of the three high-pressure machines. To date no evidence of scale has been found in the other two high-pressure units. It has been necessary to shut this machine down for periods of two weeks in order to scrape the scale off the nozzles and blading by hand. This has been a laborious and expensive job. In order to correct this condition, purifiers have been installed on the vapor outlets of the house evaporators to cut down the carry-over silicate in the make-up water. Since these have been installed, the purity of the vapor in the evaporators has been considerably increased, and the turbine has operated for a period of several weeks without showing any symptoms of scaling. Similar difficulty has been experienced at other plants, and it has been found that the scale can be taken off the blades by admitting water to the steam in the high-pressure throttle to the extent of having the steam in a completely saturated condition. After operating for several hours under this condition, it is found that the scale can be entirely washed off, so that while it is felt that this difficulty has been overcome at Deepwater, it is comforting to know that, if a repetition of this should occur, a remedy has been found. This is one of the troubles which can be directly charged against the pressure utilized in this plant, as the other plants which have experienced this trouble are also operating at 1200 lb.

Silica is soluble in water at superheated temperatures, as is evidenced in many low-pressure plants by the eating away of gage glasses on the boilers, which is a common phenomenon. Apparently as water temperatures increase, it increases the solubility of the silica, so that at temperature corresponding to saturated temperature of 1350 lb., which is 582 deg. fahr., this action is extremely pronounced.

Each of the high-pressure elements on the compound units has developed leaks on the horizontal joint. These leaks occurred on successive days and did not occur on the high-pressure unit of the du Pont system at any time, and the du Pont machine was not operating two days that the difficulty was encountered.

Considerable difficulty has also been encountered with the government mechanism and control valves on the high-pressure turbines. The manufacturers have made certain changes, and it is now felt that this has also been overcome.

On several occasions the operators have attempted to run the low-pressure element of the compound machines without the high-pressure, utilizing a control valve which was provided with the turbines to admit the high-pressure steam directly into the low-pressure elements. It has been necessary for the manufacturer to redesign this valve, as to date it has been impossible to operate in this manner without breaking the stem on the controlling valve.

In closing, it may be stated that very little difficulty has been experienced due to the pressure and temperature conditions selected for this plant, and of the difficulties which have occurred, the operators feel that they are well along on the road for remedies and corrections, which, it is hoped, can be outlined in better detail in the near future.

Discussion

Grant Campbell.[2] The problem presented by The Allied Engineers was to maintain a safe minimum water level in seven evaporator hot wells, each having a constant-speed drip pump which delivers the condensate to the boiler-feed pump suction.

Owing to the variable demand for process steam, there are times when the condensate is very much less than the capacity of the drip pumps, and this condition led to insufficient suction head, subjected the drip and feed pumps to failure, and otherwise affected the plant operation. It is also true that too high a level of water in the evaporator system lowered the efficiency of the evaporators and of the high-pressure turbine.

The plan adopted by the engineers provides for bypassing water from the boiler-feed pump discharge to each hot well when its water level is below a definite level. Seven Campbell water-level regulators of the boiler-feedwater type were installed, one for each hot well, as indicated in the diagram of connections and general arrangement, Fig. 4.

The principle upon which the operation of this regulator is based is that when a mixture of steam and hot water under pressure passes continuously to atmosphere through a system of matched orifices, the intermediate pressure between any two orifices will vary with the moisture content of the mixture.

A suitably treated sampling pipe is set in a water column with the open end below the desired average water level, and there is a continuous minute flow up the pipe to the orifice system. This flow obviously will be all steam at extreme low-water level or all steam at extreme high-water level or all water at extreme high-water level. Between these limits the moisture content of the

[2] Atlas Valve Company, Newark, N. J. Mem. A.S.M.E.

fluid mixture will increase as the water level rises, and vice versa.

The regulating element is shown in the diagram, Fig. 5, connected to the water column and with a tapping for receiving the sampling pipe.

In the element is a pair of orifices scientifically proportioned so that as the mixture of steam and water passes through the orifices to atmosphere the pressure in the element will increase or decrease as the percentage of moisture increases or decreases, or as the water level rises or falls.

The chart, Fig. 6, showing a relation between the element pressure and the water level, gives a typical condition taken from an actual boiler test.

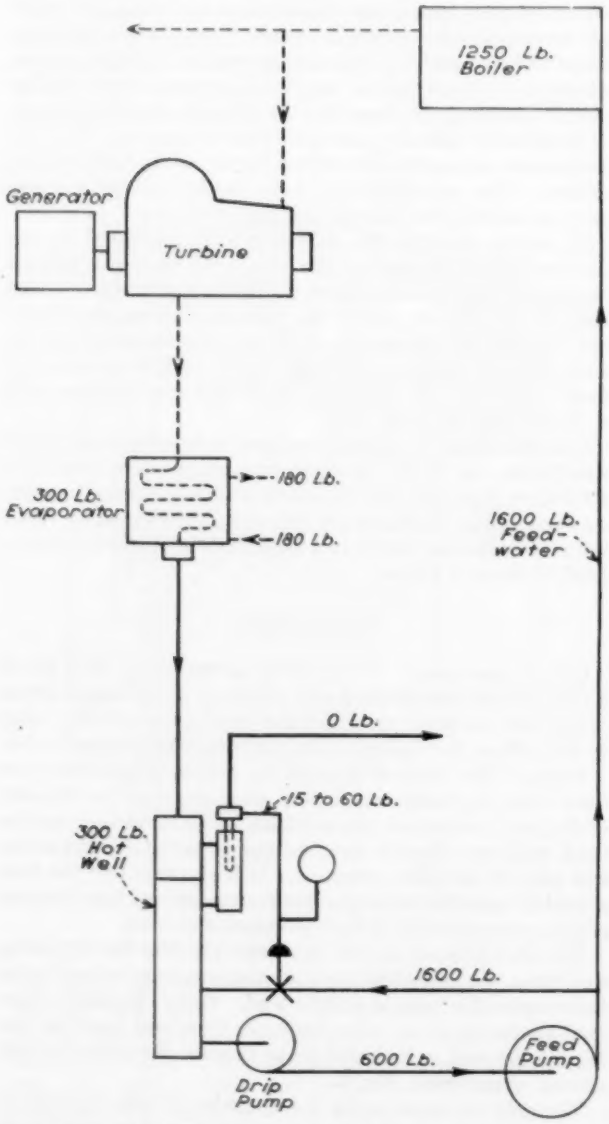


FIG. 4 INSTALLATION OF WATER-LEVEL REGULATORS

Since there is a definite pressure change in the element with any water-level change, this variable pressure applied to a pressure-actuated valve regulates the flow of input water in accordance with the water level.

In the control valves installed at Deepwater this pressure has been imposed upon a spring-loaded diaphragm which actuates a pilot stem, as shown. This arrangement leaves little for the pressure to do but to lower the pilot stem with increased water level, and vice versa.

This pilot stem, rising and falling inversely with the water

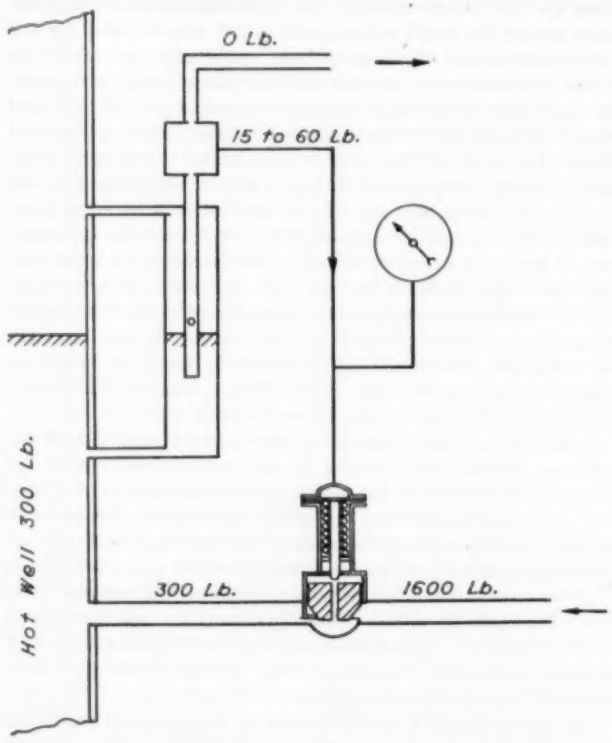


FIG. 5 REGULATING ELEMENT CONNECTED TO WATER COLUMN

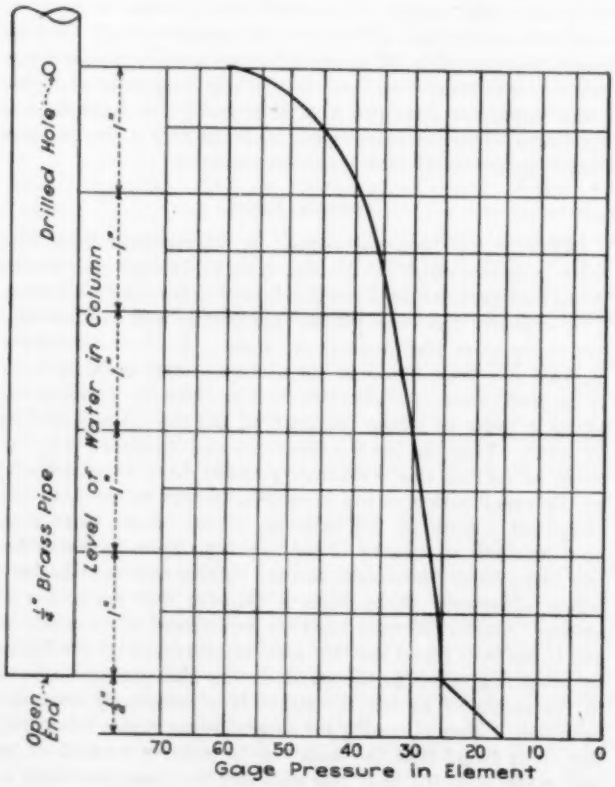


FIG. 6 RELATION BETWEEN ELEMENT PRESSURE AND WATER LEVEL

level, is aligned with a poppet loosely fitted in the valve body at the upper end and having a conical surface at the bottom end to contact with the seat and close the main valve port and with a vertical pilot port opening at the bottom into the discharge side of the valve but subject to restriction at the top by the pilot stem.

The areas and the balances used in the design are such that in operation the valve poppet always follows closely the movement of the pilot stem without offering resistance to it, the valve port opening or closing as the water level lowers or rises.

It will be seen that the regulator combination admits water to the hot well when the level of water is lowered, but provides a tightly closed valve port at high water, the valve in operation being always in perfect balance.

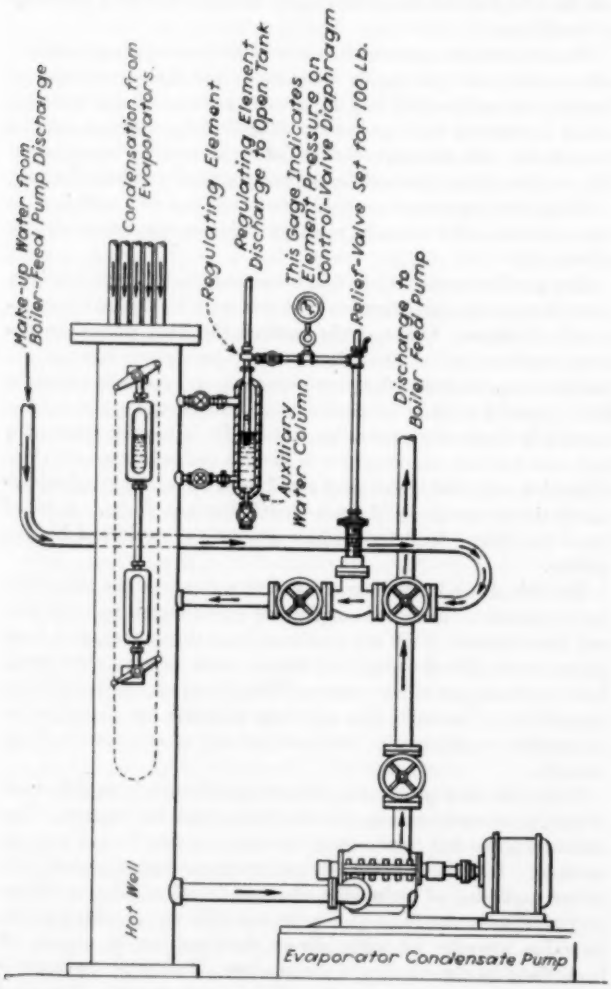


FIG. 7 WATER-LEVEL CONTROL CONNECTIONS AT DEEPWATER STATION

The conditions encountered in the Deepwater operation are different in many respects from those met in boiler-feedwater regulation, but the principles are the same. The diagram of connections at Deepwater Station is shown in Fig. 7.

The hot wells are 18 in. in diameter and hold very little water, so that the regulators have been made to respond accurately and rapidly to maintain the level within a few inches.

The steam pressure in the hot wells is varied from 250 to 350 lb., and the boiler-feedwater pressure may be from 1500 to 1800 lb., so that the control valves have been constructed to be capable of adapting themselves to widely varying loads and drops in pressure and still retain their accuracy.

For the best operation of these level-control units it is essential that there be a true water level in the water column to regulate by, and that at all times there be water in sufficient quantity and at adequate pressure to hold the desired minimum level.

For several months these level-control units have successfully maintained a safe low-water level in the hot wells materially assisting in the operation of the power station and thereby have justified the designing engineers in selecting the type of equipment used in this pioneer level-control operation.

JAMES M. TAGGART.[3] The author is to be congratulated on his frank and clear statement of the troubles encountered during operation. It is especially noteworthy that none of the real troubles was due to the high pressure of operation. The performance of certain equipment and arrangements covered in the paper merits careful consideration. Further information is, however, required before reaching a judgment as to the results obtained. The writer is taking up a few points that impressed him, with this in view, not in a spirit of criticism.

Pumping hot water at or near saturated-steam temperatures is a common source of trouble where it has been attempted in central stations. Judging by the numerous troubles encountered and from statements made to the writer, there seems to be an extended lack of comprehension of the factors involved. At the Deepwater Station, according to the paper, no trouble occurred so long as the proper water level was maintained. The requirements are of course that the head shall be sufficient to maintain a pressure in the pump above the flash point. The amount of head required therefore depends on the design of the pump and piping connection. The pressure of the liquid pumped is immaterial except as it affects the design with respect to strength and leakage. Provided a suitable type of propeller pump or pump with the first stage of the propeller type be used, similar to certain types of condenser hot-well pumps, much lower heads can be used with safety. Some types of high-speed centrifugal pumps require excessively high heads on the suction side. It would be informative to know the type and arrangement of the drain pumps used on the evaporators at Deepwater and also the minimum head at which they operated successfully. The lack of provision for maintaining a constant head in the evaporator hot well would seem to have been an oversight. The large temperature difference required for maximum load in the evaporator, namely, 72 deg. fahr., would seem to be excessive. It is possible that the maximum noted is more in the nature of an emergency loading. Otherwise either there is apparently an undue restriction in first cost or the evaporator efficiency is very low. A temperature difference of 30 deg. fahr. would give an approximate increase in power output of 20 per cent. With a good design of evaporator this temperature difference would not require excessive surface compared to the value of the return. A short description of the type size and temperature difference required for various loadings would be instructive.

The slag erosion of the rear boiler-wall refractory between the tubes was apparently due to flame impingement, since the side walls did not suffer. The form of burners and fineness of pulverization would determine the amount of impingement to a large extent. It would be interesting to know the fineness used and what, if anything, was done to shorten the flame. Running slag will wash away any ordinary refractory. Normal changes in refractory would at the best only prolong the life slightly unless greater cooling effect is obtained. In this respect carborundum brick pressed against the back of the tubes might give the extra cooling action needed.

[3] Consulting Engineer, New York, N. Y. Assoc-Mem. A.S.M.E.

Slag-tapping furnaces, judging from the experiences at Deepwater and elsewhere, are still somewhat in the experimental stage. Some method of taking care of the expansion is evidently needed. Bumping or bagging the furnace floor, combined with provision for restraining the side flow, might divert the expansion so as to prevent any continued enlargement.

The author's comments on the relative merits of the unit and bin system for pulverized coal possibly apply more especially to the conditions at Deepwater. There are unit systems where the reliability factor is high. However, it is the writer's opinion that the merits of both systems might be combined by the use of relatively small bins, either in parallel with direct feed or as feed bins. It is certain that this arrangement would provide for the low load operation and would give better combustion control at all loads than the unit system alone.

E. G. Bailey.[4] The Deepwater Station is a very interesting group of equipment, and the author has presented the picture up to date in a very frank and fair manner.

The interesting feature of this station is that it represents a fairly true cross-section of the diversity of engineering opinion in regard to certain phases of power-plant equipment. It is to be expected that there should be a diversity of opinion when developments are in progress. Some engineers are inclined to stay by the true and tried, while others are always reaching for the newest equipment offered, and still others endeavor to follow the old adage, "Be not the first by whom the new is tried, nor yet the last to lay the old aside."

As to the relative merits of the direct-fired and the bin systems, this question will not be finally settled to the satisfaction of all engineers throughout the country from results of this one station, because operating conditions, load factor, and the details of crushing and feeding coal have a great deal to do with the correct conclusions under different circumstances.

There are a great many stations as large or larger than Deepwater which are operating entirely satisfactorily on completely direct-fired units and handling an even worse grade of coal than is supplied to this station.

As to relative cost of bin system versus direct firing, the author states that the final cost was very close. This is not always the case. Sometimes the problem works out to an additional cost of 25 to 100 per cent for the bin system over the direct firing. Under certain circumstances the advantages of flexibility and other features may make this increased investment advisable, while in others there is certainly no justification for it.

As to the building space occupied, while the two systems were adapted to go into the same space, yet a glance at the two views of Fig. 1 indicates that building space could have been reduced for the direct-fired system.

The statement that it was necessary to replace three of the furnace walls due to slag erosion should be explained a little more fully; that is, the replacement consisted of the refractory back of the bare tubes. These tubes are $3^1/_4$ in. in diameter on 6-in. centers and have blocks extending up a short distance above the slag line. There has been no trouble with the portion of the wall covered by blocks. Above this, however, the fusing temperature of ash and the rate of combustion, which is approximately 20,000 B.t.u. per cu. ft., have resulted in a combination a little below the proper safety factor for continuous operation without erosion and expensive maintenance.

This bare-tube and tile type of furnace-wall construction has worked out satisfactorily under other conditions, but its unsatisfactoriness for this particular place indicates again what small differences in operating conditions are often magnified in the final result.

[4] President, Fuller-Lehigh Company, Fullerton, Pa. Mem. A.S.M.E.

As to the slag-tap furnace floors, these were installed at a time when this story was in the state of flux and the opinions of different engineers were evident from the variety of materials tried. Here is a case where the original design of brick and dolomite has worked out best, and the expected improvements in the way of mica schist and carbon were unsatisfactory, although under other slightly different conditions both of those materials have given better results. The problem, of course, for an engineer is to find a type of construction that will meet all conditions, or else if there is too great a difference in cost in applying this standard to some cases where punishment is not so severe, then he should have very definite facts and knowledge available so as to use discretion in selecting the proper construction for a given set of conditions.

Those who have operated furnaces of different arrangement for ash removal are practically unanimous for the advantages of tapping out molten slag and are greatly in favor of this method, and it is believed that temporary difficulties in working out the best design will not appreciably deter the coming adoption of this method of ash removal for the great majority of installations.

While our experience on this problem is not yet sufficient to be conclusive, a few remarks covering a diverse experience may be of interest.

The usual comment when first encountering this problem is to go to the metallurgical furnace man and copy his design for open-hearth furnaces. One quickly realizes the vast difference between the two sets of conditions. The open-hearth furnace, for instance, is provided with a very expensive and carefully prepared floor, sintered in place by gas heat; all heating and cooling of the furnace is done very gradually, and its life is mostly spent at a high and uniform temperature; while in a boiler furnace the rates of heating and cooling are very rapid, and the areas involved are many times the areas of open-hearth furnace floors. Both of these conditions tend to produce growth, as explained by the author.

The Illinois coals produce an ash which has been the most difficult to handle. From it considerable quantities of metallic iron and iron sulphide (FeS) are produced from the pyrites, and these do not mix with the slag, but due to their gravity they settle into the lower part of the furnace, filling the cracks and producing expansion. If metallic iron and iron sulphide are permitted to accumulate continuously, the conductivity of the bottom is increased.

While the floor problem is somewhat different from side walls of refractory construction, yet they have much in common. The side-wall story has been solved by water-cooling in one form or another. It therefore seems that water-cooling, together with proper surfacing of magnesite, chrome, or other suitable refractory, will solve the furnace floor in a similar way, making a construction whereby all materials on the floor can be tapped off from time to time and not accumulate. Basically the trouble has been that too cheap a floor has been installed. The kind of floors on several of the first jobs indicated that this was ample, but later experience indicates that a little more expensive construction is required. It is not a problem to require more than a relatively short period of intensive study and trying out of different materials in practice under a sufficiently wide variety of conditions as to rating, kind of coal, and method of operation to determine the various factors and limitations for use in future engineering study and application.

H. J. Kerr.[5] The authors, Messrs. Irwin and Gilbert, should be thanked for their valuable papers and are to be congratulated upon the success of the two plants which they represent.

[5] Chief Engineer, Service Department, Babcock & Wilcox Company, New York, N. Y. Mem. A.S.M.E.

Approximately five years have elapsed since the starting up of the first 1200-lb. unit at Edgar Station, Boston, and these two papers refer to two additional high-pressure plants, making in all six stations now operating successfully above 1200 lb. pressure. In addition there are two industrial plants.

While some difficulties have been experienced with the higher pressure, it can be fairly stated that the boiler rooms of such stations have developed no problems for which the solution was not readily available.

As a result, units are being built and plans are being made for many additional installations to operate above 1200 lb. pressure. In general, the tendency is to design the boilers for approximately 1400 lb., with the turbines operating at about 1250 to 1300 lb. Some consideration is being given to somewhat higher pressures, and it is as yet too soon to conclude that steam pressures will not extend to a pressure of the order of, say, 2500 lb. This question also involves the limit of superheat temperature which will be commercially permissible.

The use of high pressures has necessitated changing the designs of parts ordinarily used for low-pressure plants. The future will no doubt bring about changes in many construction details which will simplify the present designs; to name one detail—pipe flanges. In going over a high-pressure station one is impressed with the amount of material and the cost of pipe flanges sufficiently strong to withstand these pressures. A material simplification would be possible with a satisfactorily welded joint.

In general, high-pressure designs should tend to the elimination of all possible joints. In this connection, central power plants in this country have developed feedwater conditions to an extent where it is almost possible, and some day it may be possible, to eliminate all joints.

Both papers call attention to a problem which has developed in more recent years with the increase in pressure and temperature; namely, the deposit on turbine blading.

With low pressures and low steam temperatures, it was customary to discuss steam conditions in terms of 1 or $1^1/_2$ per cent of moisture, and under which conditions, except in the cases of very difficult waters, quite satisfactory results were obtained from the turbine. In later years considerable effort has been expended in obtaining the driest possible steam from boilers, and we today discuss steam conditions of the order of $^1/_4$ of 1 per cent of moisture, or in terms of four or five parts per million, or perhaps in terms of the electrical resistance of the condensate.

With boilers discharging steam of this purity, deposits on turbine blading are taking place, the annoyance depending largely on the nature of the material which deposits. Of these materials, the one which has caused the most difficulty is silica. In some instances the deposited material analyzed as high as 95 per cent of silica, so that the only practical way for its removal seems to be the actual scraping of the turbine blades.

In other cases where the material has been a combination of sodium hydrate and other salts, its removal from the turbine periodically by either washing the turbine or by dropping the steam temperature to saturation has proved practical.

It should be borne in mind that this deposit on turbine blading is taking place with a steam which is purer than distilled water obtained in a drug store. It should also be borne in mind that passing through a 60,000-kw. machine we have something of the order of 600,000 lb. of steam an hour, so that for 1 p.p.m. we have 0.6 lb. of material passing through the turbine an hour, or 430 lb. a month. If, due to pressure-temperature changes, in passing through the turbine 10 per cent of this material deposits on the blading, it is self-evident that a loss in load will result.

The writer is not prepared to say that we have gone the absolute limit in securing clean and dry steam from boilers. We still have hopes for further improvement, but the evidence is such that designs of high-pressure–temperature plants should contemplate maintaining the best possible conditions of water, and a silica water should be avoided wherever possible.

W. S. Johnston.[6] These two papers by Messrs. Gilbert and Irwin are invaluable to the membership of the Society and the public-utility industry of the country.

Both stations were designed and developed at about the same time after an exhaustive analysis of the economic possibilities by two engineering organizations working independently of each other. That they both came to substantially the same conclusion is indicative of their thoroughness in, first, sensing the fundamental factors the problem presented and, second, the economic solution thereof.

Both authors are to be commended for their frank statements regarding the operating troubles experienced in starting the equipment in service and the changes necessary to overcome them. It is to be hoped the manufacturers involved will profit by this practical experience when called upon to supply equipment for similar duty. After reading these papers and in the light of experiences at other 1400-lb. stations, one can safely assume that high-pressure steam is practical wherever the economics of the situation justifies its use. It is not too much to expect that the authors may publish in later papers more operating experiences and performance data when the stations have been in longer service.

The application of pulverized fuel is worthy of note. The Holland Station uses the storage system, while at Deepwater both the storage and direct-fired systems are employed. Mr. Irwin reports some operating troubles with the direct-fired system, but it is thought these can be overcome. Coal hanging up above the mills can be prevented by making the slope of the raw-coal bunker walls and coal chutes steeper to permit gravity movement of wet coal. The feeders might be changed from the star to the drag type. While the suggestion of incorporating a small storage bin of pulverized fuel between the mills and burners may be worth while at Deepwater, yet it is believed that after the initial period of operation has passed and the operators have become thoroughly familiar with the service characteristics of the equipment, it will be found that the direct-fired system affords the same degree of reliability now secured from the storage system.

A combination of the regenerative and reheat cycles has been incorporated in both plants. There can be no question that the regenerative cycle is a permanent feature that will be retained in station design for many years. On the other hand, the reheat cycle is felt to be transitory. It will not be necessary to resort to it when we can use higher initial steam temperatures. With 1350 lb. pressure at the turbine throttle, a steam temperature of 1000 deg. fahr. will not give greater moisture in the exhaust end than is permissible with present-day machines. At least one manufacturer will supply superheaters for this temperature. Turbine builders will now countenance the use of steam at 825 deg. fahr. with the proper arrangement of engine cylinders. Before long they will design for yet higher temperatures. Steel makers must prepare to furnish materials for piping and valves that will complement the superheaters and turbines. Then it will be possible to save on fixed charges by eliminating the investment in reheater equipment. At that time it will be interesting to compare extensions to the Holland and Deepwater Stations with the initial design.

Author's Closure

All of the high-pressure steam leaks which have occurred in the Deepwater Station have been on the small lines, with the

[6] Stone & Webster Engineering Corporation, Boston, Mass.

exception of one or two leaks which have developed on the boiler-feed lines where they connect to equipment and where the joints are not welded. The following gasket practice has been followed:

Boiler manholes—corrugated Monel metal.
Rehater manholes—folded asbestos.
Boiler, water-wall, and superheater handholes—steel-covered asbestos gasket. On this particular type of gasket to date only one failure has occurred, and that was on a handhole plate on one superheater. The gaskets on the water columns are corrugated Monel metal. Copper, soft-steel, and sheet packing have been tried, but with little success, and the Monel metal is at present giving good service.
Economizer return bends and flange connection on economizer supply pipes to the boiler drum—a stainless-steel gasket is being used in place of the original copper gaskets, and seems to be some improvement.

The stainless-steel gasket has one fault, in that if a leak does occur the fitting faces show more signs of being cut than they do when a copper gasket is used. This is probably due to the fact that the stainless steel resists wear more than the fitting faces.

As previously reported, the operating experience with the economizers may generally be stated to be satisfactory, although during the months of September, October, and November on six occasions it has been necessary to shut down one of the boilers in order to replace gaskets on the economizer U-bend. This is the same boiler on which difficulty was cited in the original report. Only three leaks have developed on the other five boilers during the same months.

No further difficulty has been experienced with slag-type furnace bottoms. No. 6 boiler, in which the magnesite bottom was installed the end of August, has been operating satisfactorily without undue expansion and without developing leaks. No. 4 boiler, which originally had a floor made up of cans of coke, but which have now entirely disappeared, has shown no change and is operating with the floor made up of slag. This floor caused the operators some concern for a week or two as the steel plates on the bottom of the floor became heated to a dull-red glow, but by raising the slag depth in the furnace the bottom was chilled and has been giving satisfactory service ever since. The measurements of the growth of this floor show that it has the least expansion. It is the intention not to make any changes in this floor unless some unforeseen difficulties arise.

The statement that iron sulphite was responsible for the using up of the coke in the bottoms of boilers 4 and 6 should be changed. This conclusion was reached due to the fact that the manufacturer advised that this trouble had occurred at other plants where no coke bottoms were installed. However, an analysis of the material cut out of the floor of No. 6 boiler shows that the material is similar to the slag which is tapped off in the normal course of operation.

One Distinct Advantage of Bin System. The last three months have borne out the earlier conclusion that there is a distinct advantage in reliability to be obtained from a bin system of fuel firing over the unit system. During the past three months, on seven occasions, it has been necessary to reduce the load on the plant from 5000 to 12,000 kw. due to difficulty with the mills used on the direct system. This has at times been due to breakage of mill parts because of wear, and may be overcome after the operators are better able to judge a proper maintenance and overhaul schedule. Even with prearranged periodic maintenance it is of course necessary to reduce the output unless it is possible to synchronize mill and boiler outage.

In the last few weeks a Pittsburgh seam coal, which it was possible to purchase at a relatively low price, has been burned in small quantities for test purposes, and it was found that using this much harder coal on the direct-fired mills takes considerably more power and reduces output and necessitates overloading the mill motors in order to keep the boilers up to capacity. It is a condition which could not persist over a long period of time, as there is danger of burning the motors from continuous overload, and it is difficult to keep the boilers up to rating. Using the same coal on the bin system, the mill capacity is also seriously decreased, but by operating the mills longer hours it is possible to maintain the boilers at any desired rating. This feature of flexible boiler capacity with varying qualities of coal should be taken into consideration when deciding the relative merits of the two systems.

Other difficulties with the mills have been due to operating conditions, such as plugging of the feeding system to the mill or the plugging of coal in the mills themselves. There have been outages of the mills on the bin system, but it has never been necessary to reduce plant load on the bin-system boilers on account of lack of mill capacity.

Refractory Erosion Most Serious Problem. Difficulty with erosion of the refractory material which backs up the bare-tube water walls on the back wall of the furnaces opposite the burner has increased and is now the most serious trouble being encountered in the boiler room. The operators are experimenting on the bin-system end of the plant by varying the secondary and primary air pressures in the hopes of shortening the flames to the extent where they do not throw the slag from the coal against the rear walls and wash them down, and still keeping the flame travel sufficiently long to prevent the flames from carrying this slag up into the lower row of boiler tubes and slagging these over. It is felt that with careful adjustment it may be possible to find the point where a balance between these two extremes will leave a satisfactory condition.

On the direct-fired end the manufacturers are experimenting with deflector blocks to be put into the burners, in order to get better control of the flame travel. On a rear wall now being rebuilt a 70 per cent alumina brick is being used in place of straight refractory, in the hope that this will reduce the washing action of the slag. If the present experiments are not successful, it may be necessary to resort to an all-metal wall, by using either fin tubes or blocks on the present bare tubes.

Silica Deposit on Turbine Blades Reduced. There is a misstatement in the original report in that it was assumed that other plants were having similar difficulty with silica deposit on the blading. We were led astray by an article in *Power Plant Engineering* (February 15, 1928), written by John Anderson, in which he stated that the analysis of the deposit found on the high-pressure turbine blading on the 1200-lb. machine at Lakeside, Milwaukee, contained 56 per cent silica, and that by injecting 10,000 lb. of water per hour into 15,000 lb. of steam per hour entering the turbine throttle when the machine was rolling at 400 r.p.m., they were able to wash off this scale so that the machine could be restored to its full-load capacity.

TABLE 1

	Net output, 1000 kw-hr.	B.t.u. per kw-hr.
April	19,975	16,752
May	35,373	14,036
June	44,707	13,828
July	43,169	14,114
August	63,775	13,341
September	63,842	12,749
October	66,989	12,419
November 1 to 15	35,520	12,244

Deepwater Economy Increased to 12,244 B.t.u. per Kw-hr. It should be borne in mind in evaluating the advantages of 1200 lb. against these particular results that a load was carried for 24 hours of the day during the past 3½ months, which necessitated

operating all turbines and five of the six boilers at their maximum capacities, which is a point beyond the maximum efficiency point on the turbines. See Table 1.

The scale on the blading at Deepwater plant has been analyzed on several occasions and varies between 75 and 95 per cent silica. An attempt to wash this off apparently had no effect, as it did not increase the machine capacity. However, when the machine was taken down for inspection, no scale was found on the blading, but the last stage diaphragm had a thin coating which was sufficient to increase the pressure drop across this diaphragm to the extent of reducing the turbine capacity.

Since the installation of purifiers on the vapor outlets of the house evaporators, the quantity of silica in the water is so small that it has been practically impossible to separate it or even to find it on laboratory analysis. It is believed, however, that some of this must be carried into the pure-water system from leaks in large-size evaporators installed for the du Pont system. The tubes in these evaporators are being rerolled. During the period when evaporators are cracked the pressure conditions which exist in the shell would tend to force raw water into the condensed-water side of the system. A 1-in. steam line has also been installed on this turbine to take steam from the first stage into the fourteenth stage, in an attempt to increase the temperature in the last stage of this machine nearer to the temperature existing in the last stages of the high-pressure elements of the compound machine. It would appear that the higher exhaust temperature on these machines must prevent the scale deposit.

One of the low-pressure turbines has developed vibration early in operation when the steam was carried at the contractual temperature. Until such time as it is possible to take this machine down it is necessary to operate it with steam at 700 deg. in place of 750 deg., which of course reduces the efficiency of this one unit. It is expected that when the machine is overhauled and a more economic loading is imposed on the plant, station economy will be improved by 400 to 500 B.t.u. per kw-hr. This condition of maintaining five boilers at their maximum loading continuously has also brought to the fore more forcibly than in other locations the necessity for continuous availability of the fuel-burning equipment and has showed reduction of load due to short-time outages of the direct-fired mills. Normally, this condition of course would not be introduced, as other boilers in the plant would be able to take the small deficiencies.

FSP-53-21

Comparative Resistance of Refractories to Coal-Ash Slags

By R. K. HURSH,[1] URBANA, ILL.

The results contained herein were obtained in an investigation conducted by the Engineering Experiment Station of the University of Illinois. Tests have been made in the laboratory slagging-test furnace of various commercial refractories and of a series of special mixtures representing clays from three manufacturing districts, as well as differences in the method of fabrication, the firing treatment, and the fineness of grinding of the materials. The slags used were ashes from coals of five different districts. The comparative resistance of refractories varied with the character of the coal ash. Dry-pressed and stiff mud bricks were more resistant than hand-made refractories. A moderate degree of firing treatment was preferable to hard burning when the brick were subjected to high furnace temperatures and slag action.

THE slagging of refractories by coal ash in boiler furnaces is due to a combination of corrosion and erosion. The molten slag attacks the refractory at the contact surface, taking some of the material into solution. More or less penetration may occur, with the result that the refractory surface subject to corrosive action is greatly increased. The piling up of the liquid slag on the refractory produces a flow due to gravity, which may be increased by the velocity of the furnace gases. The flowing away of the slag removes the dissolved material and maintains an unsaturated slag deposit on the refractory surface. At the same time particles of undissolved refractory, loosened by solution of the bonding material, are washed away with the slag. This latter action is found with refractories containing coarse grog particles, which are usually more resistant to slag action than the bonding material of the brick. It is especially noticeable with bricks which are appreciably penetrated by the fluid slag.

Effect of Slag Composition

The comparative severity of attack by the slag is dependent on the composition, temperature, and viscosity of the slag, as well as the amount deposited on the refractory. Slags high in fluxes, such as lime, magnesia, alkalies, and iron, generally have low fusion temperatures and are more fluid at the temperatures of furnace operation than are slags with higher contents of alumina and silica. The composition of these low-fusing slags is such that their reactivity with refractory compositions is also greater than that of more refractory ashes. Both factors, high flux content, and high fluidity, result in greater corrosive action at given temperatures, and the degree of attack increases markedly with temperature increase.

Effect of Furnace-Gas Composition

The composition of the furnace gases with respect to reducing or oxidizing conditions is considered as an important factor in the action of ferruginous slags. Under reducing conditions the iron will be in the ferrous form, which has a much greater corrosive action than ferric iron. However, the operating temperatures of boiler furnaces are generally high enough to produce a considerable amount of ferrous iron or magnetic oxide regardless of atmospheric conditions. Results in the laboratory slagging tests have shown little effect of differing degrees of reduction, as indicated by the tests with slag from coal used at the Cahokia station (see Table 1).

TABLE 1 EFFECT OF FURNACE-GAS COMPOSITION ON SLAGGING ACTION

Test No.	Aver. temp., °F.	Aver., % Co	Time, hr.	Slag used, lb.	Average depth of erosion, in cm. — Brick brand						
					21	18	16	19	14	17	27
36...	2900	6.2	24	170	0.75	0.78	0.72	0.71	0.76	0.55	0.63
37...	2900	2.5	24	165	0.83	0.90	0.72	0.75	0.83	0.70	0.72

Effect of Temperature Gradient in Refractory

The temperature gradient in the refractory may be a considerable factor in the slag action. If the furnace wall is well insulated so as to maintain a high temperature in the refractory somewhat back from the surface, the slag in contact with the brick will be kept fluid, and more active corrosion takes place. There is the further possibility that the high temperature will cause more or less fusion and glass formation in the refractory back of the slag interface which will have a lower resistance to slag action than the unfused material. In some cases, however, crystallization of mullite from this glass in the body might conceivably increase the resistance to slag action.

When there is a high temperature gradient through the refractory, heat is taken away from the surface fast enough to somewhat cool the slag at the contact and thus increase its viscosity. Its activity is thus reduced, and its tendency to penetrate the refractory is much decreased. This may be a considerable factor in the resistance of silicon-carbide refractories because their high thermal conductivity reduces surface temperatures and slag penetration is slight, even though the body itself may be quite porous.

Refractories Factors

The corrosion by slags will vary with refractories of different chemical or mineralogical constitution. Where the body is composed of different clays or materials, the slag attack may be especially marked on one of the constituents, resulting in rapid attack and washing away of the more resistant particles.

---

[1] Department of Ceramic Engineering, University of Illinois. Professor Hursh was graduated from the University of Illinois in 1908 with the B.S. degree in Mechanical Engineering. He spent a year as a graduate student in Ceramic Engineering and was for two years in the Clay Products Division of the U. S. Bureau of Standards. Since 1911 he has been a member of the staff of the Department of Ceramic Engineering. He is a member of Sigma Xi and was for several years Treasurer of the American Ceramic Society. Professor Hursh has been principally interested in the lines of structural clay products and refractories. He has contributed various papers in the publications of the American Ceramic Society, the National Brick Manufacturers Association, and the University of Illinois Engineering Experiment Station.

Contributed by the Special Research Committee on Boiler Furnace Refractories and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

TABLE 2 COMPARISON OF ACTION OF DIFFERENT COAL-ASH SLAGS AT THE SAME TEMPERATURE

Description of brick	Brand of brick	Comparative depth of erosion in 24 hr. at 2900 deg. fahr., ± 5 per cent — Kind of slag — East Peoria	Chicago	Cahokia	Cleveland	Lowellville[a]
SM	21	0.88	0.94	1.01	1.15	.....
SM	13	0.88	0.95	..	1.18	.....
DP	18	1.00	1.00	1.00	1.00	.....
DP	16	1.26 (?)	1.00	0.89	0.97	.....
DP	19	1.09	1.03	0.85	0.95	.....
DP	14	0.99	1.00	..	1.00	.....
High Al_2O_3	17	1.18	0.92	0.72	0.65	.....
HM	30	1.31	..	..	..	.....
DP	25	..	..	..	1.33	.....
DP	27	..	..	0.78	..	.....
DP	29	..	1.03	..	..	.....

[a] No erosion.

Inferior bonding materials in clay refractories are thus likely to materially decrease the life in boiler furnaces.

Bodies of low refractoriness do not withstand service conditions well because of the combined fusion effect in the body due to high temperature and the corrosion by the slag. Materials of high fusion or softening point may show equally poor service because they remain porous enough at furnace temperatures to permit penetration by the slag to much greater depth than do some of lower refractoriness. This is quite noticeable in the case of some of the high-alumina bricks.

Many of the fireclay refractories develop a vitreous, porcelain-like layer back of the slag contact and become so impervious that there is little, if any, penetration. Such materials offer a minimum surface for slag attack, and hence good service.

The physical structure of the body is an important factor in slagging effects. Coarse-grained bodies which permit penetration of the slag are severely attacked. Cracks, fissures, and laminations afford weaknesses in resistance to slag action. In some bodies the grog particles shrink excessively and become loosened from the bonding material. These particles are washed away readily as corrosion of the bond takes place, and the coarse grog is therefore of little effect in resistance to the slag.

The physical structure is also reflected in the method of fabrication. The dense skin-effect of stiff-mud molding would seem to offer an advantage in resistance to erosion, but this may be offset by lamination. Dry-pressed bricks are likely to be dense, providing the material is not too coarse. Maximum density should be a highly desirable factor in slag resistance. Hand-molded refractories are likely to be coarse grained and of minimum density, and therefore less resistant than either stiff-mud or dry-pressed bricks.

Laboratory Slagging Test

A laboratory slagging test has been described[2] in which the test bricks form the lining of a rotating furnace into which powdered slag is fed at a uniform rate through the gas-burner. The molten slag flows down the surface of the brick, producing the combined corrosion and erosion effects, while temperature and atmospheric conditions are controlled. Uniform treatment of each test brick is obtained and comparison of slag action is made by determining the volume loss per unit area exposed, or the average depth of erosion.

A series of commercial brands of brick, including those in use during service tests by the U. S. Bureau of Mines at the several boiler plants, have been made with different coal-ash slags. In part of these tests the same temperature has been used for each of the slags. In a further series the temperatures for the respective slags were approximately the maximums found in service in the plants.

[2] R. K. Hursh, "A Laboratory Slagging Test for Boiler-Furnace Refractories," A.S.M.E. Trans., 1929, paper FSP-51-44.

TABLE 3 EFFECT OF TEMPERATURE ON ACTION OF EAST PEORIA SLAG

Test No.	Temp., °F.	Average depth of erosion, in cm. ± 5 per cent — Brick brand — SM 21	DP 18	HM 30
21	2900	1.21	1.48	1.96
42	2600	0.87	0.94	1.15
41	2400	0.57	0.64	0.75

Comparison of Refractories With Different Slags

In Table 2 the comparative effect of the different slags, based on a 24-hr. test at 2900 deg. fahr., is shown for several brands of bricks. The figures represent the relative amount of depth of erosion, using that of brand 18 as a basis of comparison. In each case the data represent an average of two separate tests, two bricks of a brand being used in each test except in one or two cases where there was a limited supply. The ash from the East Peoria coal shows a distinctly different degree of activity on many of the bricks from that of other slags. This is an ash of much higher lime content than the others, but of somewhat lower iron content than the Chicago and Cleveland slags. It has the lowest fusion or P.C.E. value of the group of slags. It is of interest to note that the stiff-mud bricks, representing materials from two different districts, showed a somewhat better resistance than that of dry-pressed bricks, while the hand-molded refractory was much the poorest. The high-alumina brick showed inferior resistance to that of standard grades of fireclay. This is possibly due to the action of lime on the alumina to produce compounds and mixtures of lower fusibility than those of higher silica content.

The Chicago slag ranks next to East Peoria in lime content and has a greater iron content. With this slag, the high-alumina brick showed a slightly better resistance than the fireclay brick. The stiff-mud bricks show a slight advantage over the dry-pressed refractories. There is, however, little actual advantage of any brand over the others with this slag.

The Cahokia and Cleveland slags have about the same lime content, but the Cleveland ash is the highest in iron content of the group. The resistance of the high-alumina brick was appreciably better with these slags than of brand 18, indicating that such refractories might show considerably better service with slags of low-lime and high-iron contents than do the standard fireclay bricks. It is to be noted, however, that the physical characteristics of the brick may, and probably will, have an increased significance in this case.

Effect of Temperature on Slag Action

Marked increase in slag action is the result of increased furnace temperatures with the more fusible slags, as shown by the data in Table 3. The three brands of fireclay brick, made by different processes and at different plants, show a fairly consistent relationship between loss and furnace temperature. This is better indicated by the curves of Fig. 1, in which the slopes differ somewhat for the respective brick, but the comparative resistances are about the same at the three temperatures.

Comparison of Special Mixtures

In order to compare the effects of various manufacturing processes and materials on the resistance of firebrick to different coal-ash slags, eight lots of brick were made up by a single manufacturer. These represented the following variations in processing:

Brand	Material	Fabrication	Grind	Firing treatment
St 1	Dist. 1	Dry-pressed	Fine	Cone 6
St 2	Dist. 1	Dry-pressed	Fine	Cone 12
St 3	Dist. 1	Dry-pressed	Fine	Cone 14
St 4	Dist. 1	Dry-pressed	Very fine	Cone 12
St 5	Dist. 1	Stiff-mud	Fine	Cone 12
St 6	Dist. 1	Hand-made	Fine	Cone 12
St 7	Dist. 2	Dry-pressed	Fine	Cone 12
St 8	Dist. 3	Dry-pressed	Fine	Cone 12

These were tested in the laboratory slagging furnaces with slag from five different coals sent from plants at which tests were made by the U. S. Bureau of Mines. (Tests with a sixth coal are yet to be made.) The tests were arranged to make direct comparison of method of fabrication, grind, firing treatment, and materials from the three producing districts. The tests with the several slags were conducted at temperatures comparable with those found in the service tests at the respective plants. In the case of the East Peoria coal ash the test temperature was somewhat above the average found in the boiler furnaces, but approximated the maximum. This higher temperature was chosen in order to obtain a larger amount of erosion in the testing period and more satisfactorily comparable results. It has been indicated previously that the order of resistance of fireclay brick is not changed by the increase in temperature. The results of this series of tests are shown in Table 4. The values for erosion losses represent averages of determinations on three brick of each mixture used in a test. The porosity and bulk specific gravity values are averages on five brick of each lot as received.

TABLE 4 COMPARISON OF REFRACTORY MIXTURES WITH DIFFERENT SLAGS

(Based on average of three tests)

Brand	Description	Grind	Porosity, %	Bulk, specific gravity	Average depth of erosion in cm. — Kind of slag used and temperature in deg. fahr. — East Peoria 2600	Ca-hokia 2800	Chicago 2800	Lowellville 2600	Detroit 2800
Comparison of fabrication method									
St 2	DP	Fine	19.4	1.99	0.93	0.56	2.23	0	0.70
St 5	Sm	Fine	19.7	1.95	1.00	0.62	2.08	0	0.72
St 6	HM	Fine	21.8	1.81	1.18	0.65	2.41	0	0.93
Comparison of grind									
St 2	DP	Fine	19.4	1.99	1.04	0.49	1.63	0	0.27
St 4	DP	Very fine	21.0	1.96	1.05	0.50	1.66	0	0.30
Comparison of firing									
St 1	DP	Cone 6	20.5	2.03	0.93	0.67	2.27	0	0.44
St 2	DP	Cone 12	19.4	1.99	0.92	0.82	2.43	0	0.55
St 3	DP	Cone 14	18.1	2.01	0.95	0.74	2.33	0	0.56
Comparison of materials									
St 2	DP	Dist. 1	19.4	1.99	1.00	0.59	1.63	0	0.27
St 7	DP	Dist. 2	24.4	2.05	0.80	0.57	1.60	0	0.42
St 8	DP	Dist. 3	22.4	2.04	0.89	0.65	1.66	0	0.37

In the comparison of bricks made by the three molding processes the hand-made were distinctly inferior in resistance to all the slags. Dry-pressed brick showed the best resistance in all cases except that with the Chicago slag, but the advantage over stiff-mud brick appears to be slight in any case. The density of the dry-pressed and stiff-mud bricks is nearly the same, while that of the hand-molded bricks is lower. This may account, in part, for the differences shown in resistance to slag attack.

There is no appreciable difference in the action on bricks with different fineness of grinding in any of the tests. The difference in grind was not great, as the density values and the appearance of the bricks indicated.

In the comparison of firing treatment the brick fired at the lowest temperature showed less erosion in each test except that with East Peoria coal ash. The explanation for this was found in examination of the slagged brick. Slag penetration by the three slags used in tests at 2800 deg. fahr., was deeper in the cone 12 and cone 14 brick than in those fired at cone 6 but there was no noticeable difference in the case of East Peoria slag at 2600 deg. fahr. Apparently the higher firing temperature has developed a greater amount of glass in the body with possibly a higher thermal conductivity. At a furnace temperature of 2800 deg. fahr. the slag diffuses into this glass more readily than into the lower fired body and corrosion is therefore increased. A higher thermal conductivity, which is indicated by the greater depth of vitrified body back of the slag contact in the higher-fired brick, would result in higher temperature and more fluid glass in the body. This contributes still further to diffusion of the slag into the body and to corrosive effect.

In the cone 6 brick the development of glass in the body has not been sufficient to aid diffusion of slag into the body. Consequently, a thin vitreous layer is formed back of the slag contact which offers greater resistance to slag penetration. A greater temperature gradient, hence a smaller depth of high-temperature conditions, is evident in the appearance of the brick.

In the case of the East Peoria tests, the temperature, 2600 deg. fahr., is not high enough to soften the glassy portion of

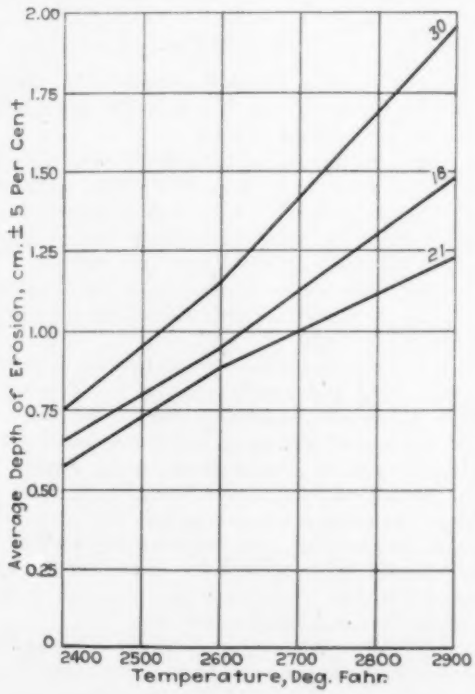


FIG. 1 EFFECT OF FURNACE TEMPERATURE ON SLAG ACTION
(The numbers at end of curves indicate brand of brick.)

the body to any appreciable degree. Hence, no noticeable difference is seen in the erosion, nor was any evident in examination of the several brick tested.

It is evident that high firing temperature is not desirable for these fireclay brick when subjected to furnace temperatures at which the glass in the body softens appreciably. The results might also be interpreted as indicating that the proper firing treatment for a refractory for boiler-furnace use is below that at which glass is formed in the body. The temperatures encountered in service will produce a vitreous coating on the brick to prevent appreciable slag penetration, and the softer burned body provides an insulation to limit the depth at which temperatures will develop.

The order of resistance of brick made in the three districts which are represented in these mixtures is somewhat variable with the different slags. (It is to be observed that these brick represent specific clays from the given districts and may not be at all representative of the region from which they come. It would be decidedly unfair to generalize in comparing the materials from various clay-producing districts on the basis of a single mixture from each; hence, the district is referred to by number only.) The brick from district 2 show a distinct advantage in the East Peoria tests and, possibly, are slightly

better with Cahokia and Chicago slags, but inferior with Detroit ash. In the latter case there is quite a marked difference in the resistance of the three mixtures. The amount of erosion by this slag is, however, small, and is less than that of the others. The relative differences between the mixtures might be lessened with a greater amount of erosion. The conclusion may be made from these data that various clays or mixtures do offer different degrees of resistance to different slags or under different furnace-operating conditions. A choice of fireclay refractories may therefore be made on the basis of specific performance with a certain coal ash and set of furnace conditions. A brick which gave good service in one place might be definitely inferior with a different coal ash.

Summary

The effect of varying reducing conditions in the furnace on the corrosive action of slag from Cahokia coal ash is slight in the tests in the laboratory furnace.

The temperature gradient in the refractory is an important factor in slagging action in two ways. With refractories of very high thermal conductivity the rate of heat loss through the body is sufficient to maintain a viscous slag at or near the contact, thus reducing slag penetration and corrosion. In the case of fireclay brick the conduction is not sufficient to have this effect, but materials of lower conductivity show somewhat better resistance to slags.

Materials of low refractoriness or fired at too high temperatures have reduced resistance to slag action, due apparently to diffusion of slag into the glassy portion of the body.

The development of a vitreous and impervious layer of the body, without excess of glass, next to the slag contact reduces corrosion. Bodies of very high refractoriness, which do not so vitrify at furnace temperatures, permit slag penetration and are more affected by slags than less refractory bodies.

A high-alumina brick showed less resistance to high-lime slags than did fireclay brick, but a greater degree of resistance to slags of low-lime and high-iron contents.

Increase of furnace temperature produces a marked increase in the erosion of refractories by coal ash. The order of resistance of fireclay brick was not changed with temperature variation.

Tests of special mixtures indicated an advantage for dry-pressed and stiff-mud bricks over hand-made, somewhat proportionate to their comparative densities.

High firing temperature tended to increase slag action of one fireclay mixture when subjected to slag action at a furnace temperature of 2800 deg. fahr. At 2600 deg. fahr. no difference due to firing treatment was observed.

A fireclay brick may afford better comparative resistance to one slag or set of operating conditions than to another; a selection may therefore be based on specific performance with respect to a given slag.

[Note: Discussion of this paper, jointly with the paper by T. A. Klinefelter and E. P. Rexford, by Ralph A. Sherman and by Harold E. Simpson will be found following the Klinefelter and Rexford paper.]

Author's Closure

In general, the results of the two methods of testing the action of the coal-ash slags on the refractories are similar. The Bureau of Standards method indicates the temperature range within which erosion losses are moderate and above which the attack of the slag becomes severe. The Illinois test indicates the relative activity of the slags on refractories under conditions simulative of those in service, as well as the comparative resistance of different refractories to a given slag under these conditions. With the slags shown by Messrs. Klinefelter and Rexford to have a low critical range, the slagging losses were generally high. The Lowellville ash, with a high critical range, produced little or no erosion in tests at 2900 deg. fahr. Some discrepancy appears in comparison of East Peoria and Cahokia slags, for which the critical ranges of temperature were found to be about the same. The results in Table 3 show greater losses due to the East Peoria slag at 2600 deg. fahr. than with the Cahokia slag at 2800 deg. fahr. From this it would seem that the temperature-viscosity relationship of a slag above its critical range is extremely important. The viscosity of the East Peoria slag decreases much more rapidly with increase in temperature than does that of the Cahokia ash. Hence its activity at temperatures above the critical range is much greater. The Chicago ash, which shows a high critical range, produced the most severe erosion in the whole series of tests at 2800 deg. fahr., considerably greater than that of the Cahokia ash, which showed a lower critical temperature range.

It is a question whether the average operating temperature in the boiler furnace is as important in relation to slag action as is the maximum temperature. The marked increase in erosion with higher temperatures indicates that the action during short periods above the average may cause greater losses than in long periods at the usual furnace temperatures.

In answer to Mr. Sherman's question concerning the differences in erosion losses on the same brand of brick in different tests, shown in Table 4, there were some differences in the temperatures and amounts of slag used in this series. No especial effort was made to duplicate the conditions exactly in the several tests with a given ash, since in each one all the brick brands were included which were to be directly compared. By correction of the figures for these variables the erosion losses of the St-2 brick in the separate tests with a given slag are practically equalized. Previous results have shown that tests can be satisfactorily duplicated, when desired, without difficulty.

Referring to the question of Dr. Simpson concerning relative slag penetration in brick fired at different temperatures, it was shown that less penetration and lower erosion loss of cone 6 brick was evident only in the test made at 2600 deg. fahr. A considerably greater amount of glass was apparent throughout the body of brick fired at cones 12 and 14, as well as deeper slag penetration. No slag was evident beyond the thin vitrified surface layer of the cone 6 brick. The heat treatment apparently was not sufficient to develop a glass of the amount or fluidity that would permit appreciable diffusion of the slag. In tests at 2800 deg. fahr., the slag penetration and the erosion losses were the same for all three firing treatments.

FSP-53-22

Action of Slags on Firebrick and Boiler Furnace Settings[1]

By T. A. KLINEFELTER[2] and E. P. REXFORD,[3] COLUMBUS, OHIO

The results contained in this paper were obtained in an investigation conducted by the Columbus branch of the U. S. Bureau of Standards for the A.S.M.E. Special Research Committee on Boiler Furnace Refractories. The paper presents a rational discussion of the action of coal ash upon boiler-furnace refractory walls and shows the consistency of results from the authors' laboratory slag tests with firebrick under actual service conditions in boiler furnaces.

WITHIN the last year the laboratory examinations made in connection with the boiler-furnace slags have developed what seems to be a practical test for predicting the action of slags on firebrick. The preliminary work leading up to this phase of the investigation will be found in previous reports.[4,5]



T. A. Klinefelter

In the latter report a description was given of the methods and the general procedure for the study of the principal components found in slags. This study remains the most fundamental on the program outlined, but one or two interesting developments have appeared from the examination of the field samples first sent in, resulting in a test which seems to have considerable practical application.

The work on the multiple component system brought out the fact that to obtain a true picture of the phases and conditions present at any given temperature it is necessary to quench the slag from that temperature. This led to a complete re-examination of all samples sent in by the Bureau of Mines field crew.

As a result of these examinations there was presented a rather different picture. Quenchings were first made from about the average operating temperatures to which the boiler settings had been subjected. It was noticed that, in those cases where the refractories had been giving fair or good service, the slag appeared to be saturated with a solid phase or phases, and in those cases where poor service had been reported the slag consisted mostly of glass with only a small amount of mullite present, particularly along the interface with the refractory.

Development of Working Hypothesis

On correlating these observations it was noted that in every case the slag and refractory interaction could be separated into three fairly distinct stages or ranges, as follows:



E. P. Rexford

Lower Range. At the relatively lower temperatures very little penetration into the refractory was apparent, the amount of glassy matrix was small, and the whole slag was saturated with the solid phase which very evidently originated from the components of the slag solution itself. These solids in most cases were skeleton magnetite and mullite, but fayalite, anorthite, akermanite, etc., were also found, depending on the composition of the coal ash.

Critical Range. As the temperature was gradually raised the following changes were noted: The skeleton magnetite dissolved first, and also the quantity of glassy matrix increased and became darker, thus apparently indicating solution of the magnetite into the liquid glass. At the same time the quantity of mullite or other solid phase decreased and the grains became smaller and finer, due probably to solution. Penetration of the glass or liquid into the refractory was more apparent, while along the edges of the refractory and in the glassy veinlets impregnating it fine needles of mullite appeared, very evidently growing from a combination of the refractory material and the coal ash, thus showing the start of severe corrosion.

This condition continued as the temperature increased until an interval or range was reached where the glassy matrix or liquid seemed to be at a maximum, the solid phases at a minimum, and the penetration into the brick was not great. For convenience this has been termed the "critical range," as at temperatures above this range severe corrosion appeared. This interval was not particularly sharp in any case, although in one or two cases an interval of 25 deg. fahr. showed a much more decided change than usual.

Upper Range. At temperatures above the "critical range" an increase in the solid phase again appeared, showing most abundantly along the interface with the refractory. In the cases examined, all of which were clay firebrick, this solid phase was mullite. Solid phases other than mullite would be expected with refractories of essentially different composition. The penetration into the brick became more and more pronounced, and the penetrating veinlets became a mass of unoriented mullite needles.

In general, then, it appears that, if the temperature and conditions in the boiler furnace are such that the slag solution is saturated with the solid phase recrystallizing from its own

---

[1] Publication approved by the Director of the Bureau of Standards of the U. S. Department of Commerce.

[2] Superintendent at Columbus Branch, National Bureau of Standards. Mr. Klinefelter received B.A. and B.S. in Ceramic Engineering from the Ohio State University. His industrial connections have been with Westinghouse Elec. & Mfg. Co., Atlantic Terra Cotta Co., Trenton Fireclay and Porcelain Co., and Gladding, McBean & Co.

[3] Research Associate for A.S.M.E. at Columbus Branch, Bureau of Standards. Mr. Rexford received B.S. and M.S. from the Massachusetts Institute of Technology. His industrial connections have been with General Electric Co. and Bethlehem Steel Co.

[4] S. J. McDowell, "A Laboratory Study of Slag Erosion," *Mechanical Engineering*, vol. 49, no. 5, 1927.

[5] T. A. Klinefelter and E. P. Rexford, "A Study of Crystalline Compounds Formed in Slags on Boiler Furnace Refractories," A.S.M.E. Trans., 1929, paper FSP-51-45.

Contributed by the Special Research Committee on Boiler Furnace Refractories and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

constituents, that slag is relatively less corrosive than a slag containing little or no solid phase. In the latter case the molten unsaturated solution present is capable of more or less continuous corrosion of the refractories, if there is not enough material in the slag to form a saturated solution of that solid phase required by the particular section of the multiple component system in which the solution lies for that temperature. Then in order to bring about a saturation the necessary constituents are absorbed from the brick, causing intense corrosion and the ultimate breakdown of the setting. Equilibrium is never reached, of course, as new slag is being constantly supplied by the gas currents. At low temperatures some of the dissolved material will be crystallized out as a solid if the melt has been saturated at a higher temperature. On heating, this dissolves, and if more is placed in the solution it, too, will dissolve within limits as the temperature is increased. In the case of the boiler setting, the slag, which is more or less viscous, is plastered against the refractory wall. If the temperature and composition of the slag are such that the liquid is capable of rapid corrosion, it flows or drips away. In doing so, whole particles of brick, especially the more refractory portions such as the grog, are floated away bodily. This condition was noted in the preliminary examinations. No doubt the so-called "washing" effect of the furnace gases aids in this process to greater or lesser extent, but only after the refractory has been softened by excess temperature or fluxing action of the slag.

It is entirely possible that, in the case of certain brick where the binding matrix is less refractory than the grog or solid particles, the boiler temperature and conditions may be such that softening and even liquefaction of the brick matrix occurs, thus causing failure without any reaction between the brick and slag. However, the evidence so far does not point to this as a factor of much importance in the case of good grades of brick at prevailing conditions. But this may be a factor which, coupled with the thermochemical action of the slag, causes failure with even the best of refractories. There has not been sufficient time to investigate this angle.

The investigation upon which the preceding discussion is based has been carried out under neutral to oxidizing conditions. While the work so far seems to check up quite well with the results shown in the commercial settings, it is felt that some of the findings will be modified for settings where the atmospheric conditions are strongly reducing. Commercial settings are of course known to be much harder on refractories when the furnace is operated under such conditions.

Method of Test

Assuming as a working hypothesis that, to obtain a relatively low corrosive slag, conditions should be such that the slag solution is well saturated at all times with one or more solid phases formed from constituents of only the slag solution, it becomes necessary to work out the three ranges mentioned for any given coal ash and refractory. With the "critical range" established, it is only necessary to keep the temperature of the boiler furnace below the "critical range" to obtain as good practical service from that refractory and slag combination as is possible from the thermochemical standpoint.

In order to work out these ranges readily, the following method and technique are suggested. As referred to, it has been found entirely impossible to study these slag and refractory combinations properly unless they are quenched from temperatures at which slag action took place in actual service. This requires the use of a control board for maintaining accurate and constant temperatures over any desired range. The furnace should be so constructed that the sample may be quickly quenched. This can be done readily by having the furnace core or heating chamber in a vertical position with a removable plug at the bottom.

By fusing the fine platinum wire by which the specimen is suspended it may be dropped into a mercury bath held close under the furnace.

A refractory to be investigated has a $^3/_8$-in. hole drilled near a corner to a depth of about $^1/_2$ in. A section including the drilled hole is now sawed or broken off and roughly shaped into a crucible about 1 in. in diameter and 1 in. in depth. This work requires only a few moments, as the drilling can be done with a corundum or periclase clip which works rapidly and cuts cleanly. The final shaping is done with a rough silicon-carbide stone.

This miniature crucible is filled with sintered coal ash and coal, or a piece of slag from the boiler setting under investigation, and then suspended in the furnace with a thermocouple next to it. Two samples can be handled almost as readily as one, the hot junction of the thermocouple being located between the two. This gives a temperature control of more than necessary accuracy, since it has been found in the work so far that in practically all cases there is no necessity for obtaining quenchings at intervals closer than 25 deg. fahr. In rare instances a closer interval in the so-called "critical range" has given a little better knowledge of conditions.

The furnace is heated rapidly to the desired temperature and is held there for from 4 to 24 hours, and the sample is then quenched. In general, the longer heat treatment gives crystals which are larger and much more easily identified than those produced in the short treatment, the proportionate amounts of phases present running practically the same.

Discussion of Preliminary Results

This hypothesis will be applied in the study of Table 1. It will be noted in this table that in every case where the service of the refractory is reported as "good" the average operating temperature is below that of the "critical range," and conversely where the service is "poor" the average operating temperature is above the critical range.

The average operating temperatures were arrived at by a study of the furnace and refractory temperatures observed at

TABLE 1 FIELD AND LABORATORY DATA OF BOILER FURNACES INVESTIGATED

Station	Equipment	Fuel	Average temperature for average rating, deg. fahr.	Critical range, deg. fahr.	Service of refractory in boiler setting	Crystalline compounds present in slag
Lowellville, Ohio	Underfeed stoker	Bituminous coal	2000–2300	2700–2750	Very good	Mullite
Trenton Channel, Detroit, Mich.	Long flame burner	Powdered bituminous coal	2600–2700	2700–2800	Good	Mullite, magnetite
Westport, Baltimore, Md.	Underfeed stoker	Bituminous coal	2400–2600	2650–2700	Good	Mullite, magnetite
East Peoria, Ill.	Long flame burner	Powdered bituminous coal	2200–2300	2200–2300	Fair	Anorthite, magnetite fayalite
Fisk Street, Chicago, Ill.	Traveling-grate stoker	Bituminous coal	2600–2700	2600–2700	Fair	Mullite, magnetite
70th St., Cleveland, Ohio	Forced-draft traveling-grate	Bituminous coal	2600–2700	2700–2800	Fair	Mullite, magnetite
	Natural draft		2300–2500		Good	
Nanticoke, Pa.	Traveling grate	Anthracite	2500–2600	2200–2300	Poor	Mullite, magnetite
Crawford Avenue, Chicago, Ill.	Chain grate	Bituminous coal	2800–2900	2500–2550	Poor	Mullite
Cahokia, Ill.	Long flame burner	Powdered bituminous coal	2600–2700	2200–2250	Very poor	Mullite, magnetite

the time the samples were collected.[6] Temperatures were taken within the bricks themselves at varying distances from the hot face, and extrapolations were made to obtain the temperature of the hot face. At the same time temperatures were taken within the furnace itself at varying distances from the hot face. As the latter temperatures, particularly those 2 ft. from the side walls, were somewhat higher than those obtained by the gradient established from the temperatures within the brick, they have been used. The readings taken under average boiler ratings, generally 150 to 175 per cent, were shown rather than the extremes. While the maximum temperatures and conditions affect the thermochemical reactions more or less, it is thought that average operating conditions are the ones of greatest value.

The installation reported as giving probably the best refractory service was that of Lowellville, with low-sulphur coal. The laboratory study of the fuel ash plus the refractory indicated a critical range of 2700 to 2750 deg. fahr., while the average operating temperature was from 2000 to 2300 deg. fahr. This suggested a prediction of good service from the refractories, and the reports stated the average life at this station to be about two years.

In case the average operating temperature is just below or falls within the critical range, one would expect only fair service from the refractories, depending upon certain conditions. If, for instance, the boiler were operated at or somewhat below average rating, the refractories would probably do quite well, while if operated at higher ratings considerable portions of the time, the operating temperatures thus exceeding the critical range, the refractories would fail much more quickly. Both the East Peoria Station and the Fisk Street Station of Chicago fall in this class. Fair results were reported from the field in both cases.

On the other hand, the cases of Cahokia and Crawford Avenue Station, Chicago, showed very poor service from the refractories as compared to the cases cited. The Cahokia critical ranges showed 2250 deg. fahr. as the upper limit, and the average operating temperature was from 2600 to 2700 deg. fahr., which is well within the "upper range" of the coal-ash refractory combination. Hence, severe corrosion was to be expected, and the field reports showed this to have been true. The Crawford Avenue Station was a similar case. It reported an average life of about 36 days for its refractories in positions receiving the maximum punishment, and the laboratory tests show this combination as having a critical range 300 to 400 deg. fahr. below the average operating temperature.

The laboratory studies show that in some cases the boiler temperature could be increased without serious detriment to the refractories in use, while in other cases considerable improvement might be expected if other combinations of coal ash and refractory were used.

Some investigating was done along this line in the case of Cahokia and Trenton Channel coal-ash refractory combinations. The Cahokia refractory was tried with the Trenton Channel coal ash and resulted in a decided raising of the critical range. Likewise, the Cahokia coal ash in combination with the Trenton Channel refractory also resulted in an increase, although not to the same extent. In this case the coal ash was a more deciding factor than the refractory.

For any given boiler furnace certain coals and refractories are available or economical to use. If a study can be made of the various ashes with the various refractories along the lines indicated, one or more combinations can probably be found where the critical range is above the average operating temperature. It is thought that by using the method outlined one will be able to obtain results with an accuracy and speed hitherto unobtainable.

[6] "A Study of Refractories Service Conditions in Boiler Furnaces," Ralph A. Sherman, Bulletin of the Bureau of Mines.

Summary

The following working hypothesis has been developed for predicting the action of slags on firebrick: To obtain a relatively low corrosive slag, conditions should be such that the slag solution is well saturated at all times with solid phases formed from constituents of the slag only. This involves working out three ranges or conditions for any given combination of coal ash and refractory, as follows:

1 The lower temperature range, where the slag is saturated with the solid phase originating from its own components and is considered to be a safe working range for the combination of coal ash and refractory.

2 A critical temperature range, where most of the crystalline phases from the slag components have come into solution and the slag is mostly liquid. The temperature should be kept below this range, as immediately above it severe corrosion of the refractory begins.

3 The upper temperature range, where the temperature is high enough to permit the brick components to unite with those of the slag to cause supersaturation and recrystallization of a solid phase. In this range severe refractory corrosion begins, increasing rapidly with further increase in temperature.

A technique is suggested for determining these ranges by the use of a quenching furnace and control board, giving on the whole a precise and accurate simulative test method.

Discussion

Ralph A. Sherman.[7] Manufacturers and users of refractories have long sought an accurate test by which the resistance of refractories to slags could be measured. The authors have presented in these two papers two methods which are widely different, but which have been developed concurrently and have had the advantage of comparison one with another and with rather carefully conducted field tests. With the completion of the trials of the two methods on the special series of firebricks which the Special Research Committee of the Society has had manufactured and the trials of panels of these bricks in actual boiler furnaces a definite evaluation of the relative merits of the two methods can be made. Mr. Hursh has reported the results of his method on all of the bricks with all but one of the slags to be used, and Battelle Memorial Institute is now engaged in a year's work which it is contributing to the problem in studying the series of combinations by the quenching method of the Bureau of Standards; Dr. Harold E. Simpson is conducting the investigation.

Both types of test may be retained as each has its particular advantages. The quenching method is undoubtedly the more fundamental of the two and gives an accurate picture of what occurs in the action of a slag on a refractory. The use of the control board allows much closer regulation of temperature than can be obtained in the rotating-furnace method. The cost of the equipment for the quenching method is greater than for the rotating furnace, and a more highly trained operator is required to conduct the tests and make the observations. A particular advantage of the rotating-furnace method is that the results are quantitative and a curve of the relation of the rate of erosion to the temperature can be plotted.

Although some idea of the increase in the rate of attack with temperature can be obtained by the quenching method, it has not yet been possible to plot a curve of the results. Another advantage of the rotating-furnace method is that different

[7] Fuel Engineer, Battelle Memorial Institute, Columbus, Ohio.

furnace atmospheres may be obtained with some degree of facility. Although it does not seem impossible to subject the specimens in the quenching furnace to different atmospheres and it is intended to attempt this phase of the work, it is probable that certain difficulties will arise, particularly in connection with the attack of reducing gases on the platinum winding of the furnace.

The authors of the paper on the quenching method have emphasized the fact that the "critical range" cannot always be determined within narrow limits, and the writer wishes to place further emphasis on this point. With one of the slags which has been studied at Battelle Institute the range was determined within limits of 25 deg. fahr., but with another slag the range could not be defined more closely than 200 deg. fahr. This is not at all serious, as without special studies the temperatures at various points in a boiler furnace at various ratings cannot be controlled more closely than 200 deg. fahr.

In closing, the writer wishes to ask Professor Hursh to clarify one point in connection with the results shown in Table 4 of his paper. The refractory ST-2 was used as a comparative specimen in the study of the four factors of burn, grind, method of fabrication, and district, but the erosion reported for this refractory for the same ash is not always the same. For example, the average depth of erosion with the Detroit ash under the comparison of the effect of the method of fabrication was 0.70 cm., while for the same refractory and the same temperature under the comparison of the effect of grind it was 0.27 cm. It is presumed that these were different tests and the length of run and amount of slag used were different, but without explanation it would appear that the results could not be closely duplicated.

HAROLD E. SIMPSON.[8] The development of a satisfactory service test for boiler-furnace refractories is one of vital interest to both user and manufacturer of such products. Such refractories are not only called upon to withstand high temperatures, but also they must withstand the corrosive action of the coal-ash slag.

The two tests given in these papers, while developed along different lines, have as their common goal a means for observing the effect of coal-ash slags on refractories. The one method inherently lends itself to careful control and precision, giving the fundamental relations existing between coal-ash slags and refractories, while the other lends itself to use in the average size laboratory and gives a comparative estimate of slag attack on refractory.

In Professor Hursh's paper, the writer would like to see the results of tests carried out at a series of temperatures, say, from 2200 to 2800 deg. fahr. at 100 deg. fahr. intervals for a particular coal ash and refractory. He would also like to ask why the author believes slag penetration to be greater in a vitrified body. If the development of glass is conducive to slag penetration in the high-fired refractories (cone 12 and 14), why does the development of thin vitreous coating in cone 6 body hinder slag penetration?

In the paper by Messrs. Klinefelter and Rexford, the writer would like to add that it has been his experience that rather severe corrosion may occur even if the slag still contains considerable solid phase. As a rule, however, the presence of a great amount of glass in the slag is an indication of considerable attack of refractory by slag.

E. G. BAILEY.[9] It would seem that the conclusions to be drawn from the paper by Messrs. Klinefelter and Rexford are about as follows:

1 Select coal with as high a fusing temperature of ash as possible
2 Select refractories to produce a high "critical range"
3 Keep the furnace temperature below this "critical range" by means of low rate of combustion, high excess air, or other means, most of which are inefficient.

In this day of high steam pressures and temperatures and high electric voltages, it seems rather reactionary to call for low furnace temperatures.

If low furnace temperatures are required, water-cooling of the furnace walls is the best way to do it, and the question might be asked, "Why use refractories anyhow?"

While water-cooling is undoubtedly here to stay, the writer still can see some good in refractories and is very much interested in studies such as those brought out in this paper. High furnace temperatures are necessary in many cases; they are desirable in all cases for the most complete combustion at high rates of liberation. At low rating a cold furnace leads to inefficiency, smoke, and oftentimes absolute failure to maintain ignition.

The essential point of this paper is the "critical range" between a given refractory and a given ash. This points toward a refractory-lined furnace, with the advantage of high temperatures, yet cooling behind this refractory that keeps the refractory and slag mixture at its surface below the "critical range" by means of the water-cooling and the high temperature gradient from the furnace to the water tubes. Further studies along the line of this paper are well worth while in this field.

With slag-tap furnaces and relatively hot refractory walls and perhaps additional refractory in other places, the recovery of ash in the furnace and its economical elimination in the early stages of gas flow are all desirable goals which can only be brought about by further knowledge of the ash-refractory fluxes and the use of a proper refractory throughout the furnace.

P. NICHOLLS.[10] In the investigations on the life and the failures of refractories it has often been disappointing that one could not make assured explanations for what occurs; also the usefulness of laboratory determinations of the characteristics of refractories has seemed to have such a limited application. For these reasons, in the committee work, the writer has been quite enthusiastic about the explanatory theory advanced in the paper by Messrs. Klinefelter and Rexford. It is the first time that an attempt has been made to connect tendency to slag with a fundamental measurement, and to generalize from it.

One must expect, however, that, by itself, this new indicator will not be a measure of the relative wastage of furnace walls. Even assuming that all imposed conditions are determinable, yet at least the viscosity of the resultant slag will be another fundamental, as will also the relationship of viscosity to temperature. The writer believes that measures of the tendency to slag will have to include these. In addition, the prediction for probable actual slagging will have to allow for the quantitative measures of the imposed conditions such as quantity of deposited ash and gas velocity and composition.

The interpretation of the work of the authors is therefore that their indicator gives a relationship between "tendency to corrode" and temperature up to their critical temperature; that the "tendency of the rate of corrosion" against temperature will have to include the additional factors of the viscosity of the liquid and some measure involving the texture of the solid refractory; and that the "rate of corrosion" will be compounded of the latter (what may be termed the fundamental) constant and the imposed conditions.

[8] Ceramic Engineer, Battelle Memorial Institute, Columbus, Ohio.
[9] President, Fuller Lehigh Company, Fullerton, Pa. Mem. A.S.M.E.
[10] Supervising Fuel Engineer, Pittsburgh Experiment Station, U. S. Bureau of Mines.

The line of study being conducted by the authors is, however, well worth continuing, and it is to be hoped that they will not be content to leave it in its present state, as is so often the necessity, and also perhaps the tendency, of investigators, but that the study will be continued to the stage where it can be embodied in a definite testing specification or other form available for practical application, instead of being carried no further than a special investigation.

Authors' Closure

In the paper there are mentioned a number of points on which considerable work is necessary before some of the criticisms brought out in this discussion may be answered more fully.

The three conclusions which Mr. Bailey drew are correct so far as the first two are concerned, but in the case of the third the assumption drawn is decidedly incorrect. It would follow of course, if the life of the refractory were the sole factor to be considered. This is not the case, and we certainly had no intention of giving the impression that other factors in boiler operation should be subordinated to that of life of the refractories.

The laboratory was given the task of analyzing the causes and processes involved in the failure of the refractories. This is what we have endeavored to do in the paper, and since some refractory and coal-ash combinations give better service than others, we have gone a step further and suggested a method of finding the best combination for an installation. There is not the faintest inclination on the part of the authors to advocate any changes in any installation which will lessen the final efficiency of the whole station operation. It is all a matter of how important a factor the life of the refractory may be in the cost accounts of any given installation. It may or may not be worth while to lengthen the periods between repairs. To do so might mean such an increase in the cost of coal alone sufficient to offset the benefits gained, or as Mr. Bailey suggested, such low temperatures with resulting inefficiency as not to be worth while. It is a matter of simple arithmetic for the individual station to be worked out by the cost accountant.

There are cases noted in Table 1 accompanying the paper where there is no doubt that a different combination of refractory and coal ash would lengthen the life of the refractory; but whether it would actually pay that station to change or not can be answered only by the station's bookkeeper, not by us. It is our personal opinion in the matter in general that there exist a good many stations which could use another coal-refractory combination to advantage, and also we believe that a precise method of determining varying degrees of corrosion with various combinations will enable a power station to write specifications for both their coal and refractories in a much more intelligent fashion than heretofore.

In regard to Dr. Simpson's finding that rather severe corrosion may take place even while containing large amounts of glass and of solid phase, we should like to caution that one must never lose sight of the fact that all these slags are corrosive in all ranges, but relatively less so in the lower and the critical than in the upper. Also that the amount of corrosion in a given range for any given combination may be less or greater than for some other combinations.

In the same way the length of the critical range varies considerably, as Mr. Sherman points out, and while we have not sufficient data as yet to correlate any property with it, we believe this will develop. We suggest that it possibly ties in directly with the viscosity of the slag, and may be something of an index of relative corrosion. In the studies mentioned in the main paper it was noticed that the most violently corrosive slags had a shorter critical range than those of less corrosion. It is a phase which calls for much further study, particularly with regard to the factor of viscosity, as Mr. Nicholls points out.

FSP-53-23

High-Pressure Chemical System for Boiler-Water Conditioning

By A. A. MARKSON,[1] NEW YORK, N. Y.

In this paper the author describes the high-pressure phosphate system for boiler-water conditioning as used at the Kips Bay Station of the New York Steam Corporation. He believes that it solves the problem of economizer deposition, and while the primary object is to avoid deposition of scale in economizers and feed lines, it is apparent that it gives a flexibility of control superior to that obtained by adding the chemical directly to the feedwater heaters or hot wells. The paper does not discuss the chemical phases of the problem.



THE PURPOSE of any system of boiler-water conditioning is to maintain such conditions in the boiler water that the production of steam will not cause the formation of deposits on the heating surfaces. If in the accomplishment of the primary object, secondary difficulties develop, these latter must be eliminated without interfering with the accomplishment of the main objective. At Kips Bay Station of the New York Steam Corporation the phosphate system of boiler-water conditioning was found to be extremely satisfactory in the elimination of deposit from the heating surfaces, but in its original application secondary difficulties either existed or arose, which had to be traced down and eliminated. These difficulties were as follows: (1) The boilers produced wet steam, (2) heavy deposits formed in the economizers, and (3) it was not possible to maintain low alkalinities in the boiler water while maintaining economizer-corrosion protection.

The problem of wet steam at Kips Bay, and its solution, has been fully discussed in a paper by A. R. Mumford, "A Study of Moisture at High Rates of Evaporation," published in Vol. 51, No. 22, of the A.S.M.E. Transactions. Study of this problem developed the following facts:

a That there was a critical value of the concentration of boiler caustic alkalinity, above which it was practically impossible to accomplish the mechanical removal of entrained moisture, and below which mechanical purification was entirely practicable.

b That the concentration of other dissolved substances and suspended matters had a negligible influence on the quality of steam delivered from the boilers, within the concentration ranges covered in the experiments.

The caustic concentration was kept below the critical value by increasing the blowdown, until its control by chemical means could be effected. The sources of the caustic were two: That added to inhibit the economizer corrosion, and that formed by the breakdown of the carbonates in the boiler water. Some caustic alkalinity is necessary in the boiler water to insure the maintenance of conditions called for by the phosphate system for the formation of non-adherent precipitates. Obviously the economizers would be subjected to accelerated corrosion by the elimination of the caustic added to the feedwater. Furthermore, the presence of carbonates in the make-up insured a constantly increasing concentration of caustic soda in the boiler water.

Because of the fact that no deposits formed in the economizers before the phosphate system was adopted it was rather obvious that the adding of conditioning chemical directly to the boiler would eliminate completely the zones unfavorable to conditioning by the phosphate system. By using a less alkaline sodium phosphate than was previously employed advantage could also be taken of the possibilities of keeping the boiler alkalinity under control by partial neutralization in the boiler, rather than by excessive blowdown.

The author has noted considerable interest on the subject of economizer and feed-line scale deposition, as evidenced by several recent contributions on the subject, both of an experimental and a speculative nature, to the Society's committee on feedwater treatment.

One theory advanced seems to hold that the product formed when phosphate is added to the feedwater is in a supersaturated condition until it hits the heat-absorbing surface, at which point equilibrium is suddenly induced by the sharp change of temperature. The product carried along in a state of supersaturation then precipitates. Another hypothesis developed to explain the formation of deposits where none should occur, is that there is a film effect at the heating surfaces in which film precipitation occurs. Based on the experiences with economizers and feed lines at both stations A and Kips Bay, a theory is advanced which differs from those mentioned.

It appears that there is much to be learned as to the exact physico-chemical nature of substances in solution. The economizers scaled up very heavily, and the deposits were not limited to the tubes, but seemed, if anything, to be worse in the headers and on the plugs. This point was enough to question the specific validity of the supersaturation and film theories for these conditions. It is the author's belief that the answer to the question of such depositions will be found in the existence of products which are more basic in nature than those which are conventionally supposed. These products, of which there may be several, may have the general formula:

$$(Ca)\ w\ (OH)\ x\ (PO_4)\ y\ (H_2O)\ z$$

instead of being straight calcium phosphates. Some of the experimental facts in support of this contention are weak, but they may be summarized as follows:

1 If the boiler-water environment is not kept at a causticity materially greater than that theoretically required to insure tricalcic phosphate, adherent deposits form.

2 If a sample of boiler water is allowed to stand without being

---

[1] Mechanical Engineer, New York Steam Corp. Jun. A.S.M.E. Graduated from Rensselaer Polytechnic Institute in 1924 and joined the New York Steam Corporation; at present is engaged in research and design work as a member of the Station Engineering Department.

Presented at a meeting of the Metropolitan Section, New York, N. Y., February 3, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

filtered and the clear liquid above the sludge is analyzed on successive days, the concentration of phosphate in the clear liquid will be found to increase, apparently due to the slow action of caustic soda on the calcium phosphate.

3 At the time the high-pressure system was first put into operation at Kips Bay, it so happened that on two boilers the internal feed lines terminated in open-ended pipes above the upcomers. It was noticed when these boilers were taken off the line for inspection 30 days after the system had been in operation, that there were well-defined circular patches under the feed-line discharge pipes, which were of the same composition as the material which was formerly deposited in the economizers.

The author believed at the time of the inspection that the cause of these patches was that the incoming feedwater had no chance to mix with the boiler water to more than a slight extent, thus resulting in a solution weak in phosphate and alkali being directly localized under the feed pipes. If this theory were true, it would follow, so it seemed, that securing a uniform distribution of feedwater in the drum would not merely spread the deposit over a greater area, but would actually cause such deposition to disappear. This proved to be true.

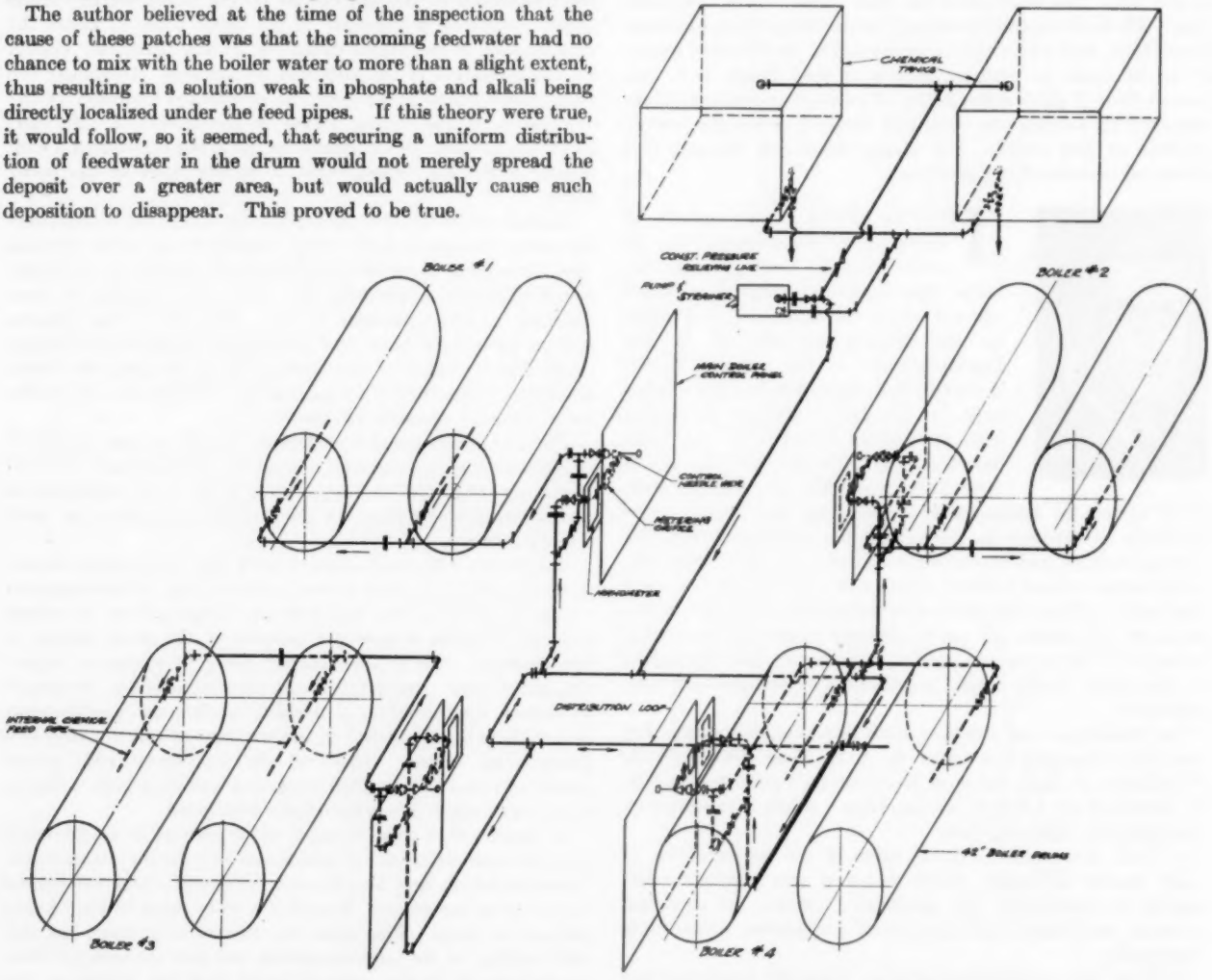


FIG. 1 SCHEMATIC LAYOUT OF HIGH-PRESSURE CHEMICAL SYSTEM AT KIPS BAY STATION

To recapitulate, the cause of economizer and feed-line scale deposition is essentially due to the maintenance of lower alkalinity in the feedwater than is required to prevent the formation of nonadherent precipitates. This required alkalinity is in excess of 100 parts per million of OH, fivefold higher than the maximum which could be economically added because of the resultant high blowdown required.

DEVELOPMENT OF THE HIGH-PRESSURE SYSTEM

In developing the system which was installed at Kips Bay, and later at Station A, certain mechanical conditions were set forth which must be met by the general scheme. In the first place, any such system ought to be simple to operate, to adjust, and to maintain. Secondly, the amount of special attendance necessary for the functioning of such a system ought to be a minimum. Third, the system ought to be under complete and accurate control at all times, with visible and positive indication of its functioning.

It is believed that the system which is to be described met these requirements very well. The general simplicity of the design is evidenced by the fact that both of the installed systems have shown no fundamental defects in 10 months of operation at Kips Bay Station and in five months at Station A.

The principle of the system may be thus briefly described: If a chemical charge of calculated strength be fed into the boilers in exact proportion to either the feedwater entering the boilers or the steam generated by the boilers, the optimum condition for internal treatment of boiler water will be attained. The assumption that the amount of treatment is proportional to the steam generated is probably true for most cases. The system accomplishes this in the following manner. A high-pressure pump maintains a constant head of chemical solution in the distribution loop. Feeds to individual boilers are taken off this loop. A metering orifice is placed in each boiler chemical line and the flow from the loop into the individual boiler is manually

controlled by means of a needle valve placed after the metering orifice. The metering orifice indicates the flow of chemical by means of a graduated manometer which is under the observation of the boiler operator. The manometer scale is graduated to correspond with the boiler-meter flow reading. The operator, therefore, has merely to adjust the control needle valve until the correct indication is obtained on the manometer, at which time treatment requirements will be satisfied.

The system is composed of the following component parts, shown schematically in Fig. 1: (*a*) Supply tank, (*b*) suction strainer, (*c*) high-pressure pump with constant-head bypass, (*d*) main feed loop, (*e*) metering orifices, (*f*) control needle valves, (*g*) indicating manometers, and (*h*) distribution pipe in boiler. The design of the metering and control system will be discussed first.

Design of the Orifice

In the design of the orifices for use in metering the chemical, the limiting factors are the available supply-tank capacity, the number of boilers to which it is desired to apply the system, and the question of probable accuracy obtainable in meteri g

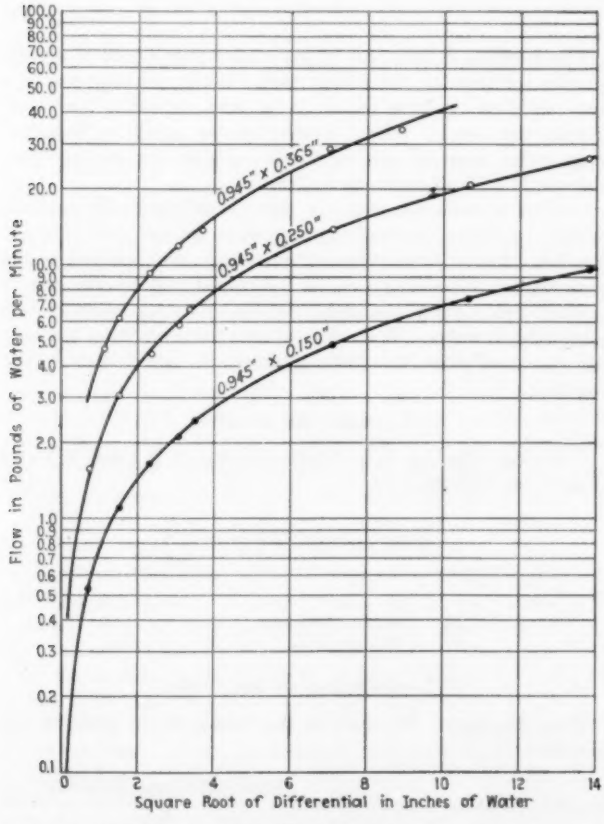


Fig. 2 Calibration Curves for Three Different Sizes of Orifices

small quantities. There is a limit of minimum flow below which metering difficulties, such as the use of orifice plates having microscopic holes, are apt to appear. Furthermore, since the total amount of solution fed into the system depends to a large extent on the number of boilers in the station, it follows that the chemical-solution tanks will have to be larger in capacity as the number of boilers goes up, in order that the tanks may not empty in too short a time. Where the number of boilers is relatively large, as in Station A, where there are now 14 boilers, a single large tank may be used, and a very small auxiliary tank employed to feed the system while the large tank is out of service for recharging. The tanks are equipped with air agitation so that outage is minimized.

At Kips Bay Station there were two 7000-gal. tanks available for holding the chemical charge. The maximum daily load on

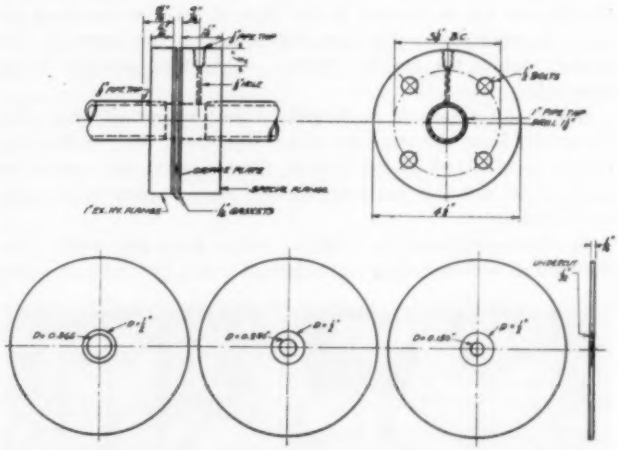


Fig. 3 Orifice Assembly and Three Orifice Plates Used

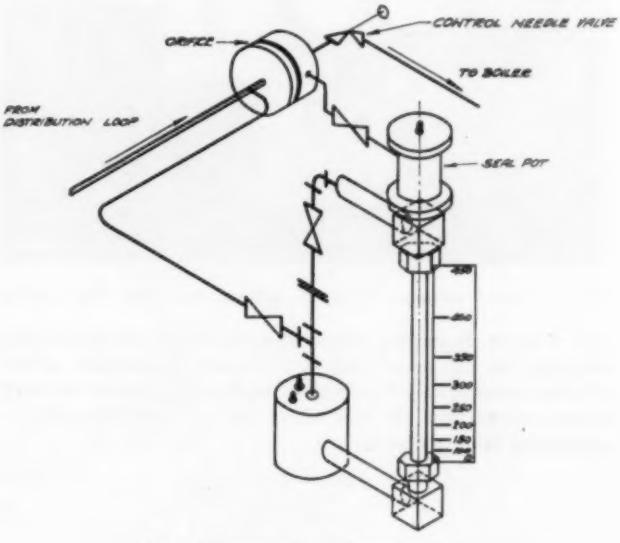


Fig. 4 Indicating Manometer

the station at that time was about 24,000,000 lb. Feeding from alternate tanks it was desirable that recharging intervals should not be shorter than 48 hr. at the maximum station load. This means that the average hourly feed would be 7000 divided by 48, or about 145 gal. per hr., which corresponds to an average hourly steam generation of 1,000,000 lb. per hr. It was therefore necessary to feed chemical at the rate of 2.4 gal. per min. for an hourly evaporative rate of 1,000,000 lb. As the maximum load per boiler is somewhat over 400,000 lb. per hr., the figure which was used in designing the metering system was 1 gal. per min. chemical flow when the steam-flow meter indicated 400,000 lb. per hr. generation.

Those familiar with orifice work will immediately appreciate the difficulties of obtaining close metering accuracy under these conditions. Nevertheless, Fig. 2 gives a calibration curve for three different sizes of orifices under a complete range of heads

If these orifices be used in the standard orifice carriers, their reproducibility will be well within 3 per cent.

Actual coefficients of discharge of these orifices were determined by numerous weighed tests on water, and the results of these tests may possibly be the subject of a contribution to the Fluid Meters Committee of the Society.

Since the calibration curve is based upon water of density 62.4 lb. per cu. ft. flowing in the pipe, it may be corrected for other densities by multiplying the values on the curve by the square root of the specific gravity of the fluid actually being metered.

If for any reason it is desired to use orifices differing very materially from the standard plates described, their coefficients should be obtained by calibration, since existent data on orifice coefficients and tap spacings are not very applicable to such small orifices.

In this installation the 0.365-in. orifice plate was used. The differential corresponding to a flow of 1 gal. per min. is about

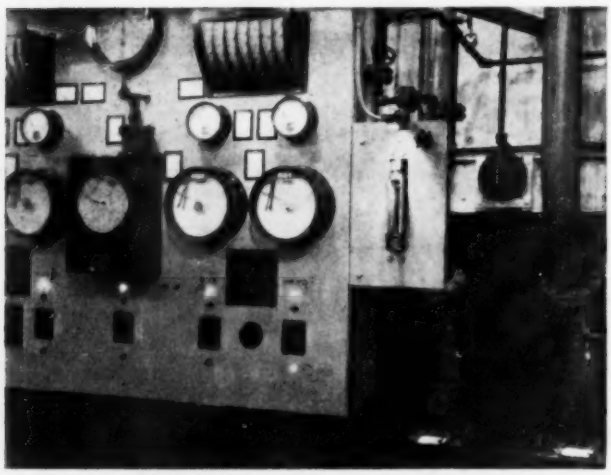


FIG. 5 THE CONTROL VALVE IS LOCATED AT THE MANOMETER

4 in. of water, depending somewhat on the density of the chemical solution. In the event that it be desired to compute orifices differing considerably from those employed by means of coefficients derived from the calibration curve, a useful formula for computing these plates is:

$$F = \frac{CD_1^2 Y^{1/2} H^{1/2}}{Q} \times 5.99$$

$$F = \left[\left(\frac{D_1}{D_2}\right)^4 - 1\right]^{1/2}$$

D_1 = Pipe diameter in inches
D_2 = Orifice hole diameter in inches
C = Coefficient of discharge
Y = Density of the solution, lb. per cu. ft.
H = Differential in inches of water of specific gravity 1.00
Q = Flow, lb. per min.

A curve giving values of $\frac{D_2}{D_1}$ against values of F makes a very convenient device for getting values quickly. In the formula the density of water at room temperature has been taken at 62.4 lb. per cu. ft.

For low values of $\frac{D_2}{D_1}$, F is sensibly equal to $\left(\frac{D_1}{D_2}\right)^2$ and the orifice-hole diameter is then given by

$$D_2^2 = \frac{Q}{5.99\, CY^{1/2} H^{1/2}}$$

THE INDICATING DEVICE

The design of the indicating device is shown in Fig. 4. It is nothing more than a high-pressure cistern manometer of somewhat special construction. Attention may be directed to the fact that it uses a $^3/_8$-in. standard boiler-gage glass and has a cup on the top combination which acts as a safety-seal pot.

GAGE COEFFICIENT

In a manometer using a heavy liquid as the indicating liquid, in contact with a lighter liquid as the medium of pressure transmission, the true differential, in terms of water of specific gravity 1.00, corresponding to the observed total deflection, is given by the observed deflection times $\gamma_1 - \gamma$, in which γ_1 = specific gravity of the gage liquid; γ = specific gravity of the lighter liquid.

If the gravity of the chemical being fed is sensibly equal to unity, this reduces to:

Differential in inches of water = deflection in inches $\times (\gamma_1 - 1)$ ..... [1]

The question is then that of selecting a gage liquid of specific gravity suitable for the orifice used. While mercury makes a very satisfactory gage liquid, it is very expensive and may require the use of orifices which are too small in diameter. Some other gage liquids which are suitable are carbon tetrachloride (1.58) and bromoform (2.9).

Carbon tetrachloride magnifies the true differential by approximately 1.7 times; bromoform reduces it by half, and mercury by 12.6. For orifices carbon tetrachloride was selected as the gage liquid. This was colored a brilliant red by the use of "pyloil" red, a dye (commonly used for coloring gasoline) which is insoluble in water. It is apparent that the necessity of knowing the gage coefficient accurately increases as lighter gage liquids are used.

CISTERN COEFFICIENT

The true deflection in a cistern manometer is given in terms of observed deflection by

$$\text{T.D.} = \text{O.D.} \times \left(1 + \frac{A_1}{A_2}\right) \text{.....} [2]$$

in which

A_1 = Gage glass area
A_2 = Cistern area

CONSTRUCTION OF THE SCALE

From Equations [1] and [2], the length of the scale can be obtained.

$$\text{Length of scale in actual inches} = \frac{\text{Max. differential in inches } H_2O}{\left(1 + \frac{A_1}{A_2}\right)(\gamma_1 - \gamma)} \text{.....} [3]$$

This gives a fixed length of scale corresponding to the fixed maximum rate of generation, the assumed differential, and the assumed amount of chemical flow. It is obvious that the intermediate scale values can be constructed geometrically or otherwise to take into account the variation of the discharge coefficient as well as the square root function of flow. The scale may be graduated, of course, in any other set of arbitrary units desired.

The Control Valve

As shown in Fig. 5, the control valve is placed at the manometer, so that the operator's duty consists of maintaining the manometer reading at the scale point corresponding to the reading of either his feedwater or steam meter in a manner analogous to controlling the air-flow pen on the boiler meter. The control valve should be placed after the orifice for obvious reasons. For the control valves it has been found that $^1/_4$-in. steel needle valves function perfectly for the conditions and have given no trouble since installation.

The Loop

In designing the loop a maximum velocity should be chosen which tends to make the loop act as a pressure reservoir. One or two feet per second is satisfactory. Welded joints are preferable, although there has been no trouble with screwed or flanged pipe in this service at the pressure of 400 lb. Check valves should be employed where necessary, and all valves and fittings ought to be of iron or steel and of the best quality obtainable.

The Pump

The pump, as might be expected, is the point in the system upon the satisfactory functioning of which practical operation depends. The first pump selected was a reciprocating triplex type driven by a 5-hp. motor. The functioning of this pump was quite satisfactory, although when it came to selecting a change-over pump, it was felt that this could be improved upon considerably in the matter of maintenance, which was a little more than would ordinarily be required because of the necessity for a steady flow curve.

A rotary pump was installed which gave some promise at first, but later developed so much trouble that the makers took it back. Final choice was a low-capacity high-head centrifugal pump driven by a small turbine. This appears to have all the ideal characteristics required for the work. The power consumption is about 25 hp., which is considerably in excess of the power requirement of a reciprocating pump. Since the exhaust steam may be used and since the amount of water treated by this system is so large, this power requirement is easily justified. Two manufacturers, at present, are prepared to supply this type of pump in heads up to 600 lb. from stock patterns.

Constant Pressure Relief

A spring-loaded relief valve is used to bypass the excess of fluid pumped back to the supply tanks and to maintain a fairly constant head on the discharge loop. The constancy of this head is very important, in order that the control needle valve will not require readjustment at constant rating. A suitable all-iron valve for the service could not be obtained, but it was found that a standard bronze valve, when chromium plated, was satisfactory. If a pump is selected of ample excess capacity, a durable needle valve may be used instead of a loaded relief valve.

In the operation of the system the pump pressure should be set at a point which will insure a negligible effect of variations of boiler pressure on the flow through the boiler-control needle valve. For example, if the boiler pressure varies 5 lb. from the usual operating point it will only affect the indication of the manometer very slightly when a drop of 100 lb. is induced across the control valve, but rather seriously if the drop induced across the control valve is, say, in the neighborhood of only 10 lb.

The suction strainer is a duplex strainer of our own making, having strainer baskets of 40-mesh monel cloth. The assembly is arranged so that a strainer can be cleaned in about 10 min. without interrupting the service.

Conclusion

In conclusion, the advantages of a high-pressure system such as the one described are:

1 It solves the question of economizer deposition due to the nature of the conditioning chemical used. The author has no available data on whether stage heaters show the same deposition, but it is obvious that the high-pressure system also solves this question.

2 While the primary object of the high-pressure system is to avoid deposition of scale in economizers and feed lines, it is apparent that it gives a flexibility of control which is far superior to adding the chemical directly to the feedwater heaters or hot wells.

3 The cost of installation of such a system is very moderate. Some limitations that occur to the writer are: (1) It is not as adaptable to the needs of a very small plant as it is to a very large plant, and (2) it requires technical supervision of a better order than is required by the other method.

This paper does not attempt to go into all the chemical phases of the problem. There is an abundance of literature on the subject of boiler-water conditioning by means of the phosphate system, which has been published by the various societies. The author wishes to express his indebtedness for the cooperation of both the operating and design deparments of the corporation which made this development possible.

Appendix No. 1

Cost figures show that a good estimate of the cost of installation of a complete system as described is about \$8000 + \$250 per boiler. This figure includes two high-pressure pumps and supply tanks and is derived from costs on four and twelve boilers.

Appendix No. 2

Average discharge coefficients for small orifices:

Upstream tap—one pipe diameter.
Downstream tap—0.8 pipe diameter.

Ratio D_2/D_1	Ave. C
0.16	0.66
0.26	0.66
0.39	0.62

Data obtained on 1-in. extra-heavy iron pipe of ordinary roughness.

Discussion

H. A. Jackson.[2] The paper presents the difficulties that arise from addition of sodium phosphate to boiler feed in order that proper boiler-water conditions may be obtained for the elimination of boiler scale. The author has concisely pointed out that to prevent feed-line and economizer corrosion the feedwater must have a definite alkalinity, and that as this alkalinity increases enormously in the boiler due to concentration, therein there is presented a dilemma, for its reduction by the employment of an acidic phosphate destroys the necessary alkalinity in the feed system. An answer lies in direct introduction of the acidic chemical into the boiler water.

The direct introduction of phosphate into the boilers has heretofore presented very serious difficulties, particularly as to proper proportioning and this the author has solved beautifully. The combination of a small-orifice chemical-solution meter, calibrated in terms of steam flow from boiler together with an

[2] Hall Laboratories, Inc.

adjusting needle valve so that the operator can keep the chemical-meter reading coincident with that of the boiler steam meter, is a smooth solution of the problem.

The proportioning of chemical to boiler feed imposes far less rigorous exactitude than, let us say, the proportioning of air flow to powdered-coal feed, for the momentary errors of the latter cause actual efficiency losses whereas the boiler water gives a chemical storage which can be momentarily drawn upon or temporarily increased within certain bounds without imposing the slightest detriment to the maintenance of the desired chemical equilibria in the boiler water. A complicated mechanical or electrical hookup of the boiler steam and chemical-flow meters to adjust automatically the control valve would be therefore entirely unnecessary.

It is perhaps not generally appreciated how really difficult it is accurately to control flows of a fraction of a gallon to one or two gallons per minute, and particularly so when the liquid whose flow it is desired to control is under pressure corresponding to those of modern boilers. The description of this pressure-feeding system and particularly the data on differential pressures through small orifices as shown by the curves should be of the utmost value to others who wish to adopt this method.

In regard to concentration of the various dissolved salts in reducing the size of steam bubbles in boiling water and in lengthening their time of rupture and hence the cause of foaming and wet steam, it is the writer's experience that the various sodium salts such as the sulphate, chloride, carbonate, and hydroxide have an effect in this regard all in the same order of magnitude, with sodium hydroxide heading the list, but not to such an extent that its concentration alone should be regarded to the exclusion of the others. The effect of suspended matter is also of the same order of magnitude and is not the predominating influence that it was formerly regarded.

With considerable dissolved material in the boiler water, the greatest change in boiling conditions is experienced in changing from a pH value of less than about 5 to one greater than 8 or 9. Above this value, the effect of increased causticity is not as great though definite, unless saponifiable matter is present in the boiler water.

The difference in foaming qualities between a boiler water having a definite caustic alkalinity and one having none is vast if there be saponifiable organic matter present. The higher caustic concentration accelerates saponification, hence supplies soap more rapidly, and likewise accentuates foaming tendencies greatly.

The author draws the conclusion that deposits similar to those formed in economizers would also be formed in stage heaters. This conclusion is correct, for that has been the writer's experience in a number of cases.

Calcium-phosphate deposits occur, however, not only in economizers and closed heaters, but also in check- and water-regulating valves and in the feed lines themselves. The author's theory of a calcium basic phosphate to account for their occurrence is very interesting.

It has been the writer's experience, however, from the results of a large number of analyses of such deposits, that the calcium found lines up with the phosphate of 2 Ca to 3 PO_4 and not in a variable relation. For the precipitation of a basic calcium phosphate, considerable hydroxide radical would necessarily be present in the water.

Even at relatively low pH values (7 for instance) the solid phase is tricalcic phosphate, but the solution phase under that condition is largely calcium hydrogen phosphate, since the tri-basic phosphate radical readily combines with the hydrogen of water to form the di-basic HPO_4 radical. As the pH value rises above 7, the tendency for formation o the di-basic radical decreases and therewith the apparent solubility of tricalcic phosphate also decreases. Thus the hydroxide alkalinity in the boiler water essential for clean surfaces may be under very pure boiler-water conditions, as low as that corresponding to a pH value between 10 and 11. If however, the boiler water results from 100 per cent make-up, and therefore contains considerable dissolved solids, and in particular if it contains silica, the hydroxide concentration must be high, corresponding perhaps to a pH value of 11.5. With the certain control that the author has developed for maintenance of phosphate in the boiler water, and if the hydroxide concentration is uniform throughout the boiler, the writer does not believe that maintenance of 100 p.p.m. hydroxide is essential for obtaining clean surfaces.

Now, the very property of tricalcium phosphate which makes it the most desirable form in which to precipitate entering calcium, namely its extremely low solubility and its precipitation as a sludge in a vessel in which its saturated solution is boiled, makes it an undesirable component to have in a feed system. That it precipitates as a sludge in the boiler is due to the fact that its solubility must decrease with increase in temperature and hence its saturated solution tends to form adherent scale on surfaces that are cooler than the contacting solution. This explains the deposits in feed lines, but not, however, those in heated portions such as economizers.

Its solubility is so slight that the addition of a phosphate solution to water containing more than a mere trace of calcium causes supersaturation to occur unless complete formation of the requisite amount of solid phase be instantaneous and this is not the case. Speed of formation of the solid phase of tri-calcium phosphate is greatly accelerated by elevation of temperature as may be easily observed by the simultaneous addition of sodium phosphate to dilute alkaline calcium solutions, one at ordinary and the other at boiling temperatures. Therefore the formation of economizer deposits is none other than the rapid increase in rate of desupersaturation by the sudden rise in temperature.

P. C. Fritz.[3] It is very evident that the adherent deposit occurring in the economizers, feed lines, and surfaces below the feed discharges in the boilers forms in an environment of comparatively low alkalinity, while the less adherent deposits and sludge occurring in the boilers forms in an environment of relatively higher alkalinity. Therefore, there is reason to believe that the material depositing in the economizers may be in the form of the calcium acid phosphate, while that depositing in the boilers is in the form of tri-calcium phosphate.

This is what may be expected when the equilibria is considered which is present in the solution of a phosphate salt.

$$PO_4^{\equiv} + H^+ \rightleftharpoons HPO_4^{=} + H^+ \rightleftharpoons H_2PO_4^-$$

An increase in the concentration of H ions shifts the equilibria toward the H_2PO_4 ions since an increase in H ions allows more PO_4 ions to combine to form HPO_4 ions and H_2PO_4 ions. An increase in OH ions means a corresponding decrease in H ions, since the product of the concentrations of H ions and OH ions equals a constant, the ionization constant of water. Consequently, a decrease in H ions causes the equilibria to shift toward the PO_4 ions, since the removal of H ions allows more H_2PO_4 ions and HPO_4 ions to dissociate, with a final result of more PO_4 ions. Therefore, it appears that the concentration of H_2PO_4 ions, HPO_4 ions, and PO_4 ions in the solution is somewhat dependent upon the concentration of the H ions, or inversely upon the concentration of the OH ions. As the solution is always above a pH of 7.0 it is probably better to speak of what happens in

[3] Assistant Chemist, New York Steam Corporation.

terms of OH ions. Therefore, we may conclude that a region of low alkalinity has more H_2PO_4 ions and less PO_4 ions than a region of high alkalinity.

Now when the calcium ions from the feedwater come in contact with these phosphate ions, it is likely that at places of low alkalinity more $Ca(H_2PO_4)_2$ and less $Ca_3(PO_4)_2$ will form than at places of high alkalinity. This is verified to a certain extent by examination of the analyses made on numerous samples of boiler and economizer deposits. These analyses show that the P_2O_5 content in the economizer and feed-line deposits is higher than that in the boiler deposits. However, the silica, carbonate, and sulphate contents in the economizer and boiler deposits are different, but even when corrections are made for these materials, the economizer deposits contain a higher P_2O_5 content. The P_2O_5 content of $Ca_3(PO_4)_2$ is 45.8 per cent, of $CaHPO_4$ is 52.2 per cent, and of $Ca(H_2PO_4)_2$ is 60.6 per cent. Therefore there is reason to believe that the deposit in the economizers, feed lines, etc., is in the direction of $Ca(H_2PO_4)_2$ and that in the boilers is in the direction of $Ca_3(PO_4)_2$.

This same line of reasoning can be applied to the problem of boiler water changing in OH concentration and PO_4 concentration upon standing. This seems to be entirely a problem of carbon-dioxide absorption. A series of tests was run on a sample exposed to the laboratory air. Another series of tests was run on a sample sealed tightly from the air. In order to guard completely against any CO_2 absorption between tests, each test was run on a new or separate sample, and the vessels were filled with the sample up to the cork so that there was no air above the water. A sample containing 220 p.p.m. of OH, 96 p.p.m. of CO_3, and less than 10 p.p.m. of free PO_4 was tested after standing unfiltered in a well-stoppered vessel for 120 hr. There was no change in the concentration of OH and CO_3. The phosphate increased slightly, but did not reach 10 p.p.m. A similar sample exposed to the air was tested after 24 hr. The OH concentration decreased to 45 p.p.m. and the CO_3 concentration increased to 410 p.p.m. Another test was made after 72 hr. standing. The OH concentration was found to be O and the CO_3 concentration increased slightly to 416 p.p.m. The phosphate increased, but still there was not 10 p.p.m. present.

There may be an explanation to this result in the fact that alkaline water is a good absorber of CO_2 which is present in considerable quantity in laboratory air. Therefore, OH ions are consumed with the production of CO_3 ions

$$2\ NaOH + CO_2 \rightleftharpoons Na_2CO_3 + H_2O$$

The concentration of OH ions is thus reduced and coincident with the decrease in OH ions is an increase in H_2PO_4 ions and a corresponding decrease in PO_4 ions as discussed previously. This means more $Ca_3(PO_4)_2$ solid must dissolve and dissociate until there are as many PO_4 ions present as required by the solubility product constant of $Ca_3(PO_4)_2$. Therefore, there is the same amount of PO_4 ions in solution and an increase in H_2PO_4 ions at the expense of solid material. The indicator tests all of the phosphate ions and calls them PO_4 ions. This may account for the slight increase in PO_4 on standing exposed to the air.

E. F. Walsh.[4] A system similar to that described was installed at the South Street Station of the Narragansett Electric Company in Providence, R. I., about six months ago and has proved very successful. The plant referred to consists of 400-lb. cross-drum boilers of 1800 hp. each. These boilers supply steam to a 35,000-kw. turbine and desuperheated steam to a lower-pressure and temperature plant. The water-heating cycle consists of a softener and stage bleeding from eighth, twelfth, and fourteenth stages. This plant employs jet condensers and requires 85 per cent make-up as raw water.

The original chemical treatment consisted of a hot-process system which was unsatisfactory and finally abandoned for the Hall system. Phosphate feed was first introduced into the softener and allowed to circulate through the entire water system to the boilers. It was found impossible to keep the high-pressure stage heaters clean. The maximum operating period was about three weeks and at the end of this period the deposits had increased to a point where it was impossible to maintain sufficient boiler-feed pressure. Feed-pump runners were found to contain heavy deposits and check valves did not function.

It was then decided to install high-pressure pumps and feed chemical directly to feed lines beyond the high-pressure heaters, thus bypassing all feedwater heaters. It was thought that the injection of chemical at that point would insure thorough distribution of chemical to all boilers and at the same time eliminate any major deposition in the lines. This, however, did not prove to be the case. In a short time it was impossible to open or close the feed valves, regulating valves failed to function, and the internal feed pipes plugged up, which required frequent cleaning.

The problem was finally solved by feeding chemical directly to each boiler with a system similar to that described in the present paper. Absolutely no trouble has been experienced since.

The writer agrees with the author in the statement that a critical value of the concentration of boiler caustic alkalinity exists. Much trouble with carry-over was experienced until it was discovered that there was a definite maximum point at which the caustic alkalinity could be carried for the production of clean steam. A slight increase above that point gave unlimited superheater trouble.

Much could be said about the control of boiler feedwater to produce clean boilers and clean steam. Every engineer contemplating such control should take advantage of the research work done in the past few years by those who have been forced to solve the issue. The writer wishes to compliment the author in bringing this important development of feedwater condition before the engineering profession. Until a chemical is produced containing the necessary available P_2O_5 and whose sodium salts are soluble, it will be necessary to use the high-pressure system if success is desired.

Author's Closure

Messrs. Jackson and Fritz are probably quite correct in their theories of economizer deposition. When Mr. Jackson states that sodium hydroxide should not be regarded as the primary variable in the foaming of boilers, he evidently refers to a body of data obtained in small-scale experiments on foaming properties.

Mr. Walsh's discussion is in confirmation of the author's statement with regard to hydroxide being the determinant in boiler foaming. Since the presentation of this paper, the author has received additional confirmation of the critical-value theory from sources of reliability. Therefore, he feels justified in strengthening his statement as to the critical function of hydroxide in boiler foaming.

Since the original presentation of this paper, the author has had additional experience with the type of system described. Beyond the completely satisfactory solution of the pump problem, there are few additional remarks to make. The flexibility of this method of feeding chemical has made available cheaper sources of treating chemical, since there is no longer need to purchase on anything but an active-ingredient basis. In addition, two considerable boiler installations have been tied into the systems described, at approximately the incremental cost figures quoted, thus materially reducing the investment per thousand pounds of steam.

[4] Test Engineer, The Narragansett Electric Co., Providence, R. I.

FSP-53-24

Furnace-Gas Compositions and Temperatures in Underfeed-Stoker-Fired Boilers, and Their Effect on Boiler Settings

By ALBERT C. PASINI[1] and EDWARD M. SARRAF[2]

This paper is a study of conditions encountered in boiler furnaces using underfeed stokers. It includes the development of methods of test procedure and their application in investigating furnace-gas compositions, and of determining the temperatures to which the refractories of the setting are subjected. It contains a discussion of the effects of variation of the fuel-bed thickness on the brick in the settings, the efficiency of combustion, and the resultant superheated-steam temperatures. The paper further shows the widely divergent furnace-gas compositions which may be obtained directly over the fire along the wall in underfeed-stoker-fired installations.

COMPARATIVE service tests were instituted at Connors Creek Power House, The Detroit Edison Company, Detroit, Mich., to investigate the reasons for a noticeable difference in refractories service in two boilers (presumably similar in design and operation), and to determine to what degree the variable rate of deterioration of the refractories might be due to conditions of design and operation.



ALBERT C. PASINI



EDWARD M. SARRAF

DESCRIPTION OF BOILERS AND SETTINGS

The boilers under test (known at the station as Nos. 5 and 9) are of the Stirling "W" type, having 23,653 sq. ft. of effective heating surface, and operating at 225 lb. per sq. in. gage pressure and 600 deg. fahr. total steam temperature at the superheater outlet. They are identical in design and construction, as shown in Fig. 1, except that in No. 9 boiler the "B" type baffle[3] in the first pass is behind the third row of tubes, whereas the baffle in No. 5 boiler is behind the second row of tubes. Further, air backs were added to the tuyère stacks along the east front wall of No. 5 boiler within the last two years, while its west front wall and both front walls of No. 9 boiler are without such air backs.

Each furnace is fired on two sides by underfeed stokers of thirteen retorts each, placed in service about twelve years ago. Sectional extension grates and separate rotary clinker grinders are employed.

The furnace settings are solid refractory walls throughout, built of a standard brand of first-quality fireclay brick, laid with thin dipped fireclay joints, and backed with second-quality firebrick. The front walls are 22½ in. thick, and the side walls 28 in. thick. The outside face of the walls is coated with an oil compound containing asbestos to reduce air infiltration.

PROCEDURE FOLLOWED IN TESTS

The test methods employed in this investigation were based on earlier preliminary work. Temperature gradients through the refractory walls were measured by the method[4] devised by the U. S. Bureau of Mines in its five-year study of boiler-furnace refractories, in conjunction with the special research committee of The American Society of Mechanical Engineers.

The first attempts to collect samples of the furnace gases were made with a water-cooled sampling tube 12 ft. in length, but this method was found to be unsatisfactory in several respects. It was impossible to collect a large number of samples simultaneously from different parts of the furnace. The sampling positions could not be located with sufficient accuracy, nor could successive samples be taken from identical positions during the same or different tests. It was not feasible to collect composite gas samples over long periods of time. Instantaneous and composite samples could not easily be taken simultaneously at the same sampling point. The water-cooled sampling tube was heavy and unwieldy, and its manipulation involved considerable labor. Its length was limited to the space between boilers, which prevented a complete traverse of the furnace walls.

As a result of these limitations, it was found necessary to devise other methods for taking samples of the gases within the furnace. The method which was finally developed involved a minimum of effort and labor, was positive and easily controlled, and permitted the taking of composite and instantaneous samples simultaneously over any desired period of time. The sampling positions were definite and remained the same for all tests.

---

[1] Technical Engineer, Connors Creek Power House, The Detroit Edison Company, Detroit, Mich. Assoc-Mem. A.S.M.E. Mr. Pasini was born February 9, 1900, in Paterson, N. J. He received his earlier education in Providence, R. I., where he also served a machinist apprenticeship course with The Builders Iron Foundry. He was graduated from the University of Michigan with the engineering class of 1923.

[2] Chemical Engineer, Harbison-Walker Refractories Company, Pittsburgh, Pa. Mr. Sarraf was born April 29, 1903, in Pittsburgh, Pa. He received the degrees of B.S. and M.S. in Chemical Engineering from the University of Pittsburgh, where he also served on the faculty of the Department of Chemistry in 1925. Since February, 1926, he has been engaged in technical sales work with the Harbison-Walker Company.

[3] Paul W. Thompson, "Tests of a Type W Stirling Boiler at the Connors Creek Power House of the Detroit Edison Company," Trans. A.S.M.E., vol. 44 (1922), p. 1005.

Contributed by the Special Research Committee on Boiler Furnace Refractories and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[4] Ralph A. Sherman and W. E. Rice, "Refractories Service Conditions in Furnaces Burning Pittsburgh Coal on Chain Grates," MECHANICAL ENGINEERING, vol. 48 (1926), p. 1115.

The locations of the test equipment are detailed in Figs. 2 and 3, and the various test methods are described in detail below.

Refractories Test Panels. The test panels (marked *A* in Figs. 2 and 3) of unused firebrick of the same brand as the settings, were approximately 36 in. × 24 in. in size and were built into the west front wall of each furnace, about 40 in. above the top of the tuyères, and 2 ft. on either side of the center of the wall.

Temperature Gradients Through the Walls. One 9-in. straight, used as a header (marked *X* in Figs. 2, 3, 7, and 8) in the center of each test panel, was equipped with five thermocouples in accordance with the method of the U. S. Bureau of Mines. The first two couples, spaced accurately 1/2 in. and 1 1/2 in. from the hot face, were platinum, platinum-rhodium couples, as the temperatures at these points were found to be above the safe limits for base-metal couples. Chromel-alumel couples were satisfactorily used at positions 3 in., 4 1/2 in., and 6 in. from the hot face. All thermocouple readings were corrected for cold-junction temperature.

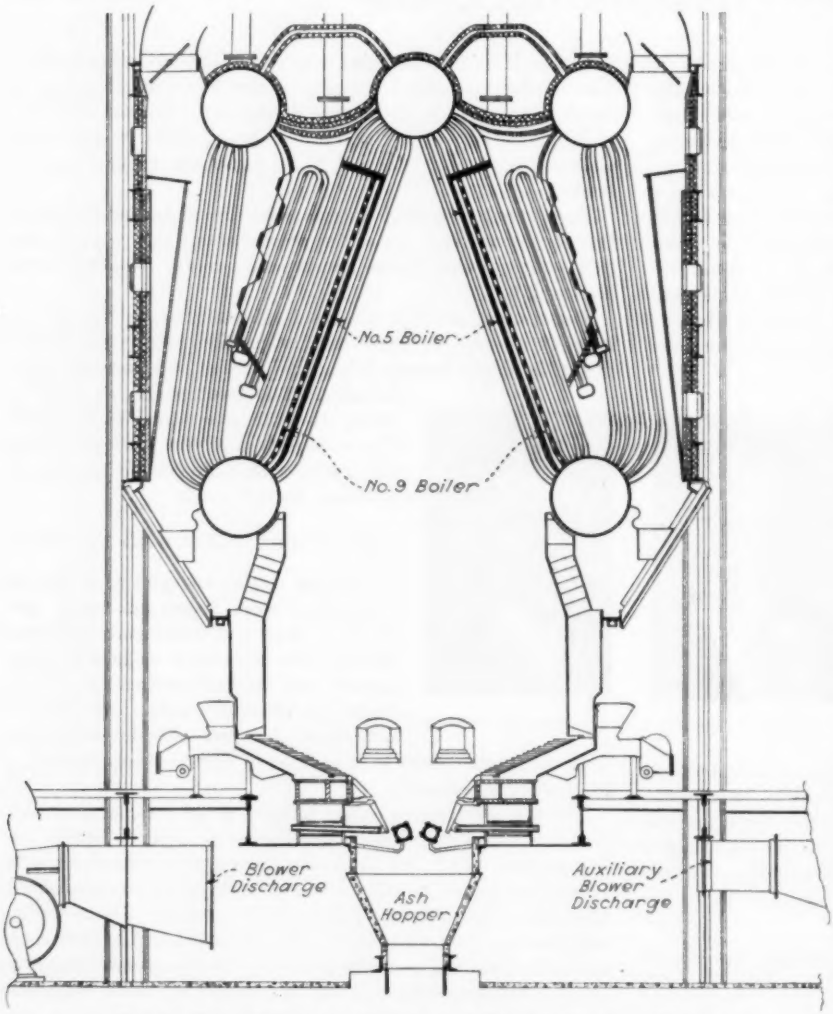


FIG. 1 CROSS-SECTION OF NOS. 5 AND 9 BOILERS (LOOKING NORTH)

Thermocouple readings were taken at hourly intervals during the 14-day period of the tests, and at 15-min. intervals while the individual tests were in progress. Extrapolation of curves drawn from these data gave the temperatures of the hot face of the wall.

Furnace-Gas Compositions. Gas samples were drawn from the furnace at eight sampling points (marked *B* in Figs. 2 and 3) located across the west front wall of the setting at significant positions with respect to tuyères and stoker rams, and at approximately the same elevation as the thermocouple brick in the test panels. Sillimanite tubes were inserted through 2-in. holes punched through the furnace wall, and projected 4 in. into the furnace. The spaces around the tubes were filled with mineral wool to within 5 in. of the hot face (to prevent breakage of the tubes as the walls shifted during the heating-up period), and the spaces at the hot face were filled with fireclay patching mixture.

The sampling-tube connections are shown in detail in Fig. 4. The sillimanite tubes were connected on the outside of the furnace wall to 1/2-in. steel piping leading to the sampling bench, through special copper-tubing connections which served also as stuffing boxes. Each sampling line was fitted with a separate laboratory-type aspirator operated by compressed air. All lines were adjusted to a negative pressure of approximately one inch of mercury.

Furnace gas was drawn continuously through the system and exhausted through the aspirator. Gas samples were taken by siphon action from the sampling offtakes. The rubber-tubing connections were made as short as possible. Before collecting each set of samples, the lines were carefully scavenged, and all connections were checked to eliminate air leakage.

Gas samples were collected in 2 1/2-gal. sampling bottles over saturated sodium chloride solution to reduce the solubility of CO_2 in the collecting medium. The samples were of two types: composite samples, drawn at a uniform rate over a period of one hour, and snap samples, drawn as rapidly as possible in order to determine the conditions at that particular time, and representing a sampling time of 30 to 45 sec. Three sets of composite samples were taken during each test, with a half-hour interval between sets. During all but the first three tests, a set of snap samples was taken at 5-min. intervals simultaneously with the composite sample at No. 5 sampling position, thus affording a cross-section of the composite sample. These snap samples were usually taken during the second set of composite samples.

All composite gas samples were analyzed for CO_2, O_2, H_2, and CO, using a standard combustion-type apparatus; and N_2 was determined by difference. Most of the snap samples were analyzed on the combustion-type apparatus, and when it proved necessary to use the Orsat apparatus for this purpose, care was exercised to insure complete removal of CO before reading the apparatus.

Composite Gas Samples at Entrances to Superheater Pass and Damper. During each test, gas samples were drawn at 15-min. intervals on both sides of the boiler at the entrance to the superheater pass and at the top of the last pass, just below the dampers, and were analyzed with Orsat apparatus for CO_2, O_2, and CO. The samples were composites taken from a 3-in. manifold, having six sampling pipes 3/8 in. in diameter extending into the pass. The gases were mixed in the manifold and drawn continuously

from its midpoint by means of a blower. The sampling connection to the Orsat apparatus was located in the system just ahead of the blower. Special high-temperature tubing was used for the sampling tubes in the superheater pass, and extra heavy steel piping at the damper.

Furnace-Gas Temperatures. An approximation of furnace-gas temperatures was secured during the tests by readings at 15-min. intervals with an optical pyrometer, focused on the closed ends of silicon carbide tubes (marked *C* in Figs. 2 and 3) located in the center of the west front wall of each furnace about 11 in. above No. 4 sampling position (see Figs. 7 and 8), and at a position in the side walls opposite the apex of the combustion chamber. The tubes in the west front walls projected 10 in. into the furnace; and at the apex of the combustion chamber, the tube projected 10 in. into the furnace in No. 5 boiler, and 4 in. in No. 9 boiler, the latter tube being smaller.

Operating-Instrument Readings. Readings were taken at 15-min. intervals during the tests of the following operating instruments:

1 Steamflow-airflow meter
2 Steam pressures: saturated, superheat
3 Total steam temperatures
4 Draft readings: under fire, over fire, damper
5 Gas temperatures at damper.

Hourly readings of the steam output were also taken during the 14-day test period.

Coal and Ash Determinations. Coal samples were taken at half-hour intervals during the test runs, at three positions in each coal spreader, for proximate analysis and determination of fusion point of the ash.

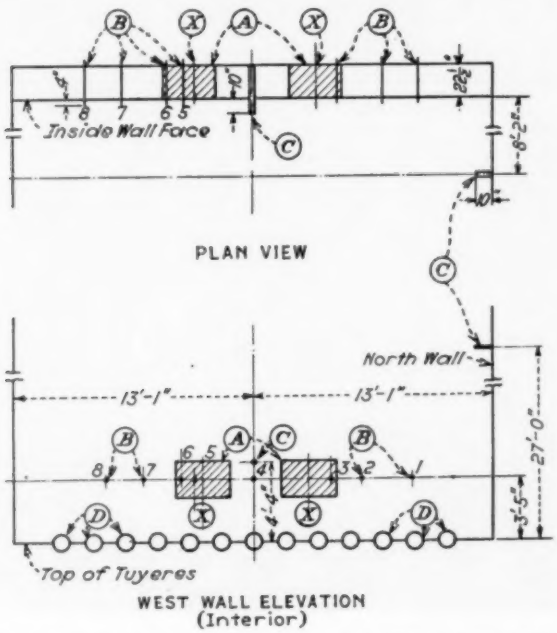


FIG. 2 LOCATIONS OF TEST EQUIPMENT IN WEST FRONT WALL, NO. 5 FURNACE

(*A*, test panel; *B*, sillimanite tubes; *C*, silicon carbide tubes; *D*, stoker rams; *X*, thermocouple brick.)

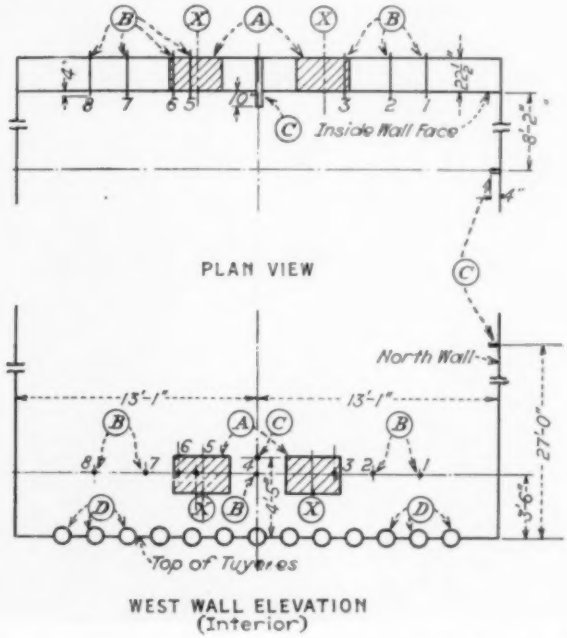


FIG. 3 LOCATIONS OF TEST EQUIPMENT IN WEST FRONT WALL, NO. 9 FURNACE

A, test panel; *B*, sillimanite tubes; *C*, silicon carbide tubes; *D*, stoker rams; *X*, thermocouple brick.)

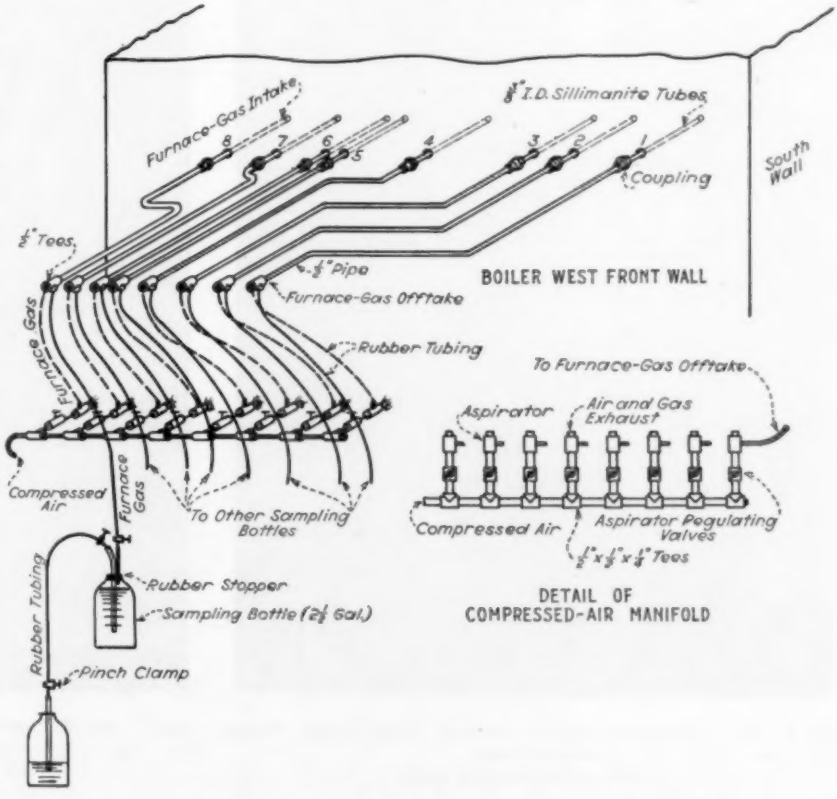


FIG. 4 LAYOUT OF GAS-SAMPLING APPARATUS



FIG. 5 NO. 5 FURNACE, NORTH END, WEST FRONT WALL, AFTER TESTS

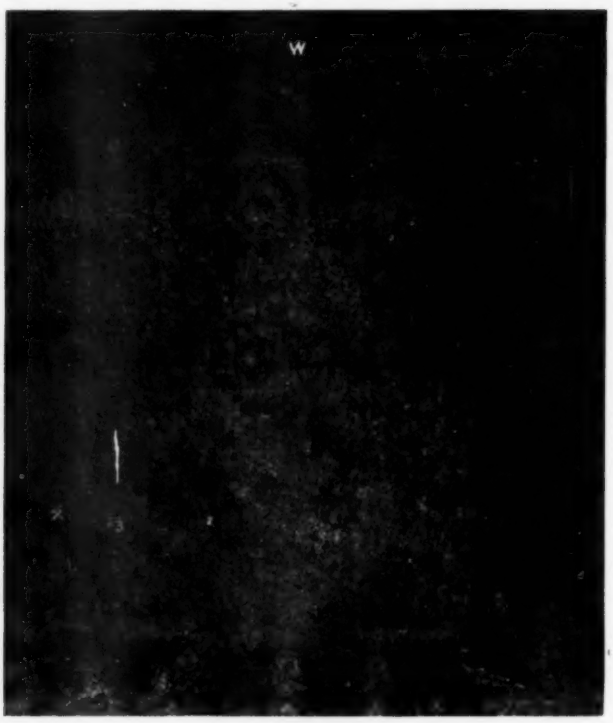


FIG. 6 NO. 9 FURNACE, NORTH END, WEST FRONT WALL, AFTER TESTS

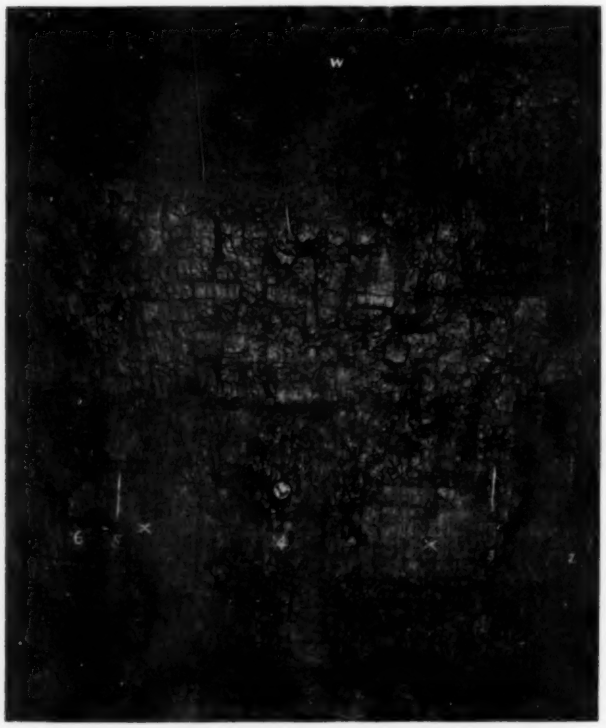


FIG. 7 NO. 5 FURNACE, CENTER SECTION, WEST FRONT WALL, AFTER TESTS
(× indicates thermocouple brick.)

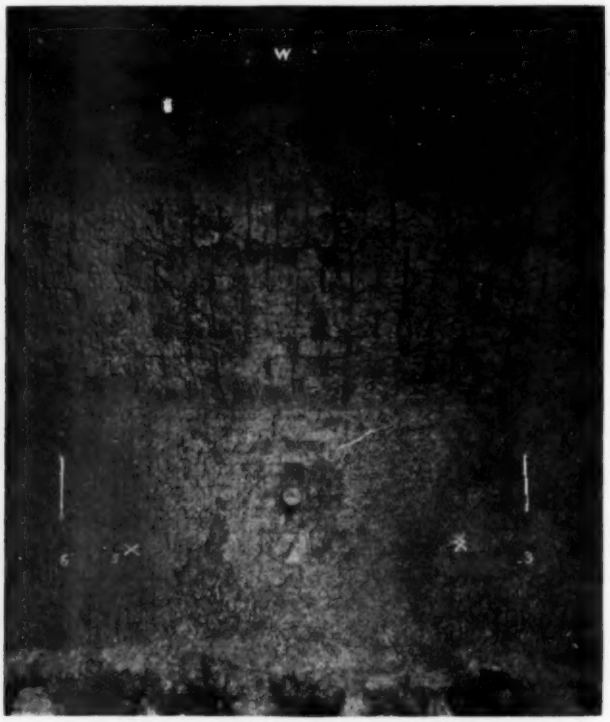


FIG. 8 NO. 9 FURNACE, CENTER SECTION, WEST FRONT WALL, AFTER TESTS
(× indicates thermocouple brick.)

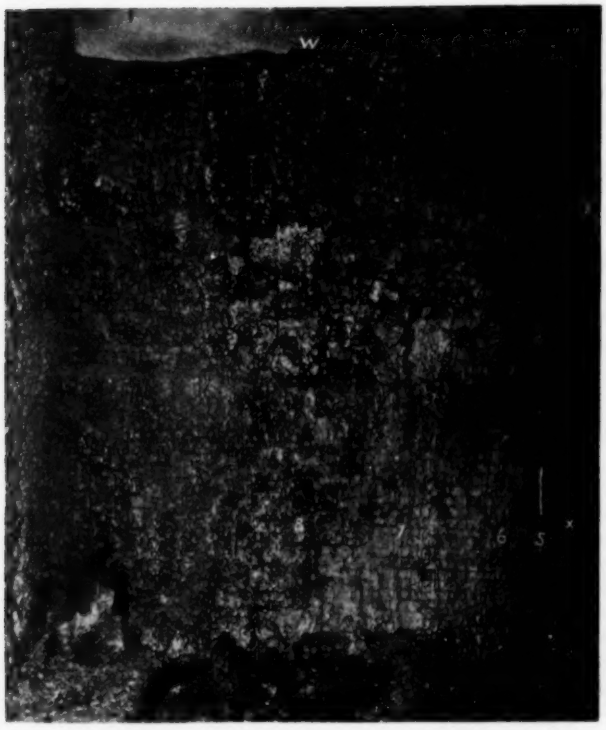


FIG. 9 NO. 5 FURNACE, SOUTH END, WEST FRONT WALL, AFTER TESTS

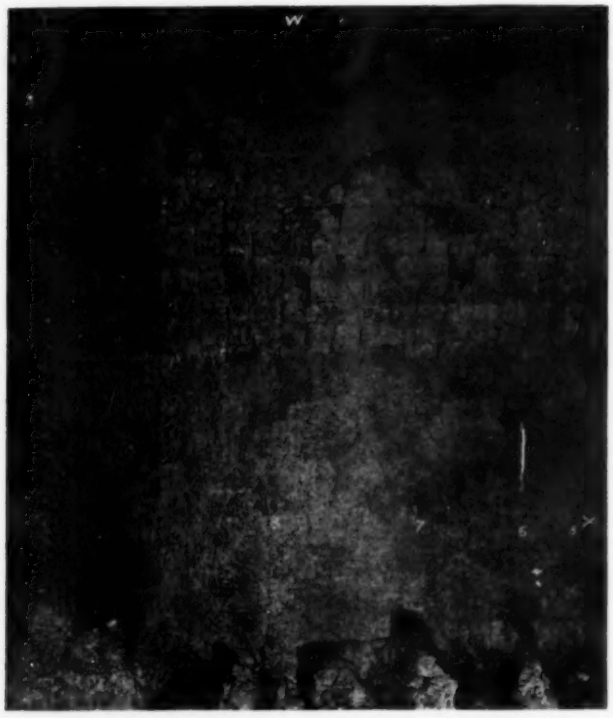


FIG. 10 NO. 9 FURNACE, SOUTH END, WEST FRONT WALL, AFTER TESTS

TABLE 1 SUMMARIZED TEST DATA, NO. 5 BOILER

Test No.	Boiler output, per cent of rating	Fuel-bed thickness	Steam-flow-air-flow meter: Steam flow, lb. per hr.	Air flow: East side	Air flow: West side	Steam pressure, lb. per sq. in.: Saturated	Steam pressure, lb. per sq. in.: Superheated	Steam temperature, deg. fahr.: East side	Steam temperature, deg. fahr.: West side	Composite gas samples, per cent:	Superheater: East side	Superheater: West side	Damper: East side	Damper: West side	Total air coefficient	Draft readings, inches of water: Under fire: East side	Under fire: West side	Damper: East side	Damper: West side	Over fire
										CO_2	12.0	12.2	11.9	11.7						
										O_2	7.3	7.4	7.6	7.4						
F	100	Normal	69	68	68	228	227	592	584	CO	0.0	0.0	0.0	0.0	1.54	0.9	0.9	0.14	0.11	0.04
										N_2	80.7	80.4	80.5	80.9						
										CO_2	13.8	13.3	13.7	13.1						
										O_2	5.3	5.9	5.4	5.8						
A	145	Normal	100	100	100	232	227	616	619	CO	0.1	0.0	0.1	0.0	1.36	2.3	2.3	0.29	0.27	0.07
										N_2	80.8	80.8	80.8	81.1						
										CO_2	14.4	13.8	13.3	14.4						
										O	4.7	5.4	5.5	4.7						
E	186	Normal	129	127	130	235	227	636	629	CO	0.0	0.1	0.0	0.0	1.31	3.5	3.6	0.47	0.44	0.08
										N_2	80.9	80.7	81.2	80.9						
										CO_2	14.1	13.7	14.2	13.4						
										O_2	4.8	5.5	4.9	5.5						
G	230	Normal	159	159	156	239	228	658	659	CO	0.0	0.1	0.0	0.0	1.32	4.9	5.1	0.77	0.77	0.06
										N_2	81.1	80.7	80.9	81.1						
										CO_2	12.3	11.9	12.7	11.8						
										O_2	6.5	7.5	6.6	7.4						
J	145	Light	100	99	99	232	228	648	639	CO	0.0	0.0	0.0	0.0	1.49	2.7	2.7	0.39	0.37	0.07
										N_2	81.2	80.6	80.7	80.8						
										CO_2	15.9	16.3	16.0	15.4						
										O_2	2.3	2.1	2.2	2.7						
K	143	Heavy	99	101	100	233	229	617	615	CO	0.6	0.7	0.7	0.5	1.14	1.6	1.6	0.23	0.21	0.07
										N_2	81.2	80.9	81.1	81.4						

TABLE 2 SUMMARIZED TEST DATA, NO. 9 BOILER

Test No.	Boiler output, per cent of rating	Fuel-bed thickness	Steam-flow-air-flow meter: Steam flow, lb. per hr.	Air flow: East side	Air flow: West side	Steam pressure, lb. per sq. in.: Saturated	Steam pressure, lb. per sq. in.: Superheated	Steam temperature, deg. fahr.: East side	Steam temperature, deg. fahr.: West side	Composite gas samples, per cent:	Superheater: East side	Superheater: West side	Damper: East side	Damper: West side	Total air coefficient	Draft readings, inches of water: Under fire: East side	Under fire: West side	Damper: East side	Damper: West side	Over fire
										CO_2	14.5	14.1	13.8	14.0						
										O_2	4.3	4.5	5.3	4.7						
C	102	Normal	71	71	68	229	227	592	603	CO	0.0	0.0	0.0	0.0	1.31	1.1	1.1	0.15	0.16	0.05
										N_2	81.2	81.4	80.9	81.3						
										CO_2	15.4	15.3	14.9	14.6						
										O_2	3.4	3.6	4.2	4.0						
B	143	Normal	99	100	100	231	228	622	625	CO	0.1	0.0	0.0	0.0	1.24	2.3	2.3	0.24	0.26	0.04
										N_2	81.1	81.1	80.9	81.4						
										CO_2	16.2	16.2	15.5	15.3						
										O_2	1.9	1.9	2.8	2.5						
D	187	Normal	129	130	131	235	229	627	629	CO	0.3	0.4	0.3	0.3	1.16	3.6	3.6	0.47	0.46	0.08
										N_2	81.6	81.5	81.4	81.9						
										CO_2	16.1	16.7	16.0	16.0						
										O_2	2.4	2.1	2.6	2.3						
	30	Normal	159	161	159	241	229	661	660	CO	0.1	0.1	0.1	0.1	1.14	5.5	5.6	0.74	0.74	0.08
										N_2	81.4	81.1	81.3	81.6						

Fires were started in the two boilers under test at 7:30 p.m. on Sunday, March 10. The load cycle on both boilers was kept practically the same during the entire test period, as was shown by the records of the steam-flow meters.

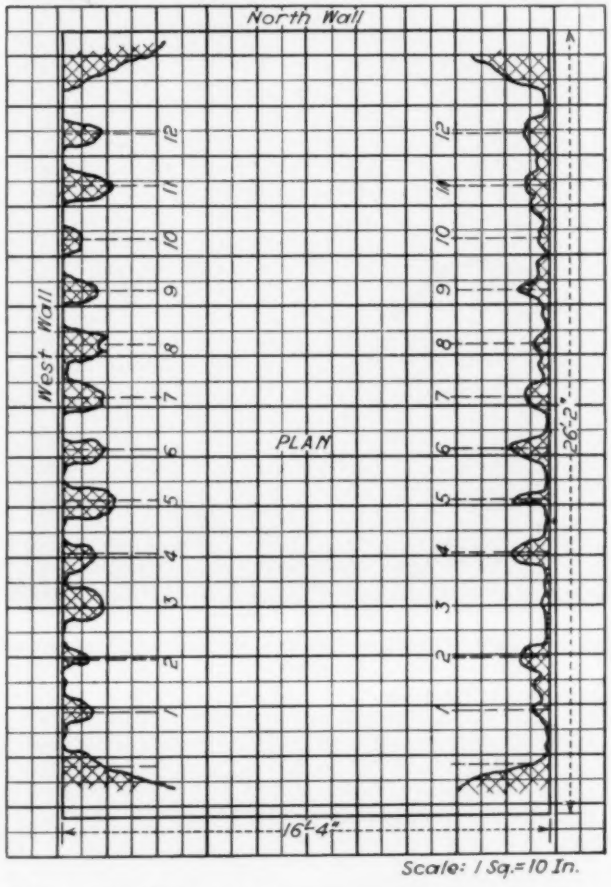


FIG. 11 PLAN VIEW, SLAG ACCUMULATIONS ABOVE TUYÈRES IN NO. 5 FURNACE, AFTER TESTS

The schedule of tests was as follows:

Test	Date	Boiler No.	Boiler output, per cent of rating	Fuel-bed thickness
A	3-13-30	5	145	Normal
B	3-14-30	9	145	Normal
C	3-15-30	9	102	Normal
D	3-17-30	9	187	Normal
E	3-18-30	5	186	Normal
F	3-19-30	5	100	Normal
G	3-20-30	5	230	Normal
H	3-21-30	9	230	Normal
J	3-22-30	5	145	Light
K	3-23-30	5	143	Heavy

Four days were allowed to elapse between starting the fires and the first test, in order to secure equilibrium conditions of heat flow through the walls. For the same reason, the load at which each test was to be run was applied to the boiler at least 6 to 7 hours before starting to take readings. While readings were being taken, the clinker grinders were not in operation, so as not to disturb the fuel bed; however, during the latter part of tests G and H, it proved necessary to run the grinders in order to maintain the desired load.

Immediately following the last test, the boilers were shut down for inspection. Photographs were taken of the settings (see Figs. 5 to 10, inclusive) and the brick in the test panels were removed from the walls and carefully inspected.

RESULTS

The various test data are listed in Tables 1 to 14, inclusive.

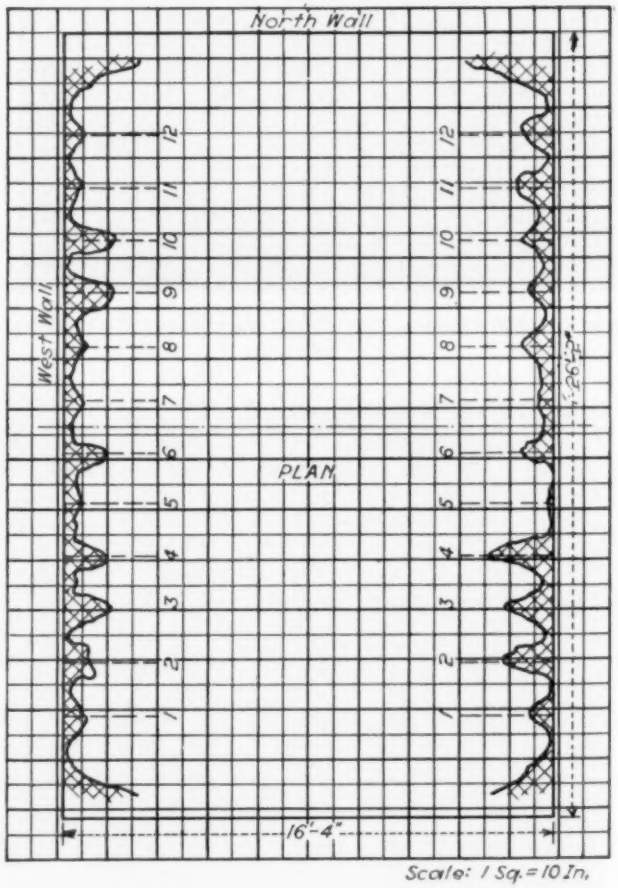


FIG. 12 PLAN VIEW, SLAG ACCUMULATIONS ABOVE TUYÈRES IN NO. 9 FURNACE, AFTER TESTS

TABLE 3 SUMMARIZED TEST DATA, COMPARATIVE TEMPERATURES (DEG. FAHR.)

Test No.	Boiler No.	Boiler output, per cent of rating	Fuel-bed thickness	Furnace-wall temperatures: North panel, Distance from hot face, inches: ½	1½	3	4½	6	Furnace-wall temperatures: South panel, Distance from hot face, inches: ½	1½	3	4½	6	Temperature of wall face (extrapolated): North panel	South panel	Average	Optical-pyrometer temperatures: Center wall	Apex combustion chamber	Gas temperatures at damper: East side	West side	Steam temperatures: East side	West side
F	5	100	Normal	2341	2289	2135	2026	1911	2333	2244	2087	1974	..	2381	2375	2378	2366	1590	513	502	592	584
C	9	102	Normal	2415	..	2116	1974	1801	2357	2253	2075	1917	1774	2468	2410	2439	2460	1696	504	493	592	603
A	5	145	Normal	2547	2444	2213	2035	1875	2353	2411	2181	1995	..	2605	2622	2614	2590	1674	576	555	616	619
B	9	143	Normal	2465	..	2182	2040	1869	2435	2315	2151	1974	1797	2521	2597	2559	2501	1816	575	565	622	625
E	5	186	Normal	2606	2512	2299	2144	1997	2630	2495	2269	2111	..	2655	2691	2673	2629	1802	637	624	636	629
D	9	187	Normal	2505	..	2190	2020	1819	2513	2392	2165	1985	1785	2564	2578	2571	2581	1850	638	600	627	629
G	5	230	Normal	2678	2583	2322	2153	2001	2655	2517	2239	2090	..	2738	2726	2732	2613	1841	713	701	658	659
H	9	230	Normal	2669	..	2295	2133	1940	2619	2496	2253	2078	1886	2730	2686	2708	2678	1960	722	714	661	660
J	5	145	Light	2493	2423	2209	2086	1966	2546	2425	2188	2079	..	2540	2598	2469	2455	1611	611	594	648	639
K	5	143	Heavy	2677	2582	2298	2141	2004	2756	2609	2272	2139	..	2737	2833	2785	2621	1775	560	547	617	615

Tables 1 and 2 give summarized boiler operating-instrument readings on Nos. 5 and 9 boilers, respectively, and Table 3 contains data on the temperatures obtained during the tests. In Tables 4 to 13 are tabulated the analyses of the composite and snap furnace-gas samples. Table 14 describes the coal used in the investigation, and lists the proximate analyses, calorific values, and ash-fusion points of the samples of coal taken during each test.

During the entire series of tests, every effort was made to hold the two boilers under the same operating conditions at all times. As a general rule, the fuel bed in No. 5 furnace was somewhat lighter than in No. 9 furnace, which is the usual experience with this particular boiler.

The following detailed comments as to the character of the fuel bed and firing conditions in general, during the various tests, may be of interest:

Fuel-Bed Thickness. In the tests at 100 per cent of rating

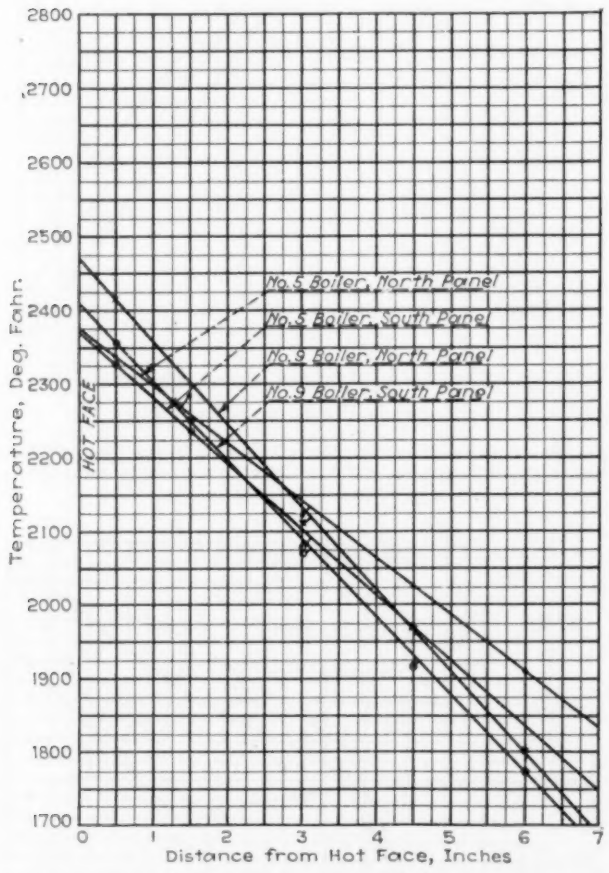


FIG. 13 TEMPERATURE GRADIENTS THROUGH WEST FRONT WALL, NOS. 5 AND 9 FURNACES, 100 PER CENT OF RATING, NORMAL FUEL-BED THICKNESS

(tests F and C), the fires on the east side in No. 5 furnace appeared to be thinner than in No. 9 furnace, while on the west side of each furnace the fires were of normal thickness. This is verified by inspection of the under-fire pressures (Tables 1 and 2).

The fires in No. 5 boiler at 145 per cent of rating (test A) appeared to be thinner on both the east and west sides than in No. 9 boiler at the same rating (test B).

At 185 per cent of rating (tests E and D), the fuel-bed thickness was the same in both furnaces except that the grinder pit carried a higher ash level in No. 5 boiler than in No. 9 boiler. During the first part of the test on No. 9 boiler, the accumulated slag on the south wall started to run.

The fuel-bed thickness in Nos. 5 and 9 boilers was about normal during the runs at 230 per cent of rating (tests G and H), with No. 9 boiler having a slightly heavier fire, as shown by an inspection of the under-fire pressures (Tables 1 and 2). It proved necessary to shorten the tests at this rating, because of the high overloads imposed on the blower motors. In order to maintain the rating during the test on No. 9 boiler, it was necessary to resort to clinker grinding and the consequent agitation of the fuel bed caused a disturbance of the furnace-gas conditions (excessive CO). Because of these abnormal conditions, the data

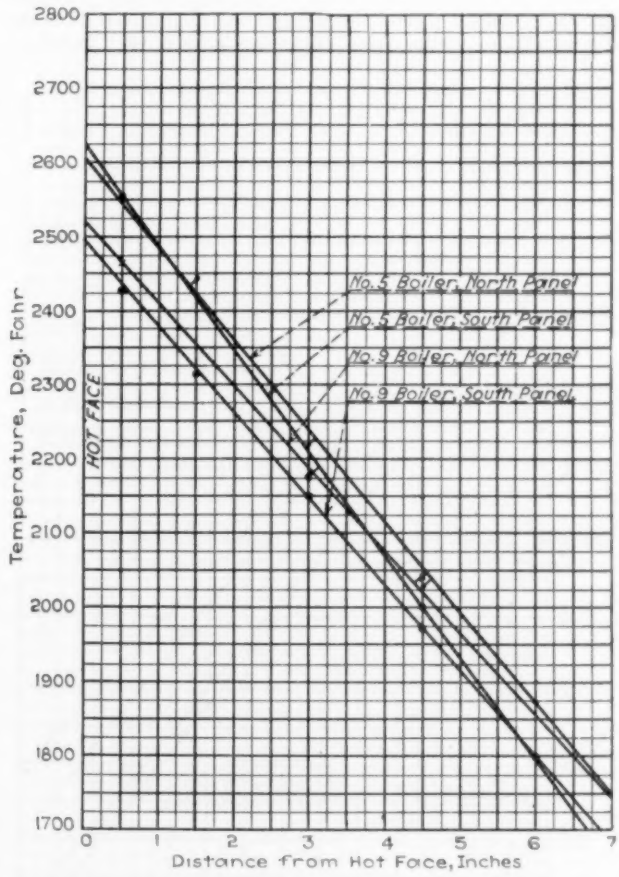


FIG. 14 TEMPERATURE GRADIENTS THROUGH WEST FRONT WALL, NOS. 5 AND 9 FURNACES, 145 PER CENT OF RATING, NORMAL FUEL-BED THICKNESS

on the last set of composite gas samples taken during test H should be discounted.

During the test on No. 5 boiler at 145 per cent of rating with abnormal excess air (test J), the fuel bed was very thin and, in fact, bare iron was showing at the base of the tuyère stacks.

The fuel bed was purposely made very thick in No. 5 furnace at 145 per cent of rating (test K), to establish excess air below the normal for that rating. This is evidenced by a comparison of the under-fire pressures with Test A (see Table I). The slag on the furnace walls was running noticeably throughout this test, and slag accumulations were heavy at the observation doors.

Condition of the Settings. Some idea of the condition of the settings following the completion of the tests may be obtained from Figs. 5 to 10, inclusive. The differences in the character of the slag formations, in the extent of shrinkage and vitrifica-

tion of the brick, and in the tightness of the joints are quite marked.

In No. 5 furnace, serious shrinkage took place in the front walls, with some cracks as wide as $^1/_2$ in. Many of the $13^1/_2 \times 9 \times 2^1/_2$-in. straights used in alternate courses had broken in two. The slag coating (originally 1 in. or more thick) had run from the walls, cleaning large areas of the front walls and lower end walls. The slag on the walls was glassy and vitreous, and stalactites of slag were frequent. Severe spalling had taken place in the leaning walls.

In No. 9 furnace, the setting was much more sound. Less of the original fly-ash and slag coating (somewhat heavier than in No. 5 furnace before the tests) had run from the walls. Shrinkage cracks did not appear through the slag coating, but when the slag was cleaned from the walls, it was evident that shrinkage

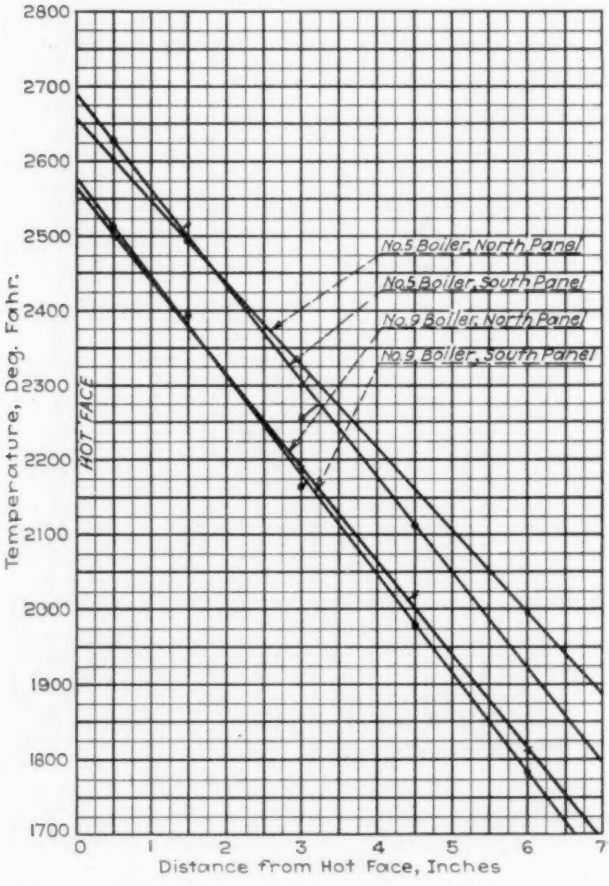


FIG. 15 TEMPERATURE GRADIENTS THROUGH WEST FRONT WALL, NOS. 5 AND 9 FURNACES, 185 PER CENT OF RATING, NORMAL FUEL-BED THICKNESS

and vitrification of the brick had occurred, although to a considerably lesser extent than in No. 5 boiler. The slag coating on the walls was not as glassy as in No. 5 boiler. Considerable spalling had taken place in the leaning walls.

The brick in the test panels in each boiler had been eroded by the slag, and in fact, as is evident from Figs. 5, 7, and 9, ridges were worn in the brick in No. 5 furnace. The brick in No. 9 furnace had wasted away $^3/_{16}$ in. to $^1/_4$ in., and in No. 5 furnace as much as $^3/_8$ in., as a result of slag erosion. Vitrifica-

FIG. 17 (AT RIGHT) VARIATION OF INSIDE WALL-FACE TEMPERATURES (EXTRAPOLATED) WITH RATING, NOS. 5 AND 9 FURNACES, NORMAL FUEL-BED THICKNESS

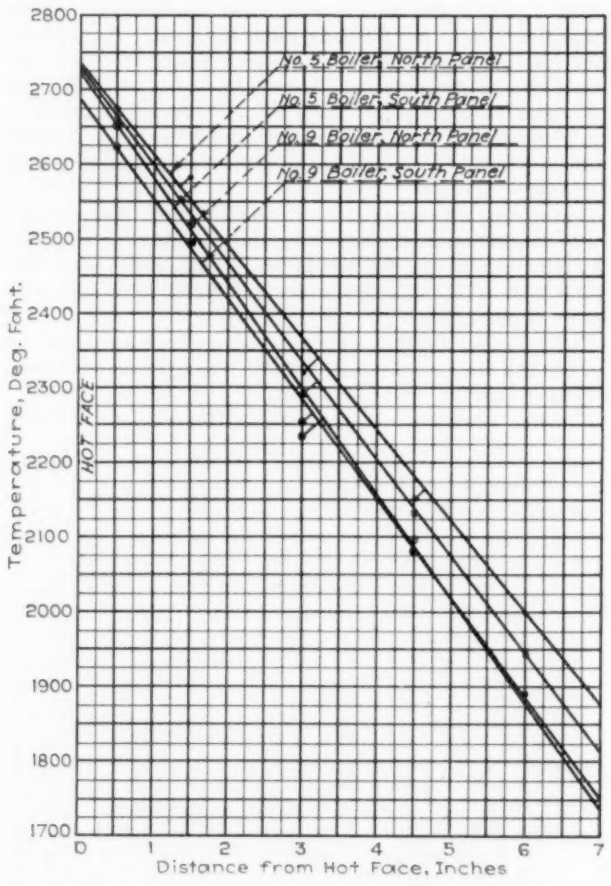
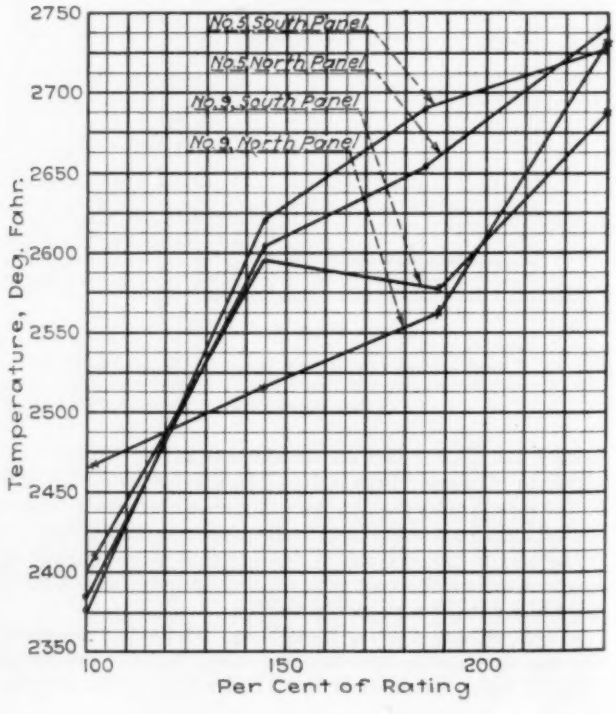


FIG. 16 TEMPERATURE GRADIENTS THROUGH WEST FRONT WALL, NOS. 5 AND 9 FURNACES, 230 PER CENT OF RATING, NORMAL FUEL-BED THICKNESS



tion was evident in the brick to a greater extent in No. 5 than in No. 9 furnace. Apparently the brick had commenced to shrink slightly at the hot face, but to such a small extent that it could not be measured.

Slag Accumulations. Slag accumulations on the walls along the clinker line were lighter than are ordinarily encountered in regular plant operation. The accumulations over the tuyères have been plotted in Figs. 11 and 12. It will be noted that the slag deposits are lighter above the air backs along the east front

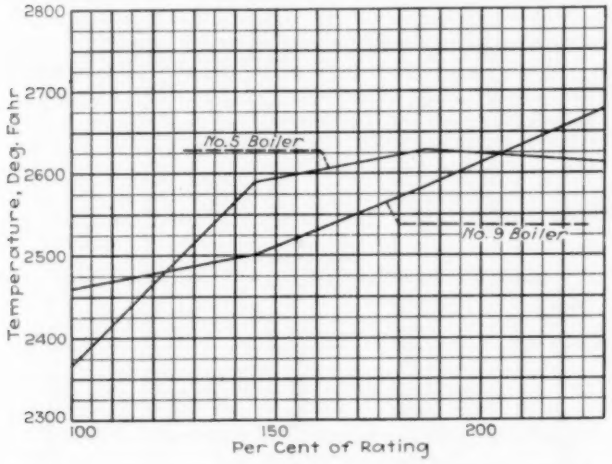


FIG. 18 APPROXIMATE FURNACE-GAS TEMPERATURES, CENTER OF WEST FRONT WALL DIRECTLY OVER FIRES, NOS. 5 AND 9 FURNACES, NORMAL FUEL-BED THICKNESS

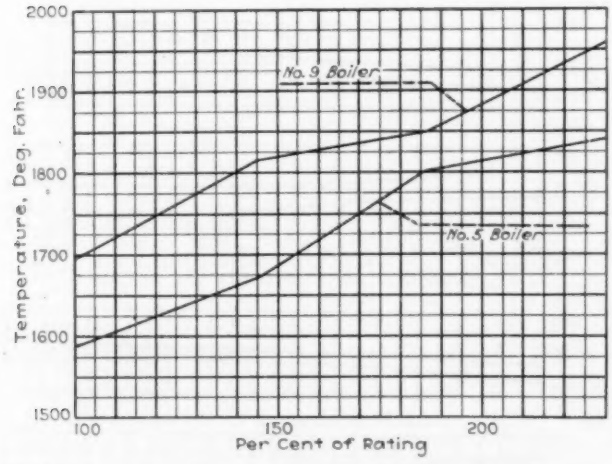


FIG. 19 APPROXIMATE GAS TEMPERATURES AT APEX OF COMBUSTION CHAMBER, NOS. 5 AND 9 FURNACES, NORMAL FUEL-BED THICKNESS

wall of No. 5 boiler than on the west front wall which has no air backs. Figs. 5 to 10, inclusive, show the walls of both furnaces to be fairly free from slag, resulting from the fact that the last tests, conducted at the higher ratings, involved high temperatures which caused the slag deposits to run from the walls.

Stoker Behavior. The tuyère stacks in both boilers were inspected following the tests. In No. 5 furnace, repairs were necessary to Nos. 2, 3, 4, 5, 6, 7, 8, 9, and 12 tuyère stacks on the west side. Nos. 5 and 12 tuyère stacks on the west side of No. 9 furnace were of the old single-piece type, and these were completely burned out, and in addition, repairs were also necessary to Nos. 3, 6, 7, and 8 tuyère stacks. The plenum chambers

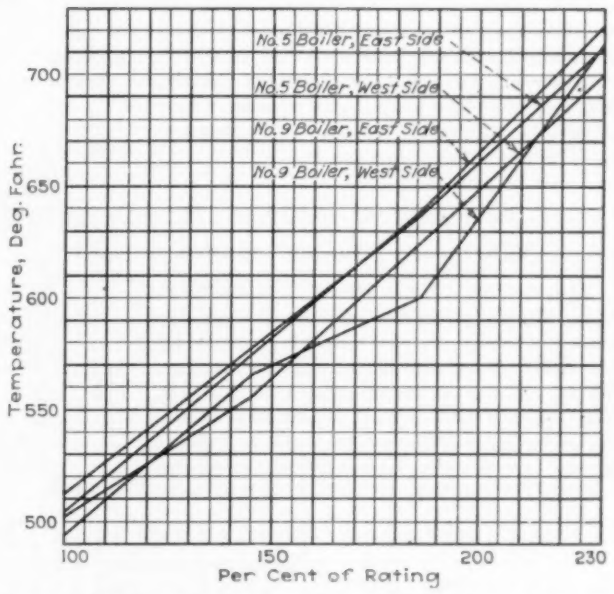


FIG. 20 GAS TEMPERATURES AT DAMPER, NOS. 5 AND 9 BOILERS, NORMAL FUEL-BED THICKNESS

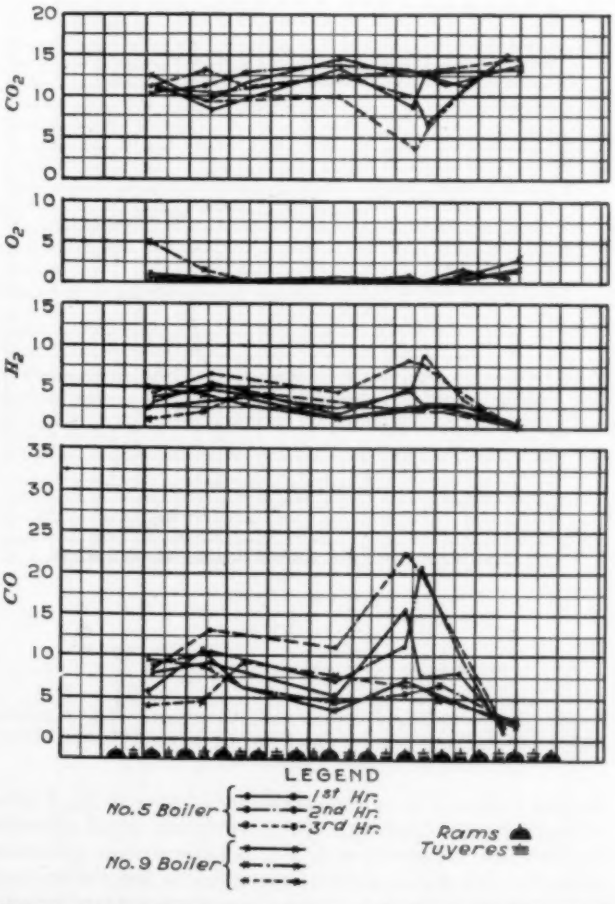


FIG. 21 FURNACE-GAS COMPOSITIONS, COMPOSITE GAS SAMPLES, NOS. 5 AND 9 FURNACES, 185 PER CENT OF RATING, NORMAL FUEL-BED THICKNESS

of both furnaces were also inspected to discover any difference affecting air distribution, and no such differences were found.

These remarks on slag accumulations, tuyères, and wind chambers are included because of their possible bearing on the admission of the air and its deflection into the furnace.

DISCUSSION OF RESULTS

Tests With Normal Fuel-Bed Thickness

Temperatures. For purposes of comparison, the temperature-gradient curves through the west front walls of Nos. 5 and 9 furnaces, for the same rating, are plotted together as Figs. 13, 14, 15, and 16, representing ratings of 100, 145, 185, and 230 per cent, respectively. Careful examination of these figures and of Table 3 shows that the temperatures of the hot face of

TABLE 4 FURNACE-GAS COMPOSITIONS—TEST F, NO. 5 BOILER

(100 per cent of rating, normal fuel bed thickness)

Composite Gas Samples

Period of sampling, p.m.		Sampling position 1	2	3	4	5	6	7	8
		Per cent							
1:00 to 2:00	CO_2	11.3	14.0	(Sampling tube broken)	15.3	14.9	15.4	14.4	13.3
	O_2	5.7	0.9		1.2	1.3	0.7	1.0	4.1
	H_2	0.0	1.2		0.2	0.4	0.4	0.7	0.1
	CO	0.4	2.9		0.6	0.7	0.6	1.8	0.1
	N_2	82.6	81.0	.....	82.7	82.7	82.9	82.1	82.4
	Free carbon	Trace	Trace	.....	No	Trace	No	No	Trace
	Solution colored	No	No	.....	No	No	No	No	No
	Oil film	No	No	.....	No	No	No	No	No
2:30 to 3:30	CO_2	13.7	15.3	.....	14.8	15.5	15.7	13.0	13.4
	O_2	2.9	0.6	.....	2.2	1.4	0.4	1.4	3.9
	H_2	0.1	0.5	.....	0.0	0.2	0.4	1.5	0.0
	CO	0.2	1.4	.....	0.0	0.6	1.4	3.4	0.2
	N_2	83.1	82.2	.....	83.0	82.3	82.1	80.7	82.5
	Free carbon	Trace	Yes	.....	No	Trace	Trace	Yes	No
	Solution colored	No	No	.....	No	No	No	No	No
	Oil film	No	No	.....	No	No	No	No	No
4:00 to 5:00	CO_2	12.6	15.4	.....	14.4	14.8	15.0	14.2	12.8
	O_2	4.9	0.2	.....	2.9	2.5	2.0	2.3	5.0
	H_2	0.0	0.7	.....	0.2	0.5	0.0	0.4	0.0
	CO	0.2	1.8	.....	0.2	0.3	0.2	1.0	0.0
	N_2	82.3	81.9	.....	82.3	81.9	82.8	82.8	82.2
	Free carbon	Trace	Trace	.....	No	No	No	No	No
	Solution colored	No	No	.....	No	No	No	No	No
	Oil film	No	No	.....	No	No	No	No	No

Snap Samples

(Taken at position 5 during second set of composite samples)

Sample	Time taken, p.m.	CO_2	O_2	CO
		Per cent		
1	2:30	16.6	0.8	1.0
2	2:35	15.4	2.1	0.3
3	2:40	16.0	2.1	0.1
4	2:45	14.3	4.4	0.2
5	2:50	15.4	2.7	0.1
6	2:55	16.5	0.8	0.5
7	3:00	16.3	0.5	1.1
8	3:05	15.5	0.4	2.1
9	3:10	15.7	0.4	2.5
10	3:15	16.0	1.6	0.2
11	3:20	16.0	1.9	0.1
12	3:25	16.0	0.7	0.8
Average		15.8	1.5	0.8
Composite sample Position 5 (second set)		15.5	1.4	0.6

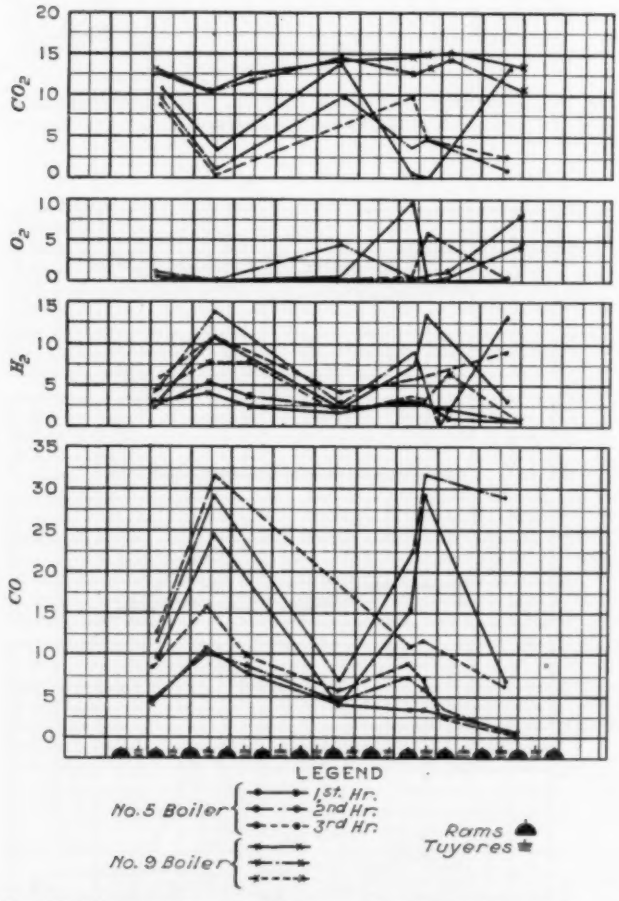


Fig. 22 Furnace-Gas Compositions, Composite Gas Samples, Nos. 5 and 9 Furnaces, 230 Per Cent of Rating, Normal Fuel-Bed Thickness

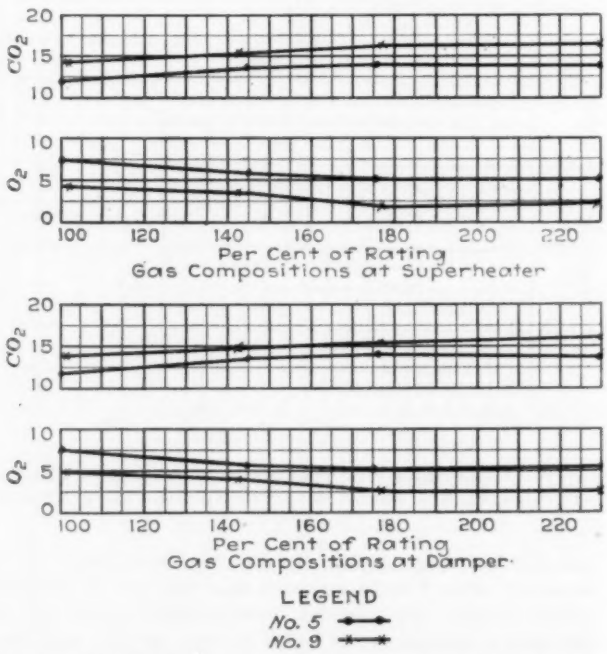


Fig. 23 Gas Compositions at Superheater and Damper, Nos. 5 and 9 Boilers, Normal Fuel-Bed Thickness

the wall (obtained by extrapolation) were higher in No. 5 than in No. 9 furnace, except at 100 per cent of rating, where, although the wall face temperatures in No. 5 furnace were somewhat lower, the temperature-gradient curves had a lesser slope than No. 9 furnace, indicating a higher average temperature through the wall. Temperatures well above 2700 deg. fahr. were obtained in both furnaces at 230 per cent of rating. The relationship between boiler rating and wall-face temperature is shown in Fig. 17.

The evidence of higher furnace temperatures in No. 5 furnace than in No. 9 was confirmed by the optical-pyrometer readings at the center of the west front wall, directly above the fuel bed.

However, the optical-pyrometer readings at the apex of the combustion chamber, and the damper gas temperatures were higher in No. 9 than in No. 5 furnace. At the same time, the steam temperatures were higher in No. 9 than in No. 5 boiler.

The explanation of this condition is apparently dependent on the fact that, since the fuel-bed temperatures were higher in No. 5 furnace, a greater proportion of the energy liberated was in the form of radiant heat. At the apex of the combustion chamber, that portion of the heat energy available by convection was less in No. 5 boiler than in No. 9 boiler, and consequently the temperatures at this point in the furnace were higher in No. 9 boiler. This also accounts for the higher steam temperatures obtained in No. 9 boiler than in No. 5 boiler.

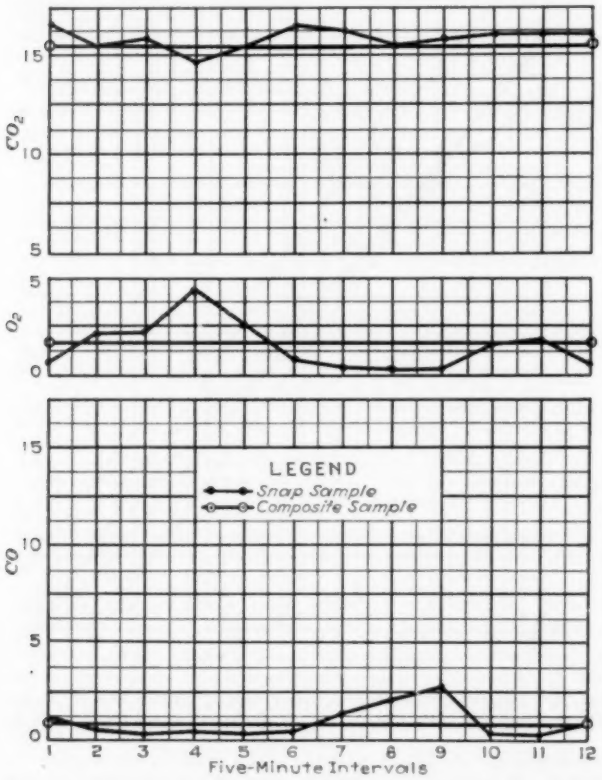


FIG. 24 FURNACE-GAS COMPOSITIONS, SNAP SAMPLES, NO. 5 FURNACE, 100 PER CENT OF RATING, NORMAL FUEL-BED THICKNESS

The optical-pyrometer readings are of little value as absolute data on furnace-gas temperatures, but are probably reasonably accurate as a measure of comparative temperature conditions in the furnaces.

The optical-pyrometer data and stack-gas temperatures given in Table 3 are plotted against boiler rating in Figs. 18, 19, and 20.

Furnace-Gas Compositions. To enable ready comparison, typical analyses of the composite furnace gas samples drawn from both boilers at the same rating (as given in Tables 4 to 11 inclusive) are plotted together, and the locations of stokers and tuyères relative to the sampling positions are also indicated. These data are shown in Figs. 21 and 22 representing ratings of 185 and 230 per cent, respectively.

It is evident that the furnace conditions immediately above the fuel bed were much more strongly reducing in No. 5 than in No. 9 boiler, particularly at the higher ratings. For some unaccountable reason, this situation existed in spite of the fact that the fuel bed in No. 5 boiler was usually somewhat thinner, offering less resistance to the

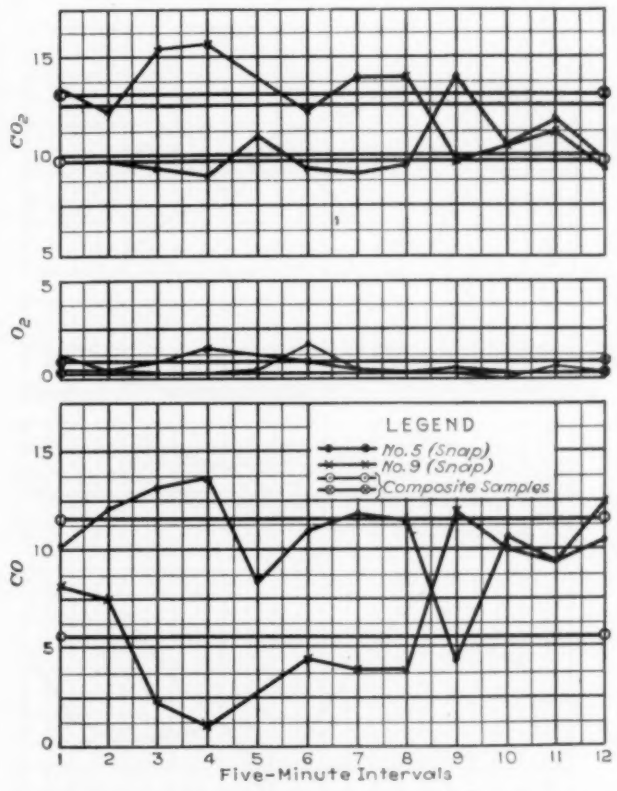


FIG. 25 FURNACE-GAS COMPOSITIONS, SNAP SAMPLES, NOS. 5 AND 9 FURNACES, 185 PER CENT OF RATING, NORMAL FUEL-BED THICKNESS

draft, and also that the gas analyses at the superheater and damper showed lower CO_2 and higher O_2 content for No. 5 furnace at all ratings (see Fig. 23).

The curves in Figs. 21 and 22 cannot be regarded as true curves representing any very definite variation of gas composition across the boiler front wall but, rather, are merely used for purposes of convenience in indicating the furnace-gas com-

TABLE 5 FURNACE-GAS COMPOSITIONS—TEST C, No. 9 BOILER

(102 per cent of rating, normal fuel-bed thickness)

Period of sampling, p.m.		Composite Gas Samples — Sampling position 1	2	3	4	5	6	7	8
		Per cent							
1:00 to 2:00	CO_2	11.7	14.7	14.8	14.6	14.1	13.1	12.3	11.4
	O_2	4.7	0.5	0.1	0.5	2.4	2.6	6.2	7.2
	H_2	0.7	0.7	1.2	0.7	0.2	0.9	0.0	0.0
	CO	1.2	1.6	3.0	2.3	1.1	1.8	0.2	0.0
	N_2	81.7	82.5	80.9	81.9	82.2	81.6	81.3	81.4
	Free carbon	No	No	Yes	No	No	Trace	No	No
	Solution colored	No	Turbid	Yellow	Yellow	Brown	No	Yellow	No
	Oil film	No	No	No	No	Yes	No	No	No
2:30 to 3:30	CO_2	12.7	15.6	15.9	16.0	15.1	13.8	14.2	10.8
	O_2	3.6	1.4	0.8	0.9	1.4	2.7	2.8	7.9
	H_2	0.5	0.0	0.3	0.0	0.0	0.2	0.2	0.0
	CO	1.3	0.8	1.0	0.8	0.3	0.2	0.3	0.0
	N_2	81.9	82.2	82.0	82.3	83.2	83.1	82.5	81.3
	Free carbon	Trace	Yes	No	No	No	No	No	No
	Solution colored	No	No	Yellow	Yellow	Yellow	Yellow	Yellow	Yellow
	Oil film	No	No	No	No	Yes	No	No	No
4:00 to 5:00	CO_2	12.9	15.2	16.0	15.7	14.6	11.2	10.8	10.1
	O_2	4.8	1.0	1.6	1.7	3.2	7.0	8.3	8.0
	H_2	0.3	0.1	0.0	0.1	0.0	0.0	0.0	0.1
	CO	0.3	1.5	0.8	0.5	0.0	0.1	0.0	0.0
	N_2	81.7	82.2	81.6	82.0	82.2	81.7	80.9	81.8
	Free carbon	No	Yes	No	No	No	No	No	No
	Solution colored	No	No	No	Yellow	No	Yellow	Yellow	Orange
	Oil film	No	No	No	No	No	No	No	No

positions at certain significant sampling points at which gas samples were drawn simultaneously.

The uniformity in composition of the several sets of composite (hourly) gas samples taken as checks during each test, is evident, and indicates uniform operation of the boilers during the test. However, it would be expected that the furnace-gas conditions would vary considerably for short intervals from time to time, and this is confirmed by the snap samples taken simultaneously with a composite sample at No. 5 sampling position. The variation of these instantaneous samples is plotted against time in Figs. 24, 25, and 26, with the composite samples indicated by straight lines. As will be noted from Tables 4 to 13, the averages of the snap samples usually checked fairly closely with the composite sample drawn at the same time.

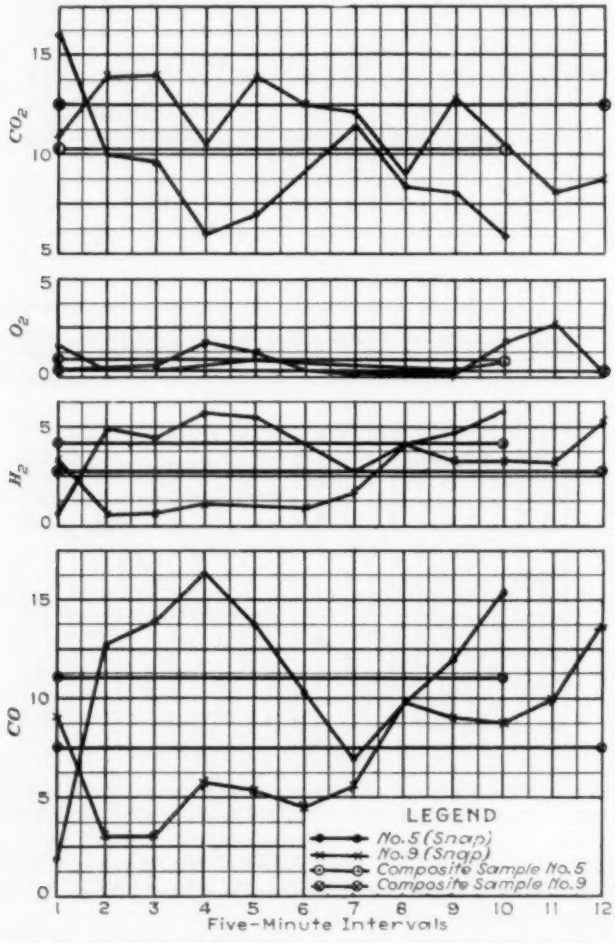


FIG. 26 FURNACE-GAS COMPOSITIONS, SNAP SAMPLES, NOS. 5 AND 9 FURNACES, 230 PER CENT OF RATING, NORMAL FUEL-BED THICKNESS

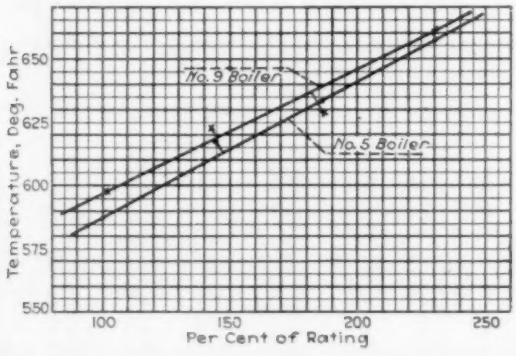


FIG. 27 VARIATION OF TOTAL STEAM TEMPEBATURES WITH RATING, NOS. 5 AND 9 BOILERS, NORMAL FUEL-BED THICKNESS

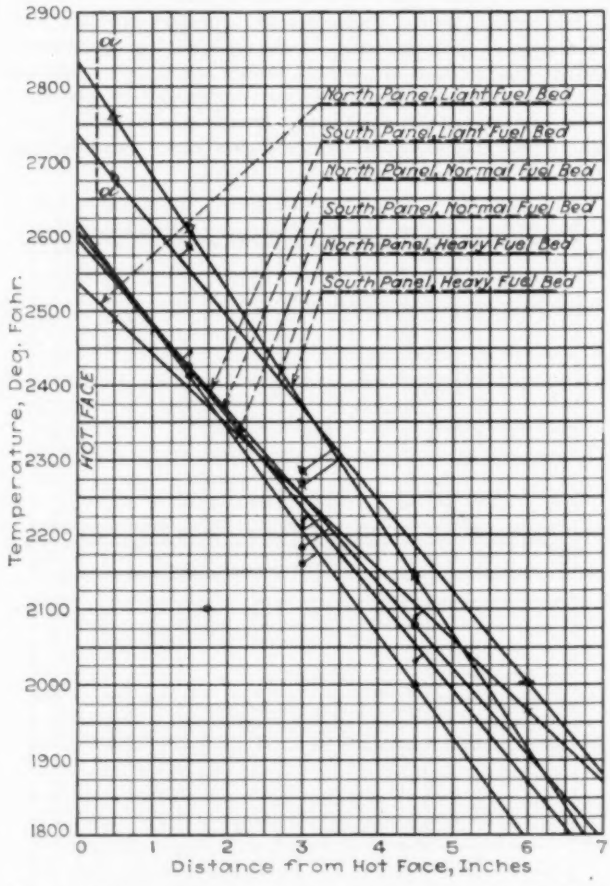


FIG. 28 TEMPERATURE GRADIENTS THROUGH WEST FRONT WALL, NO. 5 FURNACE, 145 PER CENT OF RATING, VARIOUS FUEL-BED THICKNESSES

(Broken line a–a indicates extent of erosion of wall at end of tests.)

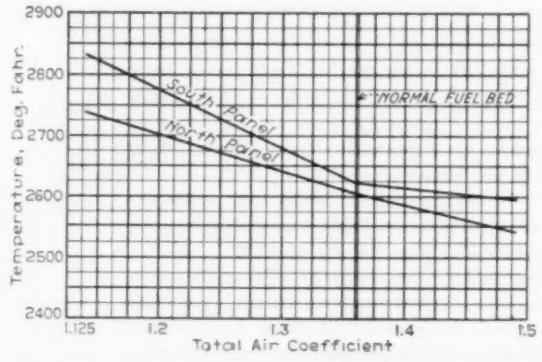


FIG. 29 EFFECT OF FUEL-BED THICKNESS ON INSIDE WALL-FACE TEMPERATURES (EXTRAPOLATED), NO. 5 FURNACE

Frequently carbon particles of colloidal dimensions were drawn over with the gas samples, and these were usually precipitated upon standing for a short time, forming a scum of soot over the saturated salt solution used as the collecting medium. The

collecting solution was at times discolored by certain substances, presumably tarry compounds distilled from the coal, which were collected during the sampling process. In certain instances, sufficient volatile compounds were carried over with the gases to form a film of oil over the salt solution.

Excess Air. It has already been mentioned that the superheater and damper gas compositions showed lower CO_2 and higher O_2 content in No. 5 than in No. 9 boiler at all ratings. These data are given in Tables 1 and 2, and plotted against boiler rating in Fig. 23. This condition was verified by the fact that the fuel bed in No. 5 tended to be somewhat lighter than in No. 9 furnace.

On checking those factors which might affect the distribution of air to the tuyères, as already noted, no differences were found in the plenum chambers, and although more tuyère stacks required renewal and repair following the tests in No. 5 than in No. 9 furnace, this factor would hardly seem to account for the difference. Of course, No. 9 boiler is baffled somewhat differently than No. 5, but other boilers at the station have the same baffling as No. 9, and behave somewhat like No. 5 boiler.

TABLE 6 FURNACE-GAS COMPOSITIONS—TEST A, NO. 5 BOILER

(145 per cent of rating, normal fuel-bed thickness)
Composite Gas Samples

Period of sampling, p.m.		Sampling position 1	2	3	4	5	6	7	8
		Per cent							
1:00 to 2:00	CO_2	10.8	10.0		14.8	12.8	12.6	8.5	14.5
	O_2	0.1	0.4		..	0.6	0.0	0.4	1.6
	H_2	4.5	4.6	(Sam-	..	2.2	2.1	6.8	0.2
	CO	8.0	9.4	pling	..	5.9	6.4	3.5	1.0
	N_2	76.6	75.6	tube	..	78.5	78.9	80.8	82.7
	Free carbon	Yes	Yes	broken)	Trace	Yes	No	Yes	No
	Solution colored	No	No		Yellowish	No	Yellow	Yellow	No
	Oil film	..	..		..	..	..	..	..
2:30 to 3:30	CO_2	11.2	9.5	.....	13.3	10.7	12.7	12.0	14.0
	O_2	4.8	0.3	.....	1.7	0.6	0.2	..	1.9
	H_2	3.6	5.6	.....	1.1	5.9	2.5	..	0.8
	CO	7.3	11.0	.....	4.1	4.9	6.5	..	1.0
	N_2	73.1	73.6	.....	79.8	77.9	78.1	..	82.3
	Free carbon	Yes	Yes	.....	Yes	Yes	Yes	Yes	No
	Solution colored	No	No	.....	No	No	No	No	No
	Oil film	..	..	.....	..	..	..	..	..
4:00 to 5:00	CO_2	10.7	9.8	.....	13.2	11.1	10.4	10.9	14.9
	O_2	4.7	5.6	.....	..	..	0.2	3.0	..
	H_2	3.6	3.9	.....	..	..	4.9	3.7	..
	CO	9.0	10.8	.....	..	..	6.6	2.9	..
	N_2	72.0	69.9	.....	..	..	77.9	79.5	..
	Free carbon	Yes	Yes	.....	Yes	Yes	Trace	Yes	No
	Solution colored	No	No	.....	No	No	Turbid	No	No
	Oil film	..	..	.....	..	..	..	..	..

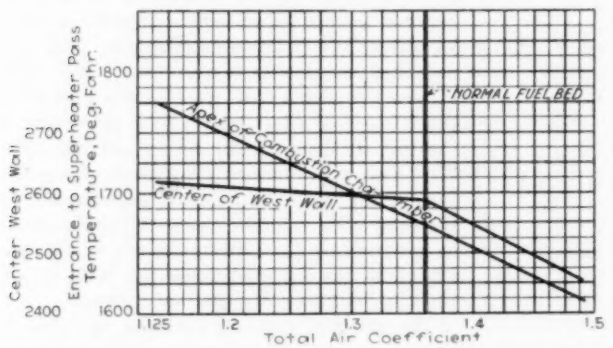


FIG. 30 EFFECT OF FUEL-BED THICKNESS ON APPROXIMATE FURNACE-GAS TEMPERATURES DIRECTLY OVER FIRES, NO. 5 FURNACE

Total Steam Temperatures. The total steam temperatures were higher in No. 9 than in No. 5 boiler, as indicated in Tables 1 and 2, and shown in Fig. 27.

Effect on Refractories. The effect of these tests on the boiler settings has already been described, and is evident from a careful inspection of Figs. 5 to 10, inclusive. Slag attack, shrinkage, and vitrification of the brick in the settings were obviously worse in No. 5 than in No. 9 furnace. Careful analysis of the steamflow-airflow meter charts since the furnace walls were last repaired showed that past load conditions averaged about the same for both boilers.

The severity of the furnace conditions during the tests may be deduced from the condition of the sillimanite sampling tubes. Several had been broken, either from having been struck by pieces of ash or slag, or from having been so weakened by slag attack as to be unable to support their own weight. As is evident from Fig. 8, the sampling tube at position No. 4 in No. 9 boiler had softened and sagged, and was eaten through by slag. Those tubes that had not broken during the tests were slagged about halfway through, with pinhole openings where the slag had completely penetrated, and were so fragile that they broke at a touch. The sillimanite tubes in No. 5 boiler were practically all broken off at the hot face of the wall during the tests, whereas most of those in No. 9 boiler were unbroken.

These factors are all the more striking when it is remembered that the sillimanite tubes are composed of super-refractory

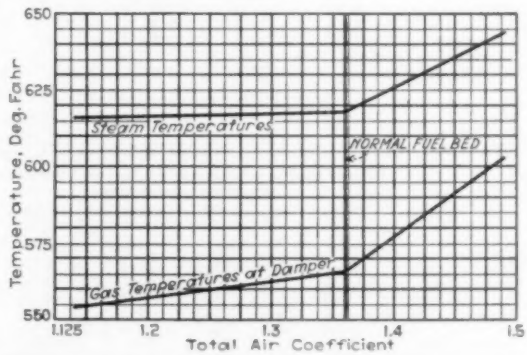


FIG. 31 EFFECT OF FUEL-BED THICKNESS ON TOTAL STEAM TEMPERATURE AND TEMPERATURES OF FLUE GAS AT DAMPER, NO. 5 FURNACE

TABLE 7 FURNACE-GAS COMPOSITIONS—TEST B, NO. 9 BOILER

(143 per cent of rating, normal fuel-bed thickness)
Composite Gas Samples

Period of sampling, p.m.		Sampling position 1	2	3	4	5	6	7	8
		Per cent							
1:00 to 2:00	CO_2	12.4	12.1	12.8	15.6	14.0	14.6	13.7	14.4
	O_2	1.5	..	..	..	..	5.7	0.4	..
	H_2	1.9	..	..	..	..	1.0	0.9	..
	CO	3.7	..	..	..	..	2.2	2.3	..
	N_2	80.5	..	..	..	..	76.5	82.7	..
	Free carbon	No	Yes	No	No	No	Yes	No	Trace
	Solution colored	No	No	No	No	Yes	No	Yes	No
	Oil film	..	..	..	..	..	..	..	..
2:30 to 3:30	CO_2	13.7	10.5	11.9	15.3	14.4	14.6	14.0	15.3
	O_2	2.6	..	0.1	4.3	0.5	0.4	0.2	..
	H_2	0.7	..	3.0	0.8	1.1	1.0	1.3	..
	CO	2.0	..	5.9	2.1	2.8	2.9	3.0	..
	N_2	81.0	..	79.1	77.5	81.2	81.1	81.5	..
	Free carbon	No	Yes	Yes	No	No	Yes	No	Trace
	Solution colored	No	No	Yes	Yes	No	No	Yes	No
	Oil film	..	..	..	..	..	..	..	..
4:00 to 5:00	CO_2	13.1	11.6	14.7	15.1	16.3	15.5	14.8	14.3
	O_2	..	4.9	1.9	6.2	0.2	..	0.2	7.9
	H_2	..	3.1	0.0	0.1	0.4	..	0.7	1.3
	CO	..	7.0	0.5	0.2	1.6	..	1.9	3.7
	N_2	..	73.4	82.9	78.4	81.5	..	82.4	72.8
	Free carbon	Yes	Yes	No	No	No	Yes	No	Yes
	Solution colored	No	No	Yes	No	No	No	Yes	No
	Oil film	..	..	..	..	..	..	..	..

TABLE 8 FURNACE-GAS COMPOSITIONS—TEST E, NO. 5 BOILER

(186 per cent of rating, normal fuel-bed thickness)

Composite Gas Samples

Period of sampling, p.m.		Sampling position 1	2	3	4	5	6	7	8
		Per cent							
1:00 to 2:00	CO_2	11.3	8.4		13.3	8.9	12.5	11.2	15.1
	O_2	0.2	0.3		0.6	0.2	0.1	1.6	0.5
	H_2	3.2	5.2	(Sam-	1.6	4.7	2.6	2.6	0.7
	CO	8.0	9.4	pling	5.2	15.8	7.3	8.0	2.2
	N_2	77.3	76.7	tube	79.3	72.4	77.5	76.6	81.5
	Free carbon	Yes	Yes	broken)	No	Yes	Yes	Yes	Yes
	Solution colored	No	No	.....	Yellow	No	No	No	No
	Oil film	No	No	.....	No	No	No	No	No
2:30 to 3:30	CO_2	10.9	10.3	.....	12.6	9.8	6.7	10.5	14.7
	O_2	0.1	0.2	.....	0.0	1.0	0.4	1.1	0.9
	H_2	3.8	4.7	.....	2.5	4.4	8.6	3.5	0.8
	CO	9.2	10.3	.....	7.1	11.4	20.8	10.0	2.2
	N_2	76.0	74.5	.....	77.8	73.4	63.5	74.9	81.4
	Free carbon	Yes	Yes	.....	Yes	Yes	Yes	Trace	Trace
	Solution colored	No	No	.....	Yellow	No	No	No	No
	Oil film	No	No	.....	No	No	No	No	No
4:00 to 5:00	CO_2	10.8	9.5	.....	9.9	3.9	6.1	10.3	14.9
	O_2	1.2	0.1	.....	0.1	1.0	0.1	0.9	1.4
	H_2	4.1	6.5	.....	4.2	8.0	7.6	4.1	0.3
	CO	8.6	13.3	.....	11.3	22.6	20.1	10.7	1.3
	N_2	75.3	70.6	.....	74.5	64.5	66.1	74.0	82.1
	Free carbon	Yes	Yes	.....	Yes	Yes	Yes	Yes	Trace
	Solution colored	No	No	.....	Yellow	Yellow	No	No	No
	Oil film	No	No	.....	No	No	No	No	No

Snap Samples

(Taken at Position 5 during second set of composite samples)

Sample No.	Time taken, p.m.	CO_2	O_2	H_2	CO	N_2
		Per cent				
1	2:30	9.8	1.5	...	10.1	..
2	2:35	9.8	0.4	...	12.0	..
3	2:40	9.6	0.2	...	13.2	..
4	2:45	8.8	0.4	...	13.7	..
5	2:50	11.0	0.5	...	8.5	..
6	2:55	9.4	1.8	...	11.0	..
7	3:00	9.2	0.4	...	11.6	..
8	3:05	9.6	0.2	...	11.4	..
9	3:10	14.2	0.7	...	4.3	..
10	3:15	10.4	0.3	...	10.5	..
11	3:20	11.6	0.2	...	9.4	..
12	3:25	9.5	0.3	3.7	10.6	75.9
Average		10.2	0.7	...	10.5	..
Composite sample, Position 5 (second set)		9.8	1.0	4.4	11.4	73.4

TABLE 9 FURNACE-GAS COMPOSITIONS—TEST D, NO. 9 BOILER

(187 per cent of rating, normal fuel-bed thickness)

Composite Gas Samples

Period of sampling, p.m.		Sampling position 1	2	3	4	5	6	7	8
		Per cent							
1:00 to 2:00	CO_2	12.3	9.5	11.5	14.5	12.4	12.6	13.0	13.5
	O_2	1.3	0.2	0.1	0.2	0.0	0.1	0.4	2.7
	H_2	2.3	6.2	3.6	1.4	2.5	2.4	2.5	1.1
	CO	5.2	10.8	6.1	3.5	7.1	6.7	5.9	1.6
	N_2	78.9	73.3	78.7	80.4	78.0	78.2	78.2	81.1
	Free carbon	Yes	Trace	Trace	No	Trace	Trace	Trace	Trace
	Solution colored	No	No	Yellow	Yellow	Yellowish	No	No	No
	Oil film	No	No	No	No	No	No	No	No
2:30 to 3:30	CO_2	10.4	11.0	12.7	13.7	13.1	12.8	11.4	13.9
	O_2	0.3	0.6	0.2	0.6	0.3	0.3	0.5	1.8
	H_2	5.0	4.2	2.8	1.2	2.1	2.3	2.8	0.7
	CO	9.3	8.9	6.2	4.6	5.5	6.1	6.7	2.0
	N_2	75.0	75.3	78.1	80.9	79.0	78.5	78.6	81.6
	Free carbon	Yes	Yes	Yes	Yes	Trace	Yes	Trace	Trace
	Solution colored	No	No	Yellow	Yellow	Yellowish	No	Yellow	No
	Oil film	No	No	No	No	No	No	No	No
4:00 to 5:00	CO_2	11.1	13.1	11.0	12.4	12.8	13.1	13.5	14.7
	O_2	4.9	1.2	0.0	0.1	0.2	0.4	0.4	1.6
	H_2	1.4	2.1	4.1	3.1	2.2	2.1	2.1	0.5
	CO	3.8	4.4	9.1	7.6	6.6	6.6	5.3	2.3
	N_2	78.8	79.2	75.8	76.8	78.2	77.8	78.7	80.9
	Free carbon	Trace	Trace	Trace	Trace	Trace	Yes	Yes	Trace
	Solution colored	No	No	No	No	No	No	No	No
	Oil film	No	No	No	No	No	No	No	No

Snap Samples

(Taken at Position 5 during second set of composite samples)

Sample No.	Time taken, p.m.	CO_2	O_2	CO
		Per cent		
1	2:33	13.3	0.6	8.1
2	2:35	12.4	0.5	7.6
3	2:40	15.5	0.8	2.2
4	2:45	15.6	1.6	0.8
5	2:50	...	..	..
6	2:55	12.3	1.1	4.5
7	3:00	14.0	0.5	4.1
8	3:05	14.0	0.4	4.0
9	3:10	9.2	0.4	11.9
10	3:15	10.4	0.2	10.0
11	3:20	11.0	0.7	9.4
12	3:25	9.5	0.4	12.3
Average		12.5	0.7	6.8
Composite sample, Position 5 (second set)		13.1	0.3	5.5

material of approximately 70 per cent alumina content, with a fusion point about 200 deg. fahr. higher than a first-quality clay firebrick. Moreover, the porosity of the tubes is very low, which should increase their resistance to chemical action and slag penetration.

One thermocouple brick from each boiler, and one adjoining brick from the same test panel, together with composite samples of the coal burned during the tests, were sent to the U. S. Bureau of Standards, Columbus, Ohio, for petrographic study of the effect of slag. Their observations will presumably be reported at a later date.

Tests With Abnormal Fuel-Bed Thicknesses

Temperatures. The temperature-gradient data (as given in Table 3) through the west front wall of No. 5 furnace at 145 per cent of rating with varying fuel-bed thicknesses are plotted in Figs. 28 and 29 and indicate exceptionally high furnace-wall temperatures with heavy fuel bed. It seems probable that much of the erosion of the brick in the test panels took place during the test with abnormally heavy fuel bed (test K). This would naturally influence extrapolation of the wall-face temperature, because the thermocouples would be nearer to the wall face, as indicated by the broken line *a–a* in Fig. 28. Of course, the slag coating on the brick would tend to counteract the effect of erosion.

The effect of fuel-bed thickness on furnace-gas temperatures at the center of the west front wall and at the apex of the combustion chamber as evidenced by optical-pyrometer readings at these points (given in Table 3) is plotted in Fig. 30, and the effect on steam and stack-gas temperatures is shown in Fig. 31.

Furnace-Gas Compositions. The gas-composition data on composite and snap samples drawn from No. 5 furnace at 145 per cent of rating with normal, light, and heavy fuel beds are given in Tables 6, 12, and 13, respectively, and are plotted in Fig. 32. As would be expected, the proportion of combustibles and reducing character of the furnace gases were greatly increased with heavy fuel beds.

Excess Air. Obviously, the proportion of excess air was increased by lighter fuel-bed thickness, and vice versa.

Steam Temperatures. As will be noted from Fig. 31, a higher total air coefficient (lower CO_2 in stack gases) resulted in higher total steam temperatures, while the stack-gas temperatures were also higher. Apparently, there is a balance point where the greater plant efficiency resulting from higher steam temperatures more than equalizes the heat losses due to greater volume of hotter stack gases. It is evident, also, that a higher total air coefficient will result in lower boiler operating temperatures and less strongly reducing furnace conditions, thus prolonging the life of the refractory settings.

Effect on Refractories. Tests J and K show conclusively that the thickness of the fuel bed

TABLE 10 FURNACE-GAS COMPOSITIONS—TEST G, NO. 5 BOILER

(230 per cent of rating, normal fuel-bed thickness)
Composite Gas Samples

Period of sampling, p.m.		Sampling position 1	2	3	4	5	6	7	8
		Per cent							
12:30	CO_2	11.0	3.5		14.4	0.5	0.0		13.1
to	O_2	0.0	0.1		1.0	9.9	0.1		0.3
1:30	H_2	3.1	10.5	(Sam-	2.1	7.4	13.3	(Sam-	3.2
	CO	9.7	24.5	pling	4.0	15.4	29.0	pling	7.1
	N_2	76.2	61.4	tube	78.5	66.8	57.6	tube	76.3
	Free carbon	Yes	Yes	broken)	Yes	Trace	No	broken)	Trace
	Solution colored	No	No	.....	No	No	No	.....	No
	Oil film	No	No	.....	No	No	No	.....	No
2:00	CO_2	9.7	1.1	.....	9.8	3.5	4.7	.....	0.9
to	O_2	0.0	0.1	.....	4.6	0.8	0.3	.....	0.2
3:00	H_2	4.8	13.7	.....	2.7	9.0	0.0	.....	13.0
	CO	11.8	28.8	.....	6.8	22.1	31.4	.....	28.9
	N_2	73.7	56.3	.....	76.1	64.6	63.6	.....	57.0
	Free carbon	Yes	Yes	.....	Trace	Yes	No	.....	Trace
	Solution colored	No	Yellowish	.....	No	No	No	.....	No
	Oil film	No	No	.....	No	No	No	.....	No
3:30	CO_2	8.8	0.4	.....		10.1	4.7	.....	2.3
to	O_2	0.2	0.2	.....		0.8	5.5	.....	0.3
4:15	H_2	5.9	10.7	.....	(Sam-	4.1	5.6	.....	8.9
	CO	12.6	31.9	.....	pling	11.1	11.9	.....	6.5
	N_2	72.5	56.8	.....	tube	73.9	72.3	.....	82.0
	Free carbon	Yes	Trace	.....	plugged)	Yes	No	.....	Trace
	Solution colored	No	No	.....		No	No	.....	No
	Oil film	No	No	.....		No	No	.....	No

NOTE: Third set of composite samples is over 45-min. period.

Snap Samples

(Taken at position 5 during third set of composite samples)

Sample No.	Time taken, p.m.	CO_2	O_2	H_2	CO	N_2
		Per cent				
1	3:30	16.2	1.5	0.3	1.7	80.3
2	3:35	9.9	0.1	4.9	12.9	72.2
3	3:40	9.7	0.1	4.5	13.8	71.9
4	3:45	5.9	0.5	5.8	16.4	71.4
5	3:50	6.9	0.7	5.6	13.8	73.0
6	3:55	...	...	...	...	...
7	4:00	11.4	0.5	2.7	6.9	78.5
8	4:05	8.3	0.4	4.2	10.1	77.0
9	4:10	8.2	0.4	4.7	12.1	74.6
10	4:15	6.0	0.8	5.9	15.3	72.0
Average		9.2	0.6	4.3	11.4	73.4
Composite sample, Position 5 (third set)		10.1	0.8	4.1	11.1	73.9

TABLE 11 FURNACE-GAS COMPOSITIONS—TEST H, NO. 9 BOILER

(230 per cent of rating, normal fuel-bed thickness)
Composite Gas Samples

Period of sampling, p.m.		Sampling position 1	2	3	4	5	6	7	8
		Per cent							
1:00	CO_2	13.4	10.4	12.4	14.2	14.4	15.1	15.3	13.4
to	O_2	0.6	0.1	0.0	0.4	0.0	0.0	0.3	4.3
2:00	H_2	3.0	4.1	2.4	1.8	3.1	2.6	1.0	0.3
	CO	4.6	10.9	8.0	4.3	3.5	3.5	3.1	0.8
	N_2	78.4	74.5	77.2	79.3	79.0	78.0	80.3	81.2
	Free carbon	Yes	Trace	Trace	Yes	Trace	Trace	Trace	Yes
	Solution colored	No	No	Yellow	No	No	Yellow	Yellow	No
	Oil film	No	No	No	No	No	No	No	No
2:30	CO_2	12.7	10.5	12.0	14.4	12.6	12.8	14.5	10.4
to	O_2	1.2	0.0	0.0	0.3	0.2	0.8	1.2	8.1
3:30	H_2	2.3	5.2	3.6	2.2	2.9	2.5	1.8	0.6
	CO	4.7	10.1	8.5	4.4	7.5	6.3	3.5	0.5
	N_2	79.1	74.2	75.9	78.7	76.8	77.6	79.0	80.4
	Free carbon	Trace	Yes	Trace	Trace	Yes	Trace	Trace	No
	Solution colored	No	Yellowish	Yellow	No	Yellow	Yellow	Yellow	Yellow
	Oil film	No	No	No	No	No	No	No	No
4:00	CO_2	6.4	6.8	9.2	5.3	11.2	9.5	13.5	10.9
to	O_2	5.6	0.2	0.3	9.5	0.0	4.1	1.8	6.8
4:30	H_2	4.4	7.5	7.6	2.6	3.6	2.4	0.1	0.1
	CO	8.5	15.8	9.8	5.8	9.3	7.0	2.6	0.6
	N_2	75.1	69.7	73.1	76.8	75.9	77.0	76.0	81.6
	Free carbon	Yes	Trace	Yes	Trace	Yes	Trace	Trace	No
	Solution colored	Yellowish	No	No	No	Yellow	No	Yellow	No
	Oil film	No	No	No	No	No	No	No	No

NOTE: Third set of composite gas samples is for a 30-min. period, and during this time changes took place in the character of the fuel bed, as evidenced by up to 1.7 per cent CO at the stack.

Snap Samples

(Taken at position 5 during second set of composite samples)

Sample No.	Time taken, p.m.	CO_2	O_2	H_2	CO	N_2
		Per cent				
1	2:30	10.8	0.3	3.1	9.4	76.4
2	2:35	13.8	0.3	0.6	3.3	82.0
3	2:40	13.7	0.4	0.7	3.2	82.0
4	2:45	10.3	1.7	1.2	5.8	81.0
5	2:50	13.9	1.2	1.1	5.4	78.4
6	2:55	12.5	0.3	1.0	4.6	81.6
7	3:00	12.2	0.2	1.7	6.6	79.3
8	3:05	8.8	0.0	4.2	9.9	77.1
9	3:10	13.0	0.0	3.3	9.2	74.5
10	3:15	10.1	1.6	3.4	8.8	76.1
11	3:20	8.2	2.7	3.2	9.9	76.0
12	3:25	8.6	0.2	5.1	13.8	72.3
Average		11.3	0.7	2.4	7.5	78.1
Composite sample, Position 5 (second set)		12.6	0.2	2.9	7.5	76.8

TABLE 12 FURNACE-GAS COMPOSITIONS—TEST J, NO. 5 BOILER

(145 per cent of rating, light fuel bed)

Composite Gas Samples

Period of sampling, p.m.		Sampling position 1	2	3	4	5	6	7	8
		Per cent							
2:00	CO_2	9.3			12.5	13.9	15.2		14.5
to	O_2	3.0			5.2	0.7	0.0		0.0
3:00	H_2	4.2	(Sam-	(Sam-	0.1	2.3	1.2	(Sam-	1.3
	CO	7.3	pling	pling	0.1	4.6	4.6	pling	3.7
	N_2	76.2	tube	tube	82.1	78.5	79.0	tube	80.5
	Free carbon	Yes	broken)	broken)	No	Yes	No	broken)	Trace
	Solution colored	No	.....	.....	No	No	No	.....	No
	Oil film	No	.....	.....	No	No	No	.....	No
3:30	CO_2	3.7	.....	.....	14.1	14.4	11.1	.....	13.3
to	O_2	13.3	.....	.....	3.4	0.3	3.2	.....	1.5
4:30	H_2	1.7	.....	.....	0.1	1.6	2.2	.....	1.5
	CO	10.7	.....	.....	1.6	5.0	5.6	.....	4.0
	N_2	70.6	.....	.....	80.8	78.7	77.9	.....	79.7
	Free carbon	Yes	.....	.....	Trace	Trace	No	.....	Yes
	Solution colored	No	.....	.....	No	No	No	.....	Yellow
	Oil film	No	.....	.....	No	No	No	.....	No
5:00	CO_2	7.5	.....	.....	13.8	13.7	9.4	.....	11.9
to	O_2	8.5	.....	.....	3.1	1.6	8.5	.....	3.2
6:00	H_2	1.9	.....	.....	0.8	1.2	0.5	.....	1.4
	CO	4.5	.....	.....	2.1	3.4	1.7	.....	4.2
	N_2	77.6	.....	.....	80.2	80.1	79.9	.....	79.3
	Free carbon	Yes	.....	.....	Trace	Trace	No	.....	Yes
	Solution colored	No	.....	.....	Yellow	No	No	.....	Yellow
	Oil film	No	.....	.....	No	No	No	.....	No

Snap Samples

(Taken at position 5 during second set of composite samples)

Sample No.	Time taken, p.m.	CO_2	O_2	H_2	CO	N_2
		Per cent				
1	3:30	12.9	1.2	1.7	6.1	78.1
2	3:35	15.5	1.1	0.7	1.9	80.8
3	3:40	13.5	0.6	1.6	5.8	78.5
4	3:45	12.7	0.8	2.1	5.9	78.5
5	3:50	13.5	0.4	2.8	5.9	77.4
6	3:55	12.7	0.6	2.3	6.6	77.8
7	4:00	13.6	0.7	1.7	5.1	78.9
8	4:05	14.5	1.0	1.1	3.1	80.3
9	4:10	12.3	0.5	2.4	7.3	77.5
10	4:15	11.3	0.8	3.0	8.7	76.2
11	4:20	11.2	0.5	3.2	9.5	75.6
12	4:25	14.2	2.9	0.2	0.7	82.0
Average		13.2	0.9	1.9	5.6	78.4
Composite sample, Position 5 (second set)		14.4	0.3	1.6	5.0	78.7

TABLE 13 FURNACE-GAS COMPOSITIONS—TEST K, NO. 5 BOILER

(143 per cent of rating, heavy fuel bed)

Composite Gas Samples

Period of sampling, p.m.		Sampling position 1	2	3	4	5	6	7	8
		Per cent							
12:00	CO_2	8.6				12.3	7.6		14.2
to	O_2	1.7				2.2	8.4		1.5
1:00	H_2	6.1	(Sam-	(Sam-	(Sample	1.7	4.9	(Sam-	2.2
	CO	9.2	pling	pling	contami-	4.1	0.0	pling	0.6
	N_2	74.4	tube	tube	nated)	79.7	79.1	tube	81.5
	Free carbon	Yes	broken)	broken)		Trace	No	broken)	Trace
	Solution colored	No	.....	.....		No	No	.....	Yellow
	Oil film	No	.....	.....		No	No	.....	No
1:30	CO_2	11.3	.....	.....	13.9	13.6	12.5	.....	14.4
to	O_2	0.0	.....	.....	2.9	0.0	0.0	.....	0.5
2:30	H_2	3.6	.....	.....	0.6	1.6	2.9	.....	1.6
	CO	7.3	.....	.....	1.1	4.1	7.1	.....	2.7
	N_2	77.8	.....	.....	81.5	80.7	77.5	.....	80.8
	Free carbon	Yes	.....	.....	Trace	Trace	Trace	.....	Trace
	Solution colored	No	.....	.....	Yellow	No	No	.....	Yellowish
	Oil film	No	.....	.....	No	No	No	.....	No
3:00	CO_2	10.1	.....	.....	13.9	11.6	11.6	.....	13.5
to	O_2	0.6	.....	.....	0.9	0.1	0.0	.....	2.1
4:00	H_2	5.0	.....	.....	1.9	3.2	3.8	.....	0.6
	CO	8.6	.....	.....	3.3	7.9	3.8	.....	2.7
	N_2	75.7	.....	.....	80.0	77.2	80.8	.....	81.1
	Free carbon	Yes	.....	.....	Yes	Yes	Yes	.....	Yes
	Solution colored	Yellow	.....	.....	Yellow	No	No	.....	No
	Oil film	No	.....	.....	No	No	No	.....	No

Snap Samples

(Taken at Position 5 during second set of composite samples)

Sample No.	Time taken, p.m.	CO_2	O_2	H_2	CO	N_2
		Per cent				
1	1:30	14.9	0.5	0.1	1.6	82.9
2	1:35	14.1	0.1	0.7	2.1	83.0
3	1:40	11.8	0.1	2.4	7.0	78.7
4	1:45	11.4	0.1	2.5	7.2	78.8
5	1:50	12.6	0.0	2.0	5.9	79.5
6	1:55	...	...	...	...	...
7	2:00	15.0	0.0	1.0	0.5	83.5
8	2:05	15.2	0.5	0.4	1.6	82.3
9	2:10	14.7	0.0	0.9	2.3	82.1
10	2:15	12.4	0.0	2.4	6.3	78.9
11	2:20	13.3	0.0	2.6	3.8	80.3
12	2:25	15.5	0.9	0.2	0.3	83.1
Average		13.7	0.2	1.4	3.5	81.2
Composite sample, Position 5 (second set)		13.6	0.0	1.6	4.1	80.7

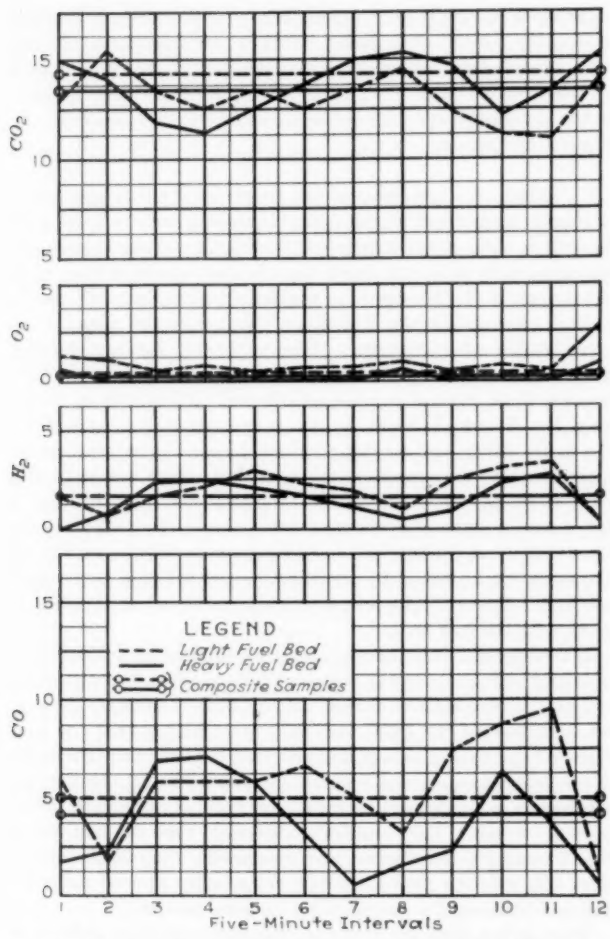


FIG. 32 FURNACE-GAS COMPOSITIONS, SNAP SAMPLES, NO. 5 FURNACE, 145 PER CENT OF RATING, VARIOUS FUEL-BED THICKNESSES

TABLE 14 COAL AND ASH DATA (DRY BASIS)

(The coal used in these tests was a 2-in. nut, pea, and slack, from the Island Creek seam, Rossmore Mine, Logan County, W. Va.)

Test No.	Volatile matter	Fixed carbon	Ash	Sulphur	Calorific value, B.t.u.	Melting point of ash, deg. fahr.
	Per cent					
F	35.00	59.20	5.80	0.87	14,150	2450
C	34.90	58.20	6.90	1.04	13,890	2420
A	34.90	58.20	6.90	0.97	13,940	2400
B	33.90	58.10	8.00	1.03	13,840	2400
E	34.80	58.60	6.60	0.90	14,060	2420
D	34.60	58.00	7.40	1.23	13,850	2400
G	33.50	59.50	7.00	1.09	13,910	2400
H	34.30	58.70	7.00	1.00	14,010	2400
J	34.50	57.80	7.70	1.18	13,910	2400
K	33.50	58.00	8.50	1.07	13,800	2440

has considerable effect on the life of the brick settings. However, it should be understood that the abnormal furnace conditions during tests J and K should never prevail for any appreciable length of time during regular operation.

The detrimental effect of abnormally heavy fuel bed on the refractories is very evident, in that the boiler temperatures were higher, the furnace atmosphere was severely reducing, and the slag[5] was far more fluid, greatly accelerating mechanical penetration and chemical attack. It seems probable that much of the erosion of the brick in the test panels took place during test K, since the slag on the walls was running throughout this test, the temperatures were very high, and the furnace gases were severely reducing. This is confirmed by the fact that the average temperatures at Nos. 1 and 6 thermocouples ($^1/_2$ in. from original hot face of brick in north and south panels) were considerably higher during the latter stages of the test, whereas the steam flow was lower.

[5] It is a generally accepted theory that the chemical attack of a coal-ash slag is more severe under reducing furnace conditions, because the iron present tends to be converted to the ferrous state, forming ferrous silicates which melt at lower temperatures than the corresponding ferric silicates obtained under oxidizing conditions.

CONCLUSIONS

At corresponding ratings, the operating conditions affecting refractories service are more severe in No. 5 than in No. 9 furnace; and particularly, the temperatures of the furnace gases and of the hot face of the wall in the lower part of the combustion chamber are higher in No. 5 than No. 9 furnace, the furnace gases are more strongly reducing, and the slagging action of the ash is greater. The evidences of these more severe conditions in No. 5 than No. 9 furnace are the greater shrinkage, vitrification, and erosion of the refractories.

Variation of the fuel-bed thickness has great bearing upon the life of the refractory brick. A heavy fuel bed produces a more strongly reducing atmosphere and higher furnace temperatures, which impose more severe conditions on the brick.

It is usually possible by increasing the proportion of excess air to reduce the severity of the conditions affecting the refractory settings. The extent to which this may be done at the expense of boiler efficiency, without decreasing plant efficiency, depends on the balance between heat losses up the stack and the increased plant efficiency resulting from higher superheated steam temperatures with convection-type superheaters.

ACKNOWLEDGMENT

Grateful acknowledgment for their many helpful suggestions is made to Messrs. W. A. Carter, Technical Engineer of Power Plants, The Detroit Edison Company, and J. Spotts McDowell, Director of Research, Harbison-Walker Refractories Company.

Discussion

DR. D. S. JACOBUS.[6] The Society is indebted to the authors of this paper for presenting the results of a series of tests made in the thorough and careful way that characterizes researches of the sort conducted by the Detroit Edison Company. The results are consistent in showing that placing the baffles which come nearest the furnace back of the third row of tubes in boiler No. 9 instead of back of the second row of tubes, as in boiler No. 5, led to lower furnace and wall temperatures and to lesser deterioration of the brickwork. This has been borne out in some of our own tests.

The falling off of the temperature as the distance from the faces of the walls increased shows how a firebrick wall may have enough strength in its cooler portions to allow it to be operated at near the fusing point of the brickwork at its inner face.

The difference in the degree of erosion of the walls in the two boilers demonstrates the importance of the temperature factor on the life of brickwork where corrosive slags are present. Even though there was apparently only about 50 to 100 deg. lower temperature on the wall faces in boiler No. 9 than in No. 5, the amount of slag erosion in the former was materially less, indicating that the rate of reaction increases out of all proportion to the increase in temperature.

The amount of excess air given in Tables 1 and 2 averages 17 per cent less for the normal fuel-bed thickness for the tests with the No. 9 furnace than for the No. 5 furnace. Where the

[6] Advisory Engineer, Babcock & Wilcox Co., New York, N. Y. Mem. A.S.M.E.

percentages of CO_2 are as high as were carried in the tests, lessening the amount of excess air may cause the flames to travel a greater distance. This may have been an additional element to that described in the paper in producing a higher gas temperature at the apex of No. 9 furnace than in No. 5 furnace.

The amount of CO in the gases drawn from the furnace and the very small amount of CO in the gases at the superheater and at the damper shows that the CO was burned in the upper part of the furnace. In tests that the writer made a long time ago at the Delray plant of the Detroit Edison Company on the same type of boilers it was found that there was a mingling action of the gases in the furnace which, combined with the large size and form of the furnace, led to an effective burning out of the combustible elements.

RALPH A. SHERMAN.[7] In the introductory chapter of the forthcoming bulletin of the Bureau of Mines, "A Study of Refractories Service Conditions in Boiler Furnaces," the following statement is made:

> Study of the data presented will show that conditions in a boiler furnace are extremely variable. It would be impossible, even if experimental investigations such as these were extended to a great number of furnaces of each type, to plot or tabulate more than an average of any service conditions, such as the temperature, gas composition, or velocity to be expected at any given rate of firing. Each furnace, even in the same plant, has its distinctive characteristics so that an individual analysis of special conditions will often be necessary.

The results of the investigation which the authors present in this paper fully confirm this statement. Although the work has been done in the thoroughgoing manner that characterizes all investigations conducted by the Detroit Edison Company, and although the existence of the different furnace conditions in two furnaces, apparently exactly alike, was clearly shown, yet the fundamental causes of these differences were not discovered.

An unusual stratification occurred in boiler No. 5, for although the gases at the front wall were more strongly reducing than in boiler No. 9, yet enough air entered elsewhere to complete the combustion before the superheater and, as the authors have pointed out, to give a lower content of CO_2 in boiler No. 5 than in No. 9. This stratification, with consequently delayed combustion, and the lower content of CO_2, are not in accord with the lower steam temperatures and gas temperatures at the damper in boiler No. 5 than in No. 9 and are to the writer the most puzzling features of the results.

The elaborate system for simultaneous sampling of the gases along the front wall was very good, but the writer wishes to call attention to the danger of using a refractory tube for the withdrawal of gas samples which may contain combustion gases. If free oxygen is present, the intimate mixing in the high-temperature tube will result in further combustion in the tube, and it is the writer's experience that even with a short distance of travel the gas collected in the sample holder may contain several per cent more CO_2, and less O_2, CO, H_2, and CH_4, than actually present at the point of sampling. In the absence of free oxygen, as in many of the samples of the investigation reported, there is opportunity in the hot refractory tube for the cracking of hydrocarbons, the deposition of soot, and the reduction of CO_2 by the incandescent carbon resulting in the collection of a gas with lower CO_2 and CH_4, and more CO and H, than at the point of sampling. By the use of a water-cooled sampling tube, the gas is quickly cooled to a temperature at which no reactions can occur, and the collection of a sample with the composition of that at the point of sampling is insured.

Many boiler plants have conditions similar to this where the refractories service is more severe in one furnace than in another apparently exactly the same. Similar investigations of this type, though not necessarily as elaborate, could not fail to lead to valuable information on the furnace conditions and, usually, to the determination and ultimate removal of the cause of the unusual conditions.

P. NICHOLS.[8] A reading of the paper is rather disappointing in that it stops short at the most interesting point. Like good detectives, the authors determined how the crime was committed, but apparently failed to arrest the criminal; perhaps they have left it open to the discussors to suggest solutions and will explain in their closure how the difficulties which were the cause of the investigation were overcome. They state that furnaces 5 and 9 were similar in all respects except that No. 9 had some advantage because three rows of tubes were exposed against two in No. 5; also, No. 5 had air backs on the east front wall. Apparently the latter difference can be eliminated as a factor because all measurements were made on the west wall, which was similar to that of No. 9. It would not be expected that the third row of tubes would make any material difference in the cooling of the front wall or on the fuel bed since, as shown by Professor Hottel, the radiation absorbed by the third row of tubes is very small, nor could the extra row of tubes appreciably affect the flow of gases near the fuel bed.

It is suggested that one more item of evidence would have been obtained by measuring the velocity of the gases at the west walls, but as it is, the authors proved that during the tests the average temperatures of the gases and walls were higher near the west face in No. 5 than in No. 9, and also that the CO in the gases was appreciably greater, but because the excess air in the flue gas was higher in No. 5, more air must have entered through the lower part of its bed. They conclude, therefore, that the fuel bed of No. 5 tended to be thicker, and presumedly they also conclude that this was the reason that its refractories showed more deterioration in regular operation previous to the tests.

They do not, however, give any explanation for this tendency to have greater thickness; presumedly the blame is not placed on differences in the human operating factor, but was caused by some peculiarity in the stoker. By a thicker fuel bed, do they mean for its whole length or was it only in the upper part? That would seem more probable as an average chance occurrence than that it should be thicker all the way down. The first explanations that one would offer for such a relative condition in No. 5 are that either the flow of air through the upper part was less for some unexplained reason or that the motion of the fuel was not the same because of some small difference in the mechanism—or it might be because No. 5 tended to get different size fuel on the average.

There is a large lack of, at least, published knowledge of the flow of the stream lines of coal in underfeed stokers. Some imagination of what this is and some ability to depict where combustion of the carbonized coal occurs are necessary to fix where the ash is released; this was of interest to us in our investigation on clinkering, but we could get very little help from operators or others.

It would be interesting to know whether, having settled that the main trouble was the thicker fuel bed, it has been possible to operate No. 5 since the tests in such a manner that the refractories give no more trouble than occurs in No. 9.

AUTHORS' CLOSURE

The authors would like to point out that this investigation was begun originally as the study of an operating problem;

[7] Fuel Engineer, Battelle Memorial Institute, Columbus, Ohio.

[8] Supervising Fuel Engineer, Pittsburgh Experiment Station, U. S. Bureau of Mines.

namely, a noticeable difference in performance between two boilers which should behave alike. The first step was to discover the character and magnitude of these differences in operation, as reported in the paper; and the second step would be to remove such differences. In the meantime, however, other conditions arose which made it impractical to carry the tests to a logical conclusion.

Although, as Mr. Nichols points out, the criminal was not apprehended, some of the methods followed in pursuing him may be of interest to other central-station operators. All data have been reported in very complete form, with the hope that others who have observed similar differences in operation between units which should behave alike may be able to offer helpful comments.

Mr. Sherman mentions the possibility of change in the composition of the gas sample during its passage through the hot end of the refractory tube, due to catalysis or to more intimate contact between combustibles and unconsumed oxygen. On the other hand, it has been claimed that the introduction of a water-cooled sampling tube into a furnace may so chill the gases in its vicinity, or otherwise alter the combustion conditions, that the gas sample collected through it may not be strictly typical. In order to compare the two methods of sampling, composite gas samples were drawn for a 45-minute period, directly over the fire in No. 12 boiler, through sillimanite and water-cooled sampling tubes whose ends were within $^1/_2$ in. of each other. The analyses of these samples, in percentages, are shown in the table.

These analyses may be considered to be "checks" within the normal experimental variations of sampling and analysis.

This paper will have served its purpose if it describes a method of studying boiler-furnace operating conditions, and shows the differences which may exist between units which should normally function similarly.

	Sillimanite sampling tube	Water-cooled sampling tube
Sample A_5:		
CO_2	13.9	14.1
O_2	2.3	2.4
H_2	1.1	0.9
CO	2.9	3.0
CH_4	0.1	0.1
N_2	79.7	79.5
Sample A_4:		
CO_2	14.5	14.6
O_2	1.8	1.7
H_2	0.8	0.4
CO	1.7	1.7
CH_4	0.3	0.2
N_2	80.9	81.4
Sample A_1:		
CO_2	11.9	12.3
O_2	4.6	4.5
H_2	0.3	0.2
CO	0.2	0.0
N_2	83.0	83.0
Sample A_3:		
CO_2	13.9	14.1
O_2	0.3	0.3
H_2	1.2	0.9
CO	2.5	3.0
N_2	82.1	81.7
Sample A_2:		
CO_2	14.0	14.2
O_2	0.4	0.7
H_2	1.7	1.3
CO	0.8	1.6
N_2	83.1	82.2

FSP-53-25

South Amboy Plant of the Jersey Central Power and Light Company

By R. C. ROE[1] and J. P. MAILLER,[2] NEW YORK, N. Y.

The Jersey Central Power & Light Company, an operating unit in the Middle West Utility System, has just put into operation a new plant on the Raritan River at South Amboy, N. J., designed and constructed by the Electric Management & Engineering Corporation of New York City. This plant at present consists of two 25,000-kw. vertical-compound turbines with condensers and all auxiliaries and three 280,000 lb. of steam per hr. boilers, furnaces, and accessories, operating at 1400 lb. pressure. Space has been provided for the installation of a third turbine within the station which is approximately 168 ft. square and 110 ft. high to the boiler-room roof and is constructed along modern architectural lines.

The plant is located at the geographical center as well as the load center of the system and will, when the transmission lines are completed, serve as a base-load plant for both the Northern and Southern divisions, which at the present time are operated as independent divisions.

THE UNUSUAL features of the South Amboy station of the Jersey Central Power and Light Company are:

1 The use of outside concrete cylindrical-type bunkers which allows extremely light and airy boiler room and firing aisle.

2 The use of unit boiler-turbine construction, one boiler per turbine with a spare boiler applicable to either turbine, which eventually will be used for the third turbine.

3 The use of double-deck, compound turbines as a single unit without provision for separate operation.

4 The use of condensers designed for higher vacuum than has been customary.

5 The use of a five-point regenerative cycle with direct-contact heaters on all stages, eliminating drip pumps or condensate cascading and bringing terminal differences practically to zero, with expected improvement in overall thermal efficiency.

6 The use of premeasuring devices for coal and air, and individual mills per burner with multiple burners per boiler, permitting great flexibility and accurate predetermination of combustion conditions prior to the actual combustion.

A complete description of the plant appeared in the July 15, 1930, issue of *Power Plant Engineering*. A brief description of the station and its preliminary operation is as follows:

Coal and Ash Handling and Pulverization

Coal may be procured in the station by rail or water, being handled by rail by a balanced skip hoist through a Bradford breaker and a belt conveyor to the station bunkers and by water through a Mead-Morrison barge unloading equipment to the same Bradford breaker and belt conveyor. Arrangements are also made for the storage and reclamation by drag scrapers.

The station is unique in the fact that it has no inside bunkers nor inside storage of coal, all active coal storage being in six outside concrete bunkers or silos, two per boiler, each of 125 tons. From each of these concrete bunkers, three spouts are taken, two spouts leading to individual pulverizers and the third spout to the corresponding spout from the adjacent bunker to feed a common pulverizer. By this arrangement the two bunkers serve five pulverizers, all of the ball-mill type. Each pulverizer is equipped with its own primary fan and burner, there being five burners for each furnace.

Coal is ground to a fineness of 88 per cent through 200 mesh, according to the manufacturer's guarantees. Ash is removed from the furnace in a molten state by the usual slag-tapping method into an Allen-Sherman-Hoff conveying system.

Air and Gaseous Products of Combustion

Air is taken from the top of the boiler room through a pair of forced-draft fans, a tubular air preheater and metering devices

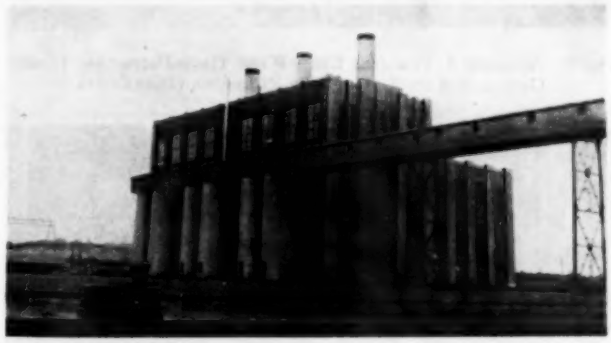


Fig. 1 Exterior of South Amboy Plant of the Jersey Central Power and Light Company

to the pulverizers and burners. The burners on two of the boilers are of the Calumet type and on the other boiler are of the Roe type. The products of combustion pass through an eight-tube boiler section, the initial superheater, the gas section of the steam resuperheater, economizer, and air preheater, and hence through connecting ducts to twin induced-draft fans which are mounted in the base of a venturi stack. Provision is made for the removal of soot by compressed-air soot blowers and removal of accumulated soot in the hopper between the economizer and air heater by the Allen-Sherman-Hoff conveying method.

Steam and Water Cycle

Steam leaves the boiler drum through the distributing tubes to the superheater which is built as an integral part of the boiler and immediately above the boiler-tube sections. It then goes through the desuperheater and main steam piping to the high-pressure turbine throttle, thence through the high-pressure turbine and intermediate piping to the steam to steam reheater, thence to the gas-steam reheater and then to the low-pressure turbine, after passing which it goes to the condenser, which is of the divided water-box, single-pass type with tubes rolled at both ends and arranged for extremely high vacuum.

---

[1] Design Engineer, Electric Management and Engineering Corporation.

[2] Chief Engineer, Electric Management and Engineering Corporation. Mem. A.S.M.E.

Presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Steam for the regenerative cycle is extracted as follows: From the fifth stage of the high-pressure unit for No. 5 heater, from the exhaust of the high-pressure unit for No. 4 heater, from the first stage of the low-pressure unit for No. 4 heater, from the seventh stage of the low-pressure unit for No. 2 heater and from the twelfth stage of the low-pressure unit for No. 1 heater. The evaporator operates in parallel with No. 3 heater and discharges condensate into No. 3 heater. The heater system, it is believed, is different from any other now installed, in that all bleeder heaters are of the direct-contact type arranged for coordinated

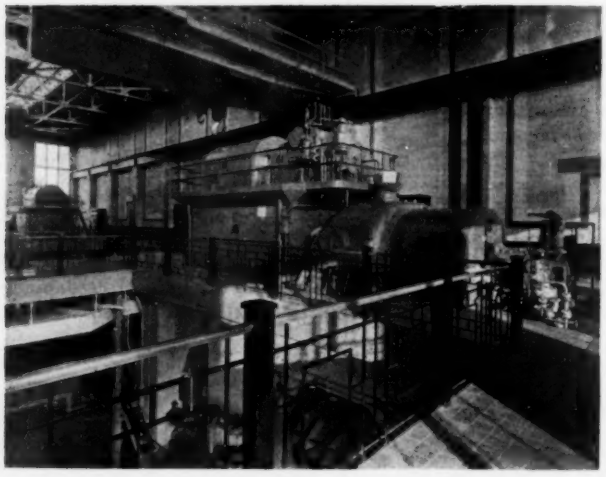


FIG. 2 NUMBER 3 TURBINE UNIT WITH HIGH-PRESSURE TURBO-GENERATOR ON TOP OF LOW-PRESSURE GENERATOR

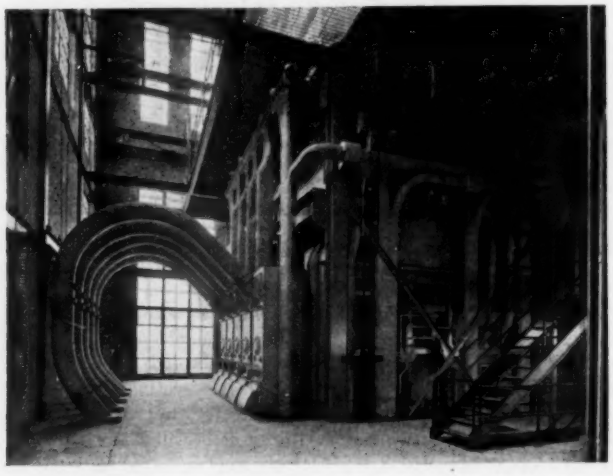


FIG. 3 NUMBER 3 BOILER—NOTE ABSENCE OF BUNKER AND LIGHT-FIRING AISLE

water and steam flow with specially designed pumps which will be described later. Provision is made for the proper air removal and deaeration by the venting of each heater through a vent condenser to the next preceding heater and finally from No. 1 heater to the main condenser at a point suitable for air removal. The condensate is taken from a storage-type hotwell at the bottom of the condenser, supplemented by an outside condensate storage for overflow and renewal purposes, by means of the first section of a three-section, variable-speed condensate pump, thence from the first- and second-stage condensers for the steam-jet air pumps, generator air cooler, oil cooler, and to No. 1 heater. From No. 1 heater, the condensate flows to the second section of the condensate pumps, then through the third-stage condenser to the steam-jet air pumps to No. 2 heater; from No. 2 heater the condensate flows to the third section of the condensate pumps, through the evaporator condenser and thence to No. 3 heater. From the storage tank of No. 3 heater, the condensate flows to the first section of a variable-speed two-section booster pump and thence to No. 4 heater. The condensate then flows to the second section of the booster pump and to No. 5 heater, and from the storage tank of this heater the condensate is taken by a constant-speed boiler feed pump through the economizer to the boiler.

The condensate and booster pumps are designed for and are controlled to give the required head and capacity between their respective heaters to accommodate varying loads on the turbine and in accordance with the bleed-point pressures, which pressures are the highest on No. 5 heater and taper down to the lowest on No. 1 heater and also vary in accordance with the load on the turbine. The boiler-feed pumps take condensate from the tank of No. 5 heater, and because of the change in heater pressure due to change in load, this pump is designed with special head-capacity characteristics for constant-speed operation, giving practically constant boiler-feed pressure throughout the load range without the use of regulating valves or other devices to accomplish this purpose.

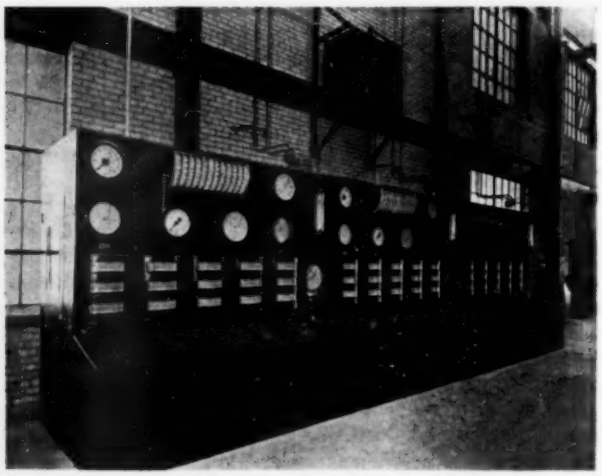


FIG. 4 BOILER-ROOM CONTROL BOARD, WITH THREE-DIAL COAL AND AIR METERS

STEAM TEMPERATURE CONTROLS

The initial steam temperature is controlled by the automatic operation of the desuperheater compensated to maintain 750 deg. at the turbine throttle by means of a butterfly bypass valve operated by thermostatic temperature control. The desuperheater takes its water from and returns the steam evaporated to the corresponding boiler drum. The steam for the low-pressure unit is maintained at 750 deg. at the throttle somewhat similarly by the regulation of the amount of high-pressure steam to the steam to steam reheater.

METERS

All the boiler-room meters, indicators, and controls are grouped on a centralized control board located in the center of the boiler room where the operator can see all the boilers.

An interesting feature of this board is the coal and air meters. The coal fed to each burner is weighed individually and indicated in pounds per hour and per cent of burner rating for varying fuel-heat contents on these meters. Likewise, in the same meter

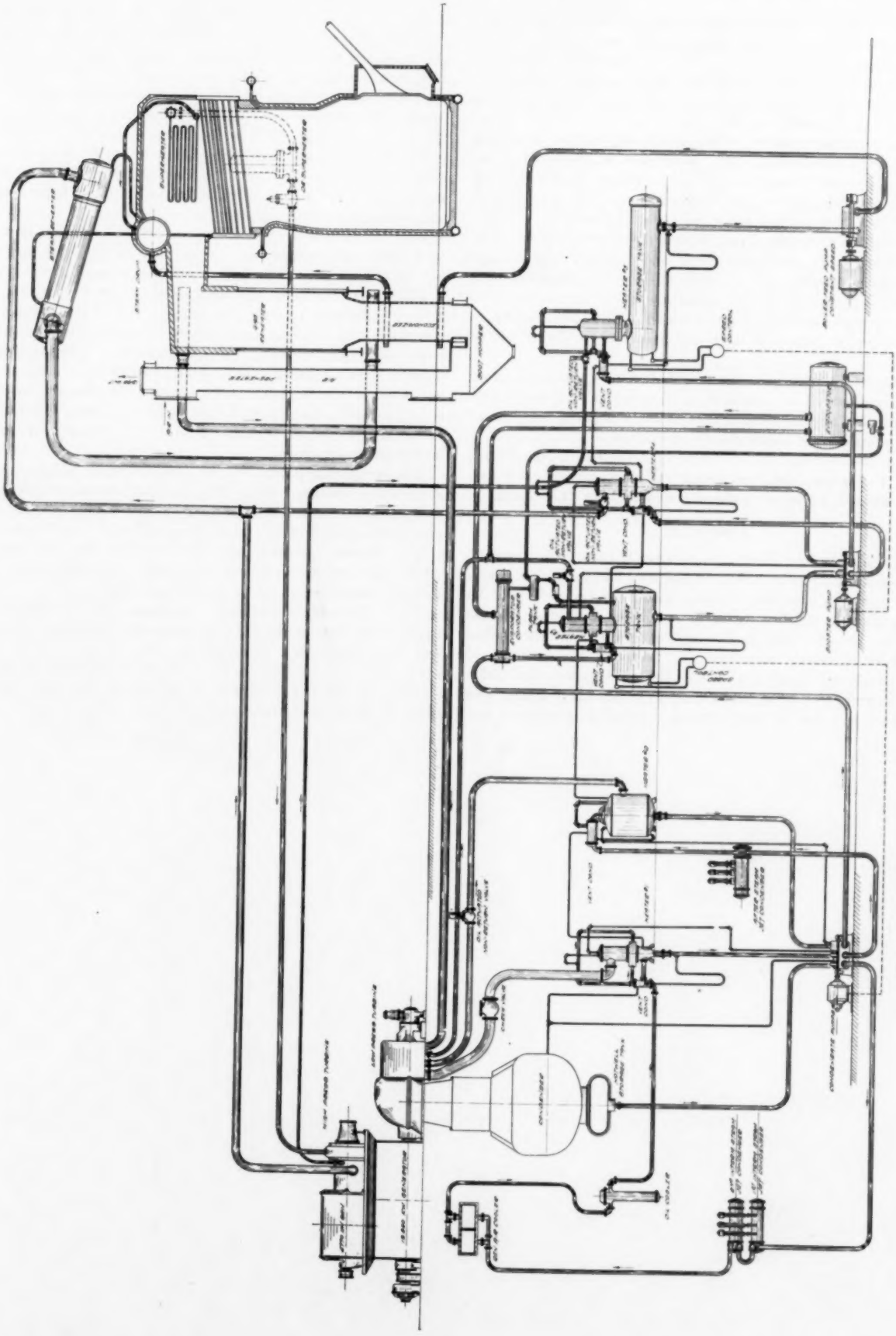


FIG. 5 SCHEMATIC DIAGRAM OF STEAM AND WATER CYCLE AT SOUTH AMBOY STATION

sections are incorporated the primary and secondary air quantities in per cent of burner rating and in pounds per hour, regardless of the temperature of the air. This arrangement permits an accurate preproportioning of the ingredients necessary for correct combustion prior to entering the furnace which many engineers believe to be a logical method.

For a permanent record of the entire combustion, recording CO_2 meters are also installed. Feedwater is measured in an orifice-type meter compensated to give accurate results with the varying temperatures of the feedwater. For the guidance of the operator, the water level of the boiler is shown on the control board by a remote liquid level indicator. In addition to these instruments, the usual draft and pressure gages, also temperature recorders, are mounted on this board, together with the control push buttons and indicating lights for properly controlling combustion. Also on this board is located the manually actuated remote control of the boiler feedwater to the individual boilers.

In the turbine room is provided a turbine control board on which are mounted the usual instruments, and in addition thereto the recording and controlling meters for the initial and reheat steam temperatures as previously mentioned.

The main electrical control board is of the bench-board type, one of the unusual features being the development of a new flush-mounted, indirectly lighted instrument for this board.

Operation

The South Amboy Plant is, as of Dec. 1, 1930, in its preliminary operating stage in a commercial way and has not had sufficient operating experience so that much can be said about its operating characteristics, except as covered by experience to date.

At the present time, both turbines and two of the boilers have been operated under load, and the plant has carried load continuously for about a month. The starting troubles which have occurred to date in this station have been very minor, and have been less than is usually to be expected in starting a new station. A total of two small gaskets were lost in the entire starting operation and one valve-bonnet gasket. There was no joint failure of any of the main steam or reheat lines or any trouble with these steam lines. Two cases of tube leakage occurred in the boilers in their seats, one of which has been repeated several times. One of these cases was due to improper rolling on erection and the other case was due to failure to provide properly for expansion stresses in the design. These were two cases of minor nature, which is a very small percentage considering the large number of rolled joints involved in one of these boilers. No major troubles are anticipated from this source.

Ignition burners were provided for each coal burner and these have operated satisfactorily. They have been a very material help in eliminating the difficulties of the starting period.

There was no serious trouble with the boiler-feed pumps, although experience to date does indicate that slight changes are necessary, and steps are being taken to modify one set of these boiler-feed pumps to make it slightly less sensitive and to provide a better packing arrangement. It is probable that similar steps may need to be taken in the other pumps. This matter, however, is well in hand, the pumps having operated satisfactorily to the extent that there has been no difficulty in putting water in the boilers whenever desired.

The circulating pumps on the condensers are equipped with synchronous motors. Some difficulty has been experienced with the pull-in torque of these motors. This is being corrected now, and in the meantime the station is not suffering from this feature.

The turbine operation has been excellent, the only trouble which has occurred being one case where too much or too little lubrication was put on the valve mechanism and it had a tendency to bind. No trouble has been experienced from the growing of parts in this mechanism and the operation of the turbine governor has been very good. The freedom from this difficulty is probably due to changes made, based on the experiences of others. Beyond the usual adjustment of apparatus, this has been the total of the plant-starting troubles.

FSP-53-26a

Combined Heat and Power Supply in Industrial Plants

By W. F. RYAN,[1] BOSTON, MASS.

The ever-increasing demand for cheap power—the foundation of American industry—has been ably met by the public utilities in the past, and the industrial power engineer is now taking his part in promoting the productiveness of human labor.

The use of high steam pressures has been the most noteworthy development in the industrial power field during the past few years. There are now about 200 industrial plants in the United States operating at pressures above 300 lb. gage, almost all of which have been built since 1925. The industrial plant which has a large demand for process steam has a natural advantage over the condensing central station, in that the latent heat in the exhaust is utilized, and the cycle efficiency of the prime movers is 100 per cent. The quantity of power which can be generated so economically is limited by the demand for exhaust steam, but can be materially increased by raising the throttle pressure.

There are few industries which can generate their entire power requirements, at all times, in all seasons, as a by-product of their process steam. There are still many such plants which generate power in condensing prime movers, but the vast majority purchase their requirements from public utilities. There is rarely any economic justification for the private generation of steam power in condensing units, although under certain conditions internal-combustion engines may be used advantageously. There is a growing tendency in this country to use the Diesel engine as an adjunct to an industrial steam-power plant; when the electric-power demand exceeds the relative demand for exhaust steam, the deficiency is supplied by the Diesel engine.

There are many plants which have large demands for process steam, but relatively little demand for electric power. The most significant trend in the industrial power field today is the cooperation of public utilities with manufacturing plants to develop "by-product" power to its economical limit. An example of such cooperation is cited in which the power output of an industrial plant is increased from 5500 kw. to 25,500 kw., the surplus power being distributed to other consumers, over the transmission lines of the public utility. The fuel cost of the additional 20,000 kw. is less than 4500 B.t.u. per kw-hr. The combination results in a decrease of 10 per cent in capital cost and a decrease of 20 per cent in fuel consumption, as compared with the separate generation of electricity by the public utility and of power and heat by the industrial plant.

Gratifying progress has been made in the conservation of fuel by developing the high-temperature end of the industrial power and heat cycle. An enormous quantity of potential energy is still being dissipated in the relatively low-temperature wastes from our manufacturing plants. For the present, the high-temperature field offers more attractive return on investment, but the future will undoubtedly see much progress in the recovery of heat at low temperatures. This source of energy will become economical long before such projects as the generation of power from the tides, which have greater appeal to the imagination, but which inherently involve enormously greater capital outlay.



THE development of industrial heat and power supply in the United States is an outstanding illustration of Henry Adams' dynamic theory of history; that is, progress has been proportional to the square of the time.

From the early New England water wheels to the advent of the Corliss engine, progress was continuous, but at so slow a rate that it was almost imperceptible to contemporary engineers. The economical prime mover which George Corliss developed gave the industrial power plant its first notable impetus, and by the end of the nineteenth century the use of steam power was almost universal in American industry.

At the beginning of this century the typical large manufacturing plant had a number of small hand-fired boilers, frequently distributed among several isolated boiler plants, in order to produce steam as near as possible to the point of consumption. Reciprocating engines, driving lineshafts by means of belts and rope drives, were also widely distributed throughout the works. Frequently these engines were run condensing, and live steam was used for heating and processing. In other words, heat and power supply were not effectively combined, but were essentially separate services.

Electrification of mills began a little later, together with centralization of boiler plants and the substitution of purchased power or turbine generators for the old engine drives. The use of exhaust steam for heating and processing became widespread, and condensing prime movers began to disappear. By 1925 most of our larger industries possessed efficient boiler plants, and great improvement had been made in the economy of steam use.

[1] Mechanical Engineer, Stone & Webster Engineering Corporation, Mem. A.S.M.E. Mr. Ryan was born in Woodbury, Conn., in 1889, and was educated in the public schools of Everett, Mass., and at Harvard University. His professional experience has been almost exclusively in steam-power engineering, as prior to assuming his present duties he has been employed in work of this kind by the Interborough Rapid Transit Company, the Wright-Martin Aircraft Corporation, the Harry M. Hope Engineering Company, and by subsidiaries of the Allied Chemical & Dye Corporation.

Contributed by the Power Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Communicated to the Second World Power Conference, Berlin, Germany, June, 1930.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Since 1925, the development of the industrial power plant has been truly remarkable. Higher pressures, higher temperatures, and higher rates of heat transfer have brought about even greater changes in the industrial steam-power plant than they have in the central station. Higher pressures and temperatures have materially increased the yield of "by-product power" from process steam, while higher rates of heat transfer have more than offset the increases in capital cost of plant which higher pressures and temperatures would otherwise have produced.

High-Pressure Steam in the Industrial Plant

In 1925, the author discussed the advantages of higher steam pressures in the industrial plant.[2] It was pointed out that

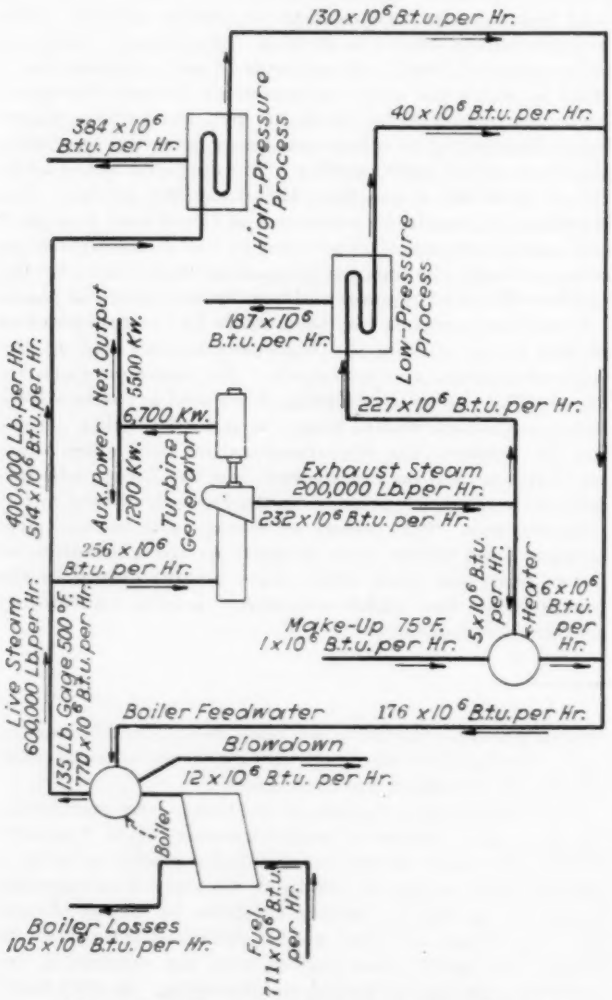


Fig. 1 Heat Balance, Isolated Industrial Plant

whereas the efficiency of a central station was increased only 10 per cent by increasing boiler pressure from 400 lb. gage to 1200 lb. gage a similar increase in pressure would more than double the effectiveness of non-condensing prime movers supplying high-back-pressure steam to a manufacturing process. The mechanical difficulties of generating and using very high steam pressures had been solved by the public utilities, and our industries were not slow to appreciate the value of this new development in their own plants. Early in 1929 an American technical magazine[3] was able to list 141 industrial plants using steam pressures above 300 lb. gage. Only two of these high-pressure plants had been installed prior to 1924.

The majority of high-pressure non-condensing plants have proved thoroughly reliable in operation. The turbine wheels are small in diameter and the blades are short, so that in spite of high speed, usually 3600 r.p.m., the stresses are low and the availability factor high. Many of these prime movers have never been taken out of service except for routine inspection and repairs.

Thanks to the pioneer work of central-station designers and operators, no unusual difficulties have been experienced with steam-generating equipment except when feedwater conditions have been unfavorable. In many industries the process steam cannot be returned as condensate, and the use of treated water has proved very troublesome, if not impracticable, in the higher-pressure plants. This condition may compel the use of an intervening evaporator condenser between the prime mover and the process-steam supply, or it may compel the use of very extensive evaporators for make-up water. The former increases the capital cost of the plant and reduces the yield of by-product power; the latter increases capital cost and operating expense. Even under these unfavorable conditions, however, a high-back-pressure non-condensing power plant will ordinarily produce electric power at a cost that is gratifying to the owner.

Where the amount of power required is large in proportion to the demand for exhaust steam, the reciprocating engine is advantageous. In capacities up to 5000 kw., a non-condensing engine will consume less steam per unit of output than a non-condensing turbine. The thermal efficiency of the engine is less, because radiation and external friction are greater. All of the other losses in both types of prime mover result in higher heat content of the exhaust, and no heat is actually lost thereby. These losses degrade the energy, however, and from this point of view the engine is superior.

The steam engine is in common use for driving low-speed pumps, compressors, and other variable-speed machines in industrial plants. In these applications, better efficiency of the driven unit, as well as lower water rate of the prime mover, favor the use of the reciprocating machine.

The cost of electric power in a high-back-pressure industrial plant is largely a matter of bookkeeping, because of the lack of uniformity in dividing fixed charges and operating expense between the two items of electric power and process steam. In the larger plants, the cost should not exceed 5 mills per kw-hr. Much lower costs have been reported, one as low as 1.5 mills per kw-hr., but it is probable that these very low costs are the result of some arbitrary method of accounting in which the process steam carries more than its proper share of operating expense, or they may be due to a neglect of capital charges. The latter are rarely included in published power costs, either by industrial plants or central stations.

Cooperation of Public Utilities With Industries for Combined Production of Steam and Power

While the advantages of private power generation in high-pressure prime movers are obvious, and are being convincingly demonstrated, very few industries are capable of developing this idea to its economic limit. The big power-consuming industries—for example, automobile manufacture—are not great consumers of process steam; frequently a seasonal heating load is the only steam demand. On the other hand, the big consumers of exhaust steam—oil refineries, salt works, alkali plants, etc.—are relatively small consumers of electric power. Moreover,

[2] "The Value of Higher Steam Pressures in the Industrial Plant," W. F. Ryan. Trans. A.S.M.E., vol. 47 (1925), p. 779.

[3] See *Power*, May 28, 1929.

there is frequently a regrettable lack of synchronism between the demands for these two forms of energy. For example, in many industries the greatest single item of power consumption is for pumping cooling water; this load is greatest in the summer months. An important item of steam consumption is usually the heating of buildings; this demand occurs only in the winter. There are other variations between summer and winter, between day and night, and between Sundays and week days that make private generation of power unduly expensive in most industries, and impracticable in many.

The full economic value of the high-back-pressure prime mover can be developed only by cooperation between industries and public utilities for the combined production of process steam and electric power. Cooperation of this type has been discussed academically for some time, but it is only now beginning to find practical application. This step is the outstanding development in the field of industrial power and heat supply today.

As an example of the possibilities of cooperative steam and power production, consider a relatively large industrial plant, having an average steam demand of 600,000 lb. per hour. Two-thirds of this steam is used at boiler pressure, and the remainder at a lower pressure, after passing through non-condensing turbine-generators. The power demand of the mill, which is only 5500 kw., is easily and economically supplied in this way. Fig. 1 shows the essential features of the plant heat balance, with heat flows given to the nearest million of heat units, above 32 deg. fahr. Steam is generated in an efficient boiler plant; the entire electric-power requirements are generated in non-condensing prime movers with a cycle efficiency of 100 per cent and a thermal efficiency of about 95 per cent. The whole installation approaches the limit of economical heat and power supply for this particular industry.

The territory in which this plant is situated is served by a central station, which is also highly efficient. The essential features of its heat balance are outlined in Fig. 2. Steam is supplied to the turbine throttles at 625 lb. gage and 750 deg. fahr. The turbines exhaust at a vacuum of 29 in. hg. There are four stages of regenerative feedwater heating, and the boilers are as efficient as those of the lower pressure industrial plant. The overall fuel consumption is at the rate of less than 13,000 B.t.u. per kw-hr. generated, and less than 1 lb. of good steam coal per kw-hr. of net output. This also approaches the economic limit for a station of this type.

Fig. 3 shows the economies that can be effected by combining the functions of these two power stations in a single plant, producing the same quantities of steam for process and the same net output of electric energy. Instead of the total heat consumption of 990 million B.t.u. per hour required for the two individual plants, we have a heat consumption in the combined plant of only 800 million B.t.u. per hour, a saving of 19 per cent. Had we assumed a more efficient central station, with higher initial pressure, reheat, and additional stages of feedwater heating, the fuel saving would have been proportionately greater.

The above fuel saving is based on the ideal condition illustrated in Fig. 1, where the demand for exhaust steam is precisely equal to the supply. This never occurs in practice. Either there is an excess of exhaust steam which is blown to the atmosphere or concealed by wasteful application in process, or there is a deficiency of exhaust steam, and live steam is supplied through a reducing valve. In one case there is a total loss of valuable heat, in the other a degradation of energy and loss of potential power. Under any practical conditions, therefore, the saving should be greater than that indicated by these heat-balance diagrams.

Many industrial boiler plants are operated in whole or in part on waste fuel. If 50 per cent of the heat supplied to the boilers in Fig. 1 is waste fuel, and this waste is utilized in the combined plant of Fig. 3, then the saving in purchased fuel is 30 per cent.

Analysis of the more important wastes in these heat balances shows that the fuel saving in the combined plant is effected by a 22 per cent decrease in auxiliary power consumption (largely due to elimination of condenser auxiliaries), 13 per cent reduction in blowdown losses, 28 per cent reduction in other boiler losses, and 100 per cent elimination of condenser losses. Reduction of radiation, leakage, friction, generator losses, etc., not taken into account on these diagrams, should offset, in large part, the transmission losses incident to the combination.

Under favorable circumstances, the cost of a combined steam and power plant of this kind will cost less than two separate plants for the two functions. If the combination involves transmission of power or steam over great distances, or if the loss of steam in process necessitates the installation of expensive evaporators, the case may be reversed, but there are obvious savings in the elimination of condensers and circulating-water supply, and the reduction in the number of boilers and other items of equipment. These savings are not offset by the greater amount of high-pressure piping and equipment. For the particular case under discussion, the capital cost is estimated as follows:

Cost of complete industrial power plant, in accordance with Fig. 1, including land, auxiliary structures, etc...	$2,168,000
Increment cost, for capacity assumed in Fig. 2, for large central station, including land, auxiliary structures, etc.....	2,220,000
Total cost, separate plants.....	$4,388,000
Complete cost, combined plant, in accordance with Fig. 3, including land, auxiliary structures, etc.....	3,928,000
Saving effected by combination.....	$460,000

If the combination plant is so situated that it can supply steam and power to several contiguous industries, then the cost of such a plant might be taken on an increment basis, as well as the central station. The same applies if an existing central station is so situated that it can supply one or more industrial plants without duplicating existing coal-handling facilities and the like. Both of these cases result not only in a greater reduction of initial cost, but also in material savings in operating labor and maintenance.

Now, to discover the potential savings from the cooperative effort of industries and public utilities, and to illustrate these savings by diagrams, is a very simple engineering problem. To put the idea into practice, and to convert these potential savings into cash, is a business problem which is not simple at all. There is a deep-seated and well-founded reluctance on both sides to invade what, up till now, they have each considered the domain of the other. However, our electric-power companies have neglected few opportunities to extend their service or to reduce their costs. Our manufacturers have been equally diligent in reducing production costs. These heat units, and the dollars they represent, will not be allowed to roll into the atmosphere and into the ocean forever.

The first requirement for the success of such a cooperative enterprise is that the advantages must be equitably divided between the two parties. If the plant is owned and operated by the manufacturer, the utility must be assured of the continuity of the power supply. If, on the other hand, the utility owns and operates the plant, the manufacturer must be assured of the reliability of his steam supply. If a plant is built under joint ownership, there must be a just division of costs between the two services. All three of these plans are in use in this country today. All require mutual confidence, good engineering,

and intelligent management, and each party must have a sympathetic understanding of the problems of the other.

Another field in which public utilities and the industries may profitably cooperate is in the better utilization of waste fuel. There are some industrial plants which can supply their entire heat and power requirements with combustible waste from their manufacturing operations. To improve the efficiency of power generation or heat utilization in such plants would only create an unmarketable surplus of waste fuel. If surplus power could be sold at a figure advantageous to both producer and purchaser, then it would pay to use efficient apparatus and prevent unnecessary waste.

The pioneering is now being done by the more enterprising managements, and we may look for a rapid and widespread development of cooperation between public utilities and the large steam-using industries.

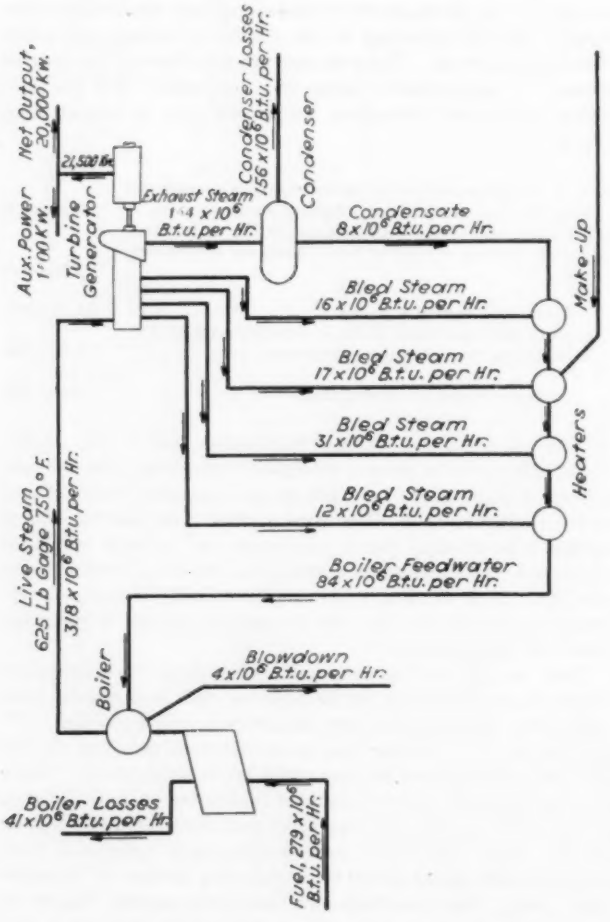


FIG. 2 HEAT BALANCE, CONDENSING CENTRAL STATION

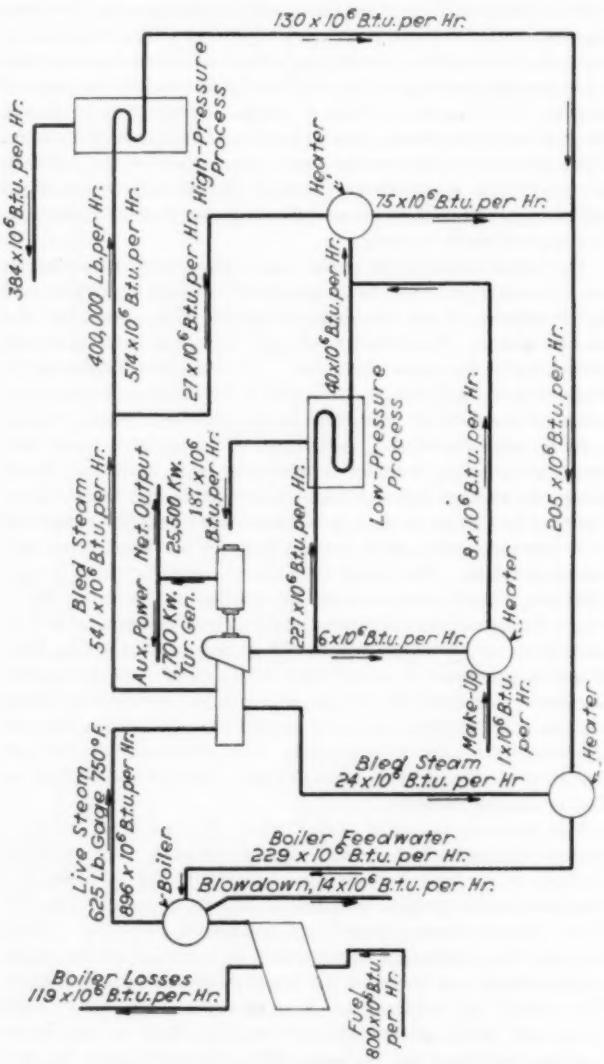


FIG. 3 HEAT BALANCE, COMBINED CENTRAL STATION AND INDUSTRIAL PLANT

INTERNAL-COMBUSTION ENGINE IN THE INDUSTRIAL PLANT

The Diesel engine has been used for many years as a prime mover in industrial plants, where the cost of oil fuel and other conditions have made its use economical. Industries which require process steam have generally found it cheaper to use steam-driven prime movers, but even in this case, oil and gas engines have been effectively used in conjunction with steam boilers. Boiler feedwater has been used as a cooling medium in the cylinder jackets, and in some cases the heat in the exhaust gas has also been utilized for preheating feedwater.

TABLE 1 SUMMARY OF SAVINGS EFFECTED BY COMBINED PLANT, FIG. 3

	Industrial plant, Fig. 1	Central station, Fig. 2	Total separate plants, Figs. 1 and 2	Combined plant, Fig. 3	Saving effected by combined plant, per cent
Gross output, kw	6,700	21,500	28,200	27,200	3
Auxiliary power, kw	*1,200*	*1,500*	*2,700*	*2,100*	22
Net output, kw	5,500	20,000	25,500	25,500	
Fuel input, B.t.u. per hr	711,000,000	279,000,000	990,000,000	800,000,000	19
Blowdown losses, B.t.u. per hr	12,000,000	4,000,000	16,000,000	14,000,000	13
Other boiler losses, B.t.u. per hr	105,000,000	41,000,000	146,000,000	105,000,000	28
Condenser losses, B.t.u. per hr	0	156,000,000	156,000,000	0	100
Capital cost of plant	$ 2,168,000	$ 2,220,000	$ 4,388,000	$ 3,928,000	10
Reduction in capital cost effected by combination					$460,000
Annual saving in fuel cost based on coal at $3.50 per ton of 2000 lb					$210,000

Very recently, a number of Diesel engines have been installed to perfect the heat balance of industrial plants. Attention has been called to the lack of synchronism between steam and power demand in the average industrial plant. In the past it has been common practice to design the electric-power plant to meet average conditions, and to blow exhaust steam to the atmosphere whenever the demand for power was out of proportion to the demand for exhaust steam. Condensing units to take care of this condition are impracticable in plants which lack an abundant supply of cold water. A purchased-power connection is also frequently undesirable, on account of the high cost of furnishing current for intermittent use; also the hazards of interconnection make it difficult to secure an exact balance by this method. In this situation the Diesel-engine generator becomes a very effective method of perfecting the balance.

All of the power that can be generated non-condensing, and without waste of exhaust steam, is obtained from steam prime movers; the deficiency is supplied by the Diesel. The heat saving is obvious; a non-condensing steam engine may reject 20,000 to 40,000 B.t.u. for every kilowatt-hour generated, whereas the heat loss from a Diesel need not exceed 4500 B.t.u. per kw-hr.

Internal-combustion engines are also commonly used for starting up service in isolated plants. Where all auxiliaries are motor driven, as is now common practice, and where electric current is required for fuel supply, which is usually the case in pulverized-coal plants, the boiler cannot be put into operation after a complete shutdown without some auxiliary source of power. A small gasoline-engine-driven generator is usually sufficient for this purpose.

Recovery of Heat From Low-Temperature Waste

As noted in the foregoing pages, much progress has been made toward developing the full possibilities of the high-temperature end of the industrial heat-power cycle. If we continue to progress in proportion to the square of the time, the possibilities in that field will soon be exhausted. Fortunately, there will remain almost unlimited room for improvement in the low-temperature ranges.

There are industrial plants in the United States which use as much as 100,000,000 gallons of cooling water per day. In some cases this water is heated as much as 40 deg. fahr. This represents a potential energy source, at Carnot efficiency, of more than 30,000 kw. continuously. It would be costly, but nevertheless entirely feasible from an engineering point of view, to install apparatus which would extract more than 10,000 kw. continuously from this warm water, without any expenditure for fuel.

Other plants discharge smaller quantities of water at much higher temperatures. Some chemical plants in Europe are actually extracting electric power from hot waste liquors. Vapor flashed from the hot liquor is drawn through a turbine into a condenser. The operating pressure may be as low as 5 lb. absolute.

Except in exceptional cases, the development of power from this source is not yet commercially practicable in the United States, but it would appear to offer a much more economical source of power than exists in the differential temperature between the surface and the depths of the sea. The work of Messrs. Claude and Boucherot on the latter project is of exceptional interest, and the author does not wish to disparage their experiments in any way. The ultimate value of this research, however, may be found in the recovery of energy from the higher heat head of industrial wastes.

For many years the boiler-feedwater economizer was in disfavor among American central-station designers. It was considered to give more trouble than its heat recovery was worth. When we began to force our boilers to higher outputs, and the temperature of gas from the boiler increased accordingly, the economizer came back into fashion, not only as a desirable feature, but almost as a necessity for an efficient boiler plant. We may look for a renaissance of the waste-heat boiler in the industrial plant for much the same reason. We are pushing all production apparatus to higher and higher rates of output. As the waste gases get higher in temperature and greater in quantity, the possibilities of heat recovery become correspondingly more attractive. Waste-heat boilers have been developed for high steam pressures, several have been installed for pressures above 350-lb. gage, and this further increases the recoverable energy.

Hot gases are also being used directly, or through the medium of an air heater, for drying and for other manufacturing requirements. At the Iron Mountain plant of the Ford Motor Company stack gas from the power plant is used for drying wood. This plant, which utilizes not only the latent heat of the exhaust steam but also the sensible heat of the stack gases, eliminates the two major losses in the central station. This is the goal for which we should strive in low-temperature ranges. If operated at the highest economical steam pressure, distributing cheap surplus power to less fortunately situated neighbors, such a plant would represent the ideal industrial heat-power cycle.

Forecast

Not many years ago the industrial-power engineer spoke contemptuously of his central-station brother as a "B.t.u. chaser." The implication was that the central-station designer pursued ideals of thermal efficiency without regard for sound economics. In spite of this alleged extravagance, the public utilities have waxed great and prospered. Industry has seen the light. It is chasing a few heat units on its own account, and finding that dollars may be caught in the same net.

The charge was also made that public utilities could be extravagant with impunity, because monopoly assured them of an adequate return on any investment they chose to make. This, of course, was nonsense. In the industrial field the central station is in competition with private generation of power. In the domestic field it is in very keen competition for the consumer's dollar. Electric light may now be regarded as a necessity in the American home, but one does not have to have four base plugs in every room, an electric washing machine, an electric ironer, an electric radio, or an electric refrigerator. Still less does any one need an electric range. Yet all of these luxuries are in quantity production solely because the electric-light industry is successfully meeting competition. Here again industrial management has caught the idea; it has learned to distinguish between extravagance and financial courage. Industrial executives are now willing to make sound investments in power equipment, and they are finding them gilt-edged investments.

Therefore we may look for an undiminished acceleration of progress in the industrial-power field. More high-pressure plants and higher pressure plants will be built. Public utilities and industries will cooperate to obtain the maximum power yield from process steam; the day may come when we shall reverse our present nomenclature, and call the single-purpose central station an "isolated plant." Industrial-power engineers will attack their present low-temperature wastes with more success; they will contribute their share to retard the ever-increasing entropy of the universe. It is difficult to believe that the first differential of the progress curve can always be positive, but Henry Adams assures us that it can, and that it will.

Discussion

EDGAR J. KATES.[4] Mr. Ryan alludes to the fact that very recently a number of Diesel engines have been installed to perfect the heat balance of industrial plants; the principle is that all of the power that can be generated non-condensing and without waste of exhaust steam be obtained from steam prime movers and that the deficiency be supplied by the Diesel.

Why not carry the idea further, and rather than merely add a Diesel engine as an auxiliary to a complete full-size steam plant, design instead a combination power plant with steam and Diesel capacity carefully proportioned so that the load will be shared according to an analysis of capital investment as well of operating cost? In other words, do not install a Diesel engine just for the purpose of occasionally improving plant efficiency by preventing the waste of exhaust steam. Greater savings can be effected by planning a well-balanced combined plant in which both types of prime movers will be used more hours per year than a thermal analysis alone would call for. It is evident that that part of the steam power plant whose exhaust can be utilized only a short time per year will not justify its capital investment if Diesel capacity is available to carry that part of the load. The principles involved in the joint use of Diesel and steam engines are given in a recent paper presented to the Oil and Gas Power Division, A.S.M.E.[5]

Complete analyses of actual installations frequently show that the most profitable design of plant is a combination of steam and Diesel power units in which neither the steam nor the Diesel capacity by itself is sufficient to carry the full load. Such combinations will result in a considerably lower capital expense than if Diesel power is added to a full-size steam power plant, and despite some loss in thermal economy the decreased investment will result in lower overall costs.

A further saving can sometimes be made in such plants through the use of steam power equipment of lower cost and somewhat lower efficiency than would be desirable if the plant were all-steam. Inasmuch as the power required beyond that obtainable from the non-condensing steam equipment can be generated at slightly additional cost by Diesel engines, it does not pay to increase the capital investment in the steam plant too much in order to obtain the utmost thermal economy from the steam equipment. For instance, in a combination plant a four-valve steam engine may be better than a uniflow engine because its lower initial cost may offset the slightly increased fuel consumption under conditions of moderate steam pressure and atmospheric exhaust.

There seems to be an error in the author's statement that "the heat loss from a Diesel need not exceed 4500 B.t.u. per kw-hr. even if no attempt is made to recover heat from jacket water or exhaust." The heat loss of a modern Diesel engine is about 7500 B.t.u. per kw-hr.; of this loss about 4500 B.t.u. per kw-hr. is recoverable from the jacket water and exhaust.

Mr. Wood refers mainly to large industrial power plants that are able at times to deliver considerable amounts of power to the public-utility lines to which they are connected. The protection of service to the industrial plant is cited by the author as one of the advantages of the interconnection.

There are hundreds of small industrial plants whose capacity is not enough to justify the complication and expense of delivering surplus power to the public utility, but which, like the larger plants, would benefit greatly by a connection with the public utility that would protect their power service.

However, judging by the almost prohibitive rates which many public utilities charge for such a connection, which they generally designate as "breakdown service," the utilities seem to oppose rather than to welcome this type of service. For instance, a utility in a large Eastern city makes a service charge of $24 per year for each kilowatt of maximum demand contracted for under the breakdown agreement, even if no power is actually supplied. At the usual rate of return of 8 per cent on the investment, this service charge pays, for each kilowatt of maximum demand contracted for, the full rate of return on an investment of $300. Since this is approximately equal to the utility's whole investment per kilowatt installed, the implication is that the utility is setting aside solely for the use of each breakdown service customer the full amount of capacity called for by that customer's *possible* demand. That, of course, is not the case, as the diversity factor of the breakdown service supplied to a group of customers is quite large. In fact, if the business were cultivated, it is likely that its diversity factor would be much larger than that of any other class of service. Consequently only a small fraction of the maximum breakdown demand of any one customer needs to be actually reserved for his use, and this justifies a large reduction in the service charge.

If breakdown service were less costly, the owner of a private power plant would be justified in depending entirely upon the utility for reserve power and in reducing his own plant capacity and investment. The gain would be mutual, as not only would the plant owner save capital investment and space, but the utility would earn a reasonable return on its own additional investment. On the other hand, under existing conditions, the plant owner is put to an unnecessary capital expense because he cannot afford to pay the present utility rates for breakdown service, and the utility is deprived of additional profitable business.

JOHN H. LAWRENCE.[6] The subject of power and heat supply in industrial plants is one which was long neglected by industrial executives. Although the industrial power plant was the first power plant, its existence was practically ignored, and it was almost left to shift for itself.

The advent of the central station brought new ideas into the power field, and, as the author says, "the public utilities have waxed great and prospered" because they have studied their problems.

Granted that power is not the salable product of an industrial company and is only one step in the manufacture, still it is regrettable that executives have not had their organizations keep pace with developments in a subject which is absolutely essential to their companies. Power is essential, and if this is so, why not procure it at the lowest cost consistent with good business judgment? The writer adds the last phrase because sometimes a company cannot afford to tie up too much of its capital in plant other than what is necessary to produce its product.

However, we can find today plants making an investment to cut down manufacturing cost of products, where the same investment made in the power plant could net a larger return. This is done because the executive has ignored the power plant and has been satisfied as long as steam and power have been available. In the meantime, the heat, for which he has paid good money to the coal man, is going up the stack or into the condensing water. Heat units are elusive things, and if they get an opportunity to escape, they do not wait to ask permission.

Fortunately, industry has seen the light, and there are many excellent examples of efficient power plants in industry today. However, the writer is not so optimistic concerning the extent of improvement in industrial power plants. The average plant today looks as if it were handed down from the Ark.

[4] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

[5] "Joint Use of Diesel and Steam Engines to Balance Heat and Power," Edgar J. Kates, Trans. A.S.M.E. (1930), paper OGP-52-4.

[6] President, Thomas E. Murray, Inc., New York, N. Y. Mem. A.S.M.E.

There is an enormous field for improvement in industrial power. The surface has hardly been scratched. It is up to the engineering profession to educate the industrial executive in the economics of steam and power development.

The central station has been developed to such a point that very little improvement can be made unless some new cycle or process is developed, and the only hope at the present time for materially reducing the cost of power in central stations is to reduce fixed charges.

Many industrial plants are in an enviable position. Due to their processes, they are in a position to manufacture power at a rate lower than it can be purchased from any utility company, for the investment cost in excess of that required for process steam and the heat required to develop the power are less than in a central station.

On the other hand, one may find industries making their own power when they should be purchasing from the central stations.

Economics should govern, but unfortunately some people are not as familiar with that term as they are with slices or hooks on the golf course. We should endeavor to get the plant owner to look at his power plant from the standpoint of economical production of power and not of the necessary expense of providing something with which to drive machinery.

The author and all of the others who write papers on this subject are spreading the cause of good engineering and also of conservation of the fuel resources. As a nation, we are wasteful, and this is particularly true of the manner in which we have been using fuel in our industrial plants.

In tackling the problem of improving the industrial power plant, we should not blindly follow central-station design. What may be good for the central station may be very costly for the industrial plant. Unfortunately in the past designers patterned too closely after the central stations, and as a result progress in industrial power-plant design was held back. Today, the problem is attacked from the standpoint of building a plant which will be coordinated with manufacturing processes, so far as use of heat is concerned, instead of giving out power and steam promiscuously in a haphazard fashion.

The author's suggestion of interconnection of public utilities and industries is well worthy of serious thought, as under certain conditions large savings can be made in this way. Until recently public utilities were loath to interconnect with industrial plants owing to possible disturbances on the systems due to plants over which they had no control. The central station had to meet this problem, and today there are many such interconnections. However, the industrial operator should not be content merely because he has an interconnection; he should know that he is using heat in the most economical manner, and this is true whether or not he has an interconnection.

E. D. Dickinson.[7] For several years the writer has given this subject much study, and it has been interesting to note that this question of the desirability of generating by-product power is being answered as all engineering questions are ultimately answered, on its economic merits. In other words, in any instance where it can be shown that it is a sound economic policy to generate electric power from process steam, it is then the problem of the engineer to see to it that the installation be entirely reliable and proportioned so as to satisfy all the requirements on which the decision has been based.

The author has very clearly illustrated the savings to be made by cooperation of public utilities with manufacturing plants to develop by-product power to its economical limit.

At a meeting of the Affiliated Technical Societies of Boston in December, 1925, the writer predicted that there would be many cases in the future where there would be cooperation between the industries and the public utilities to their mutual advantage.

It is gratifying to observe that each year there is increasing interest shown in this phase of heat and power supply. This statement regarding the present trend is based on a continuous survey of all requests for quotations on steam turbines.

That the manufacturers have anticipated the use of by-product power to its economical limit is manifest by the many recent installations of highly efficient back-pressure and extraction turbines for by-product power application. These by-product power turbines are essentially special. It is not unusual to find such a turbine of 5000 kw. rating taking as much steam as a 20,000-kw. condensing turbine and exhausting no more than a 1000-kw. non-condensing turbine, the balance of steam being extracted at one or more openings.

The question of reliability is of paramount importance. Manufacturers of steam turbines recognize this, with the result that they are prepared to furnish so-called industrial turbines; that is, turbines arranged for extraction and back pressure that are as conservatively designed and in every way as reliable as the large turbines operated by the public utilities.

The writer would stress the necessity for most careful study of all operating conditions before drawing specifications for the turbine. The greatest gain can be secured only when the turbine is properly proportioned to meet the operating conditions.

The many problems of governing and control are not generally appreciated by many, but are well understood by those who have given the subject much study. It will be appreciated that when there are several variables and that when fine control in pressure is to be maintained at several extraction points, in addition to close speed regulation, the governing mechanism must be powerful, strongly built, and at the same time sensitive. Adequate provision must be made to properly safeguard the turbine at all times.

The matter of bookkeeping that the author mentions is no doubt one of the most important and yet most unreasonable elements tending to retard the by-product power situation. Figures always have been and always will be convincing, but if they are based on unsound presumptions they will lead to false conclusions. Without doubt there will be found many desirable applications for interchanging power between the public utilities and the industrials.

Charles W. Conrad.[8] The excellent papers of Messrs. Ryan and Wood appeal to all utility and industrial power engineers because we are interested in the most economical utilization of heat. It is hoped that these papers will be one more step in the education of managements of both the utility and the industrial plant.

In a paper presented by the writer at the October 26, 1927, meeting of the Boston Section of A.S.M.E., it was stated, "The future will find our large utilities deeply interested in the development of a number of medium-size combination power and steam supply plants suitably located to furnish manufacturers, or in establishing manufacturing districts around their base plants to obtain a market."

Therefore the writer would like to supplement Mr. Wood's means of securing advantages of massed resources of power production with the proposal that the utility should plan for a manufacturing district around its large generating station. Land or buildings could be leased or sold to manufacturers planning new factories or branch factories. This opens a field for the future

[7] Designing Engineer, Turbine Engineering Department, River Works, General Electric Company, Lynn, Mass. Mem. A.S.M.E.

[8] Plant Engineer, Bird & Son, Inc., East Walpole, Mass. Mem. A.S.M.E.

for which the medium-size as well as the large utility can plan.

JAMES M. TAGGART.[9] Power is becoming a fluid commodity, due to interconnection of generating systems and facilities for power storage inherent in a combined steam and hydraulic system. Improvements in design of generating plants and increases in size have resulted in a production cost largely dependent on fuel burned and fixed charges for steam plants and fixed charges alone for hydraulic plants. On this account cooperation between industrial companies, having both steam and power requirements or waste heat, and power supply concerns has become of extended importance economically. In the past there have been many cases where the conditions were such that cooperation would have been just as beneficial, but it has been extremely difficult for the interested parties to comprehend the facts. Papers like those of Mr. Wood and Mr. Ryan should help clear up the situation.

On account of the many varying factors each case involves special engineering and economic features that require special study to solve effectively. The success of such special installations depends inclusively on suitable arrangements, correctly selected equipment, and designing of the essential details to fit the conditions.

While in some cases, as suggested by Mr. Wood, the power supply company will be best fitted to build and operate the combination plant, on the other hand often it will work out better for the industrial company to provide the plant. The latter would be especially true where the plant is relatively small, where its location is isolated, and where its arrangement and operation naturally tie up with the industrial plant.

Just as the arrangement of each cooperative plant is special, so also is the value of the heat or power interchanged. As a rule, in addition to the reduction in total B.t.u. required, there should be a reduction in total investment required and an increase in reliability of operation. Mr. Wood mentions 5000 and 12,000 B.t.u. as a proper ratio of the heat per kilowatt for power produced with exhaust utilization compared to power produced in a high-pressure central station. While this is comparatively correct for power at the generator outlets, it is normally different for the usual conditions. Thus for power delivered to an industrial plant there will be transmission and transformation losses often over 20 per cent in amount. When power is to be delivered to the power supply company, the points of supply and the location of the load need to be considered.

Mr. Ryan speaks of a cost of 1.5 mills per kw-hr. as probably in error for plants utilizing their exhaust steam. In such plants there is always a question as to just what should be charged for power. Where the load factor is high, as is true for certain types of plants, and the coal cost is low, with fixed charges at 15 per cent, 1.5 mills per kw-hr. is possible. With direct steam drive, where the conditions are favorable, an equivalent power effect can be obtained at less than 1.5 mills. It is true, however, that in most industrial plants careless design, skinning down of investment, and unintelligent operation produce an actual power cost exceeding 5 mills per kw. even where all the exhaust is utilized.

Power-supply companies tend to underestimate the value of a possible power supply from industrial works and to totally disregard the value of an increase in capacity. Industrial concerns producing their own power on the other hand normally overestimate their loads. Unprejudiced and accurate engineering is required to achieve a mutally profitable and efficient cooperation.

GUY B. RANDALL.[10] Mutuality of interest in more and cheaper electric power by the public utilities on the one hand and all the other types of industries, as well as the public at large, on the other is readily apparent, at least in the abstract. The papers of Mr. Wood and Mr. Ryan are successful attempts to give this mutual interest a practical and concrete form through the medium of high steam pressures. The great possibilities appear to lie in:

1 Joint ownership as quite clearly described in the two papers
2 Simple interconnection of electric circuits.

Joint ownership may develop a fair field of action in some of the larger projects. A start has been made. But there are serious objections to it in the case of the majority of industrial companies unless the utilities use the new medium to insure the equivalent of far more favorable rates relatively than their present rate structures provide. The writer's observation indicates they may be disposed to do this to some extent, but the savings to be derived by the industrial plant are heavily reduced by the higher cost of public utility construction and the profit the utility must earn. In both papers construction costs are given which for this discussion represent large power developments and the cost ranges from $100 to $114 per kw. In the writer's recent experience a first-class 650-lb. industrial plant station ranging in capacity from 15,000 to 20,000 kw. can be built completely new for $70 to $85 per kw. including all legitimate charges and profits. (A general description of the 650-lb. plant of the Champion Coated Paper Company, on which this statement is based, is given in the December, 1929, issue of *Power Plant Engineering*.) This price does not include the effect of substantial price reductions in the last six months. Nor is the average industrial plant likely to look with favor on the contractual relations required for joint ownership which, as Mr. Wood points out, may be complex. It would appear that joint ownership and so-called mass production have yet to demonstrate that they may offer the public utility a suitable profit without serious infringement on the industrial plant potential profit.

Industrial plants do not suffer, in general, from lack of funds. In the writer's experience with several companies he has yet to find one which could not obtain money for a development program provided it was worked out in clear and concise fashion and where the profits therefrom met the required minimum. In competitive industries profits cannot be standardized. But the high thermal efficiencies obtainable today in up-to-date plants of moderate cost open the door to attractive savings. Almost any plant is a good prospect. If it is a large buyer of power, it may prove to be a better prospect. While many industries vaguely feel the urge to action, they also know from previous painful experience that it is quite possible to get a brand new, up-to-date set of equipment or its equivalent in the form of a contract while others absorb the lion's share of the profit to be derived therefrom. Even with highly detailed specifications in connection with construction work, it may be found that one bid for high-pressure equipment is double another. This simply means that both engineering and purchasing must be done with circumspection and with the widest possible canvass of the market until practice and prices get on a more uniform basis.

Simple interconnection of electric circuits would seem to offer the really great and essential opportunity for development. Under this scheme, if the isolated plant has surplus power, it dumps it into the general pool of power. If it needs some, it draws from the pool. There are no financial complications. But it must get paid reasonably for what it delivers, and it must pay reasonably for what it gets. This relationship is simple and direct, with points of resemblance to both modern

[9] Consulting Engineer, New York, N. Y. Assoc-Mem. A.S.M.E.
[10] Superintendent of Power, Champion Coated Paper Company, Hamilton, Ohio. Mem. A.S.M.E.

railroading and modern banking. A power pool would take power on consignment somewhat like a railroad takes freight. For a moderate tariff it would take it at one point and deliver it at another. It would be able to consolidate these varying and variable deliveries into such a large fund of power that the greatly reduced discrepancy between supply and demand could be smoothed out by its own firm power resources which, like a bank's own financial resources, would be small in size compared to the total deposits. You will notice that the emphasis is placed on the electrical transmission and distribution network (which today probably compares as to reliability with railroad lines of thirty or forty years ago), while the utility power station shrinks in relative size and becomes a balance wheel or equalizer to take care of unusual conditions. If every plant which has use for process steam or process heat were working on 1400-lb. pressure and the best heat cycles of today, the utility steam stations would probably be nearer their no-load point than their normal load point so far as a national pool is concerned. Almost regardless of where you live, just check up on this carefully with a radius of twenty-five or fifty miles and you will appreciate the full force of this statement. This does not mean that properly managed public-utility stations are going to permanently lose load. It is, rather, a striking way of indicating where it would seem the future additions to capacity should come from.

As matters now stand the public utility does not in general seem disposed to buy firm power from an industrial plant, and it is "willing" to purchase dump power only at a price close to the utility's fuel cost of a kilowatt-hour. It is easy to demonstrate that at this price there is little or no incentive to design to get the maximum of this, the world's cheapest power. It is also easy to demonstrate that by the standard of present utility rates dump power is worth far more than this. But an attractive price for dump power gives the utility a double handicap. It increases the incentive to build modern industrial power plants, which in turn means loss of more revenue-bearing load for the utility—a vicious circle. No utility company can wisely ignore the plain implication of this trend. But looking at the problem from the standpoint of national economics and national welfare, it would appear that the sooner we develop a large power network with great freedom of pool action—that is, the sooner the public utility regards itself as something of a common carrier rather than an exclusive manufacturer—the sooner will the country enjoy an abundance of really cheap power. Our national power problem is not solved until the cheapest power has a reasonably good road to market. Meanwhile, it would appear advisable for the industrial plants not to make long-time dump power commitments, for the market may improve considerably within the next ten years.

Both Mr. Wood and Mr. Ryan regard evaporators as necessary for the high-pressure plant. For the average industrial plant, the writer's studies and experience indicate that evaporators offer little benefit while they add to the cost and complication and greatly lower the thermodynamic efficiency where the make-up is large. Let us not forget that the following pressure changes are thermodynamically of approximately equal order: a decrease of $^1/_2$ lb. at 1 lb. absolute, 12 lb. at 30 lb. absolute, 55 lb. at 190 lb. absolute, 110 lb. at 400 lb. absolute, 150 lb. at 600 lb. absolute, 325 lb. at 1200 lb. absolute. Regardless of whether evaporators are used, final reliance for trouble-free operation at the present time must be placed on boiler-water chemical control with secondary phosphate conditioning. From recent experience, for example, at 650 lb. a large boiler operating on 30 to 40 per cent make-up of bad water is as clean after six months of operation as it was after six days. As nearly as one can determine, the boiler can be kept clean forever with primary hot treatment of lime and soda ash and secondary phosphate conditioning. In this particular case, 650 lb. pressure without evaporators was thermodynamically equal to something like 800 lb. with evaporators. The increase in expense and complication caused by evaporators with large make-up is too great to ignore. This does not mean that there are not many legitimate evaporator applications. In dealing with sea-water impurities, for just one example, they are probably imperative. But the work of Dr. Hall and his associates in boiler-water conditioning has opened up a field which is not generally appreciated. People just do not believe it when one talks about running a boiler forever without the use of a tube cleaner.

Mr. Ryan also directs attention to the superior engine efficiency of the reciprocating unit up to 5000 kw. The writer would in general prefer to have a small high-pressure plant with turbines (in place of engines and oily condensate filters) and without evaporators. The cost is much less and the elimination of the evaporators offsets the higher engine efficiency of the reciprocating unit. It is believed that this condition will hold down to units as small as 1000 kw. Further, there is no serious objection to a small condenser with the turbines. In general, it is cheaper to have a little condensing capacity to care for fluctuations than to boost the steam pressure so high that a bypass valve is the only medium necessary to obtain balance. This particularly applies to plants which cannot get a reasonably profitable outlet for dump power through the public utilities or through local groups of industrial plants.

Mr. Ryan questions industrial power costs of 1.5 mills per kw-hr. including fixed charges. In Ohio that is possible if no reserve equipment is carried. With reserve equipment, a total cost of 2.0 to 3.0 mills including fixed charges is readily achieved provided only the load factor is good. It is interesting to note that at the beginning of this century a kilowatt-hour was priced in dimes, fifteen years ago it was priced in cents, while today it is priced in mills. In fact, we are now very much interested in small fractions of a mill. We are approaching the price range of one of Fred Low's old editorial dreams of "power cheap as tap water at the kitchen sink."

R. J. S. Pigott.[11] Mr. Ryan's excellent paper brings out again the fact that the power and heat supply for any industrial plant requires an individual solution. There is no such thing as a general solution. A further complication of the situation is that in many cases it is very difficult to determine the precise heat and power distribution in any industrial. Furthermore, any solution for a particular plant must be made flexible enough so that considerable changes in processes which materially alter the balance between heat and power might be made without destroying the ability of the system adopted to handle the change of balance to advantage. Without doubt the most useful information one could obtain would be a write-up of every individual solution that actually has been developed and in particular the operating results where these figures are available. These, as a general rule, are rather hard to get because the industrial is usually more interested in running the new plant after it has been obtained than in checking up how well it justifies the original computations.

Another factor which has recently reentered the field is the Diesel engine. When internal-combustion engines (and in particular the Diesel) were first commercially pushed, they were much too enthusiastically sold on the basis that they required little skill to run them. This was certainly not true, and is not true today. The result was that the internal-combustion engines and in particular the Diesel engine got a black eye which has taken 15 or 20 years to cure, but we have now available in-

[11] Chief Engineer, Research Department, Gulf Production and Pipe Line Company, Pittsburgh, Pa. Mem. A.S.M.E.

ternal-combustion engines whose reliability is certainly on a level basis with other forms of prime mover. In marine service and in the oil industry this condition has been obtained to a degree quite surprising to those interested in the electric-power industry. The Diesel engine very often offers a solution for industrial heat-balance situations that can be reached no other way.

With regard to low-head heat recovery as another avenue of improvement in thermal economy, it does seem as if something useful might be done in this direction. At Youngstown, for example, the whole of the Mahoning River is raised to a temperature of 120 deg. in July and August near some of the steel mills, and a few cases have been reported where the temperature rose to 136 deg. Of course much of this heat actually need not be wasted; the writer's experience indicating that in general the steel mills are extraordinarily lavish in their use of cooling water, and the process as a whole is relatively uneconomical in the use of heat. The main obstacle in the way of providing low-head heat recovery equipment will be the size and cost of apparatus developed.

Many of the industrial plants must lose much of the condensate from process steam and are therefore under the necessity of providing a large percentage of boiler feed as make-up. The solution generally adopted so far for high-pressure plants is to use high-pressure evaporators to deliver steam for industrial service. However, it is very probable that we shall be able to supply boilers operating at quite high pressures with treated make-up and have no more trouble than is now experienced with evaporated make-up and condensate.

Another discussor has mentioned the use of hot-process lime-soda treatment followed by phosphate secondary treatment in a 600-lb. plant. The writer can supplement this experience with another case where the plant operating at 450-lb., 70 per cent make-up is operated with the same treatment and producing no scale in the boilers that will not wash out. In this case, however, a very considerable deposit of scale is thrown down in the high-pressure heaters (140 lb., 360 deg.). This plant is operated with a continuous blowdown system to keep the boiler-drum concentration down to about 200 grains per gallon total solids.

Mr. Wood's interesting paper brings forward again, and forcibly, the necessity for serious consideration of more widespread combination of power and heat facilities between public utilities and industrials. The battle between the industrial plant and the public utility on the question of which is better equipped to furnish the cheapest power has in the past 15 years become more clearly defined as an economic question of how to furnish heat and power. It has been fairly obvious that where by-product heat at a lower heat level than the initial steam conditions is required by an industrial plant, two to three times the heat may be usefully obtained from fuel by means of a combination plant over that which a straight condensing plant can deliver. There have been a great many industrial plants built in which full advantage of this situation has been taken, but it is relatively recently that serious consideration has been given to the question of how much more can be saved by combining the facilities of a public utility with those of an industrial. Those cited by Mr. Wood are perfectly clear and determinable in each case, and it may be pointed out again that there is no general solution for the problem, but each industrial must be calculated individually, since the accident of location and the synchronization or lack of synchronization of heat and power loads have a most profound effect upon the solution.

The major obstacle to the more rapid development of interconnections by way of combination plants between public utilities and industrials is mainly lack of confidence of the two groups in each other. For example, as a general rule the public utility engineers look upon the power produced by an industrial plant as being less reliable than that which they produce themselves. On the other hand, the industrials, and especially the larger ones, feel that their own power service is more reliable than that of the public utility. Such a situation is somewhat absurd. It may be fairly said at this time that the production of power by a public utility is actually more reliable than that produced by industrials. The reasons are perfectly obvious. The business of the public utility is to make power, and almost any one will admit that a specialist is best at his own game. The making of power in an industrial establishment is certainly not the main issue, and neither receives the careful attention that the product of the industrial receives, nor is it usually of the size which justifies the employment of high-grade specialists. Nobody will deny that the large public utilities employ more highly trained technical men that any other single group. One can find case after case were an industrial management is afraid that the public utility cannot serve them with power as continuously as they themselves when the industrial boiler plants are showing efficiencies as low as 45 per cent and generating equipment is being operated without any spare capacity whatever.

Of course it is true that some of the very large industrials, such as the steel mills, have plants of their own of sufficient size to justify the services of power specialists and have reliability and operating facilities on a par with those of public utilities. In this situation public-utility service is slightly less reliable than that of the industrial plant, as the outage of the transmission line service is superposed on that of the public-utility station. However, these examples are few and far between, and so far as the writer's experience goes, the plants and operation in the steel mills and other large industrials are definitely inferior of those of public-utility operation.

Another obstacle arises which is equally serious. In many cases the power available from an industrial plant as a by-product for delivery to utility lines is of the nature of off-peak or dump power, and it is more or less the custom of public utilities to rate this power as only valuable to the extent of fuel increment cost. Therefore they are only willing to offer figures of 1 to $1^1/_2$ mills per kw-hr. delivered to their lines, whereas the current they furnish may be charged at 7 mills to 9 mills. It is true that if the power is purely off-peak it is really worth only this much to the public utility, but a great many of the plants which might be considered for interconnection or combination service will deliver large quantities of power which is not dump or off-peak power, and in this case is worth just as much to the public utility, provided it is reliable, as if they made it themselves. This is likely to be the case for certain industrials such as steel mills, paper, rubber, and chemical plants which operate largely on a basis of a 24-hour day, seven days a week. The writer feels that, in the interests of getting a safer consideration of the value of such power, the method of figuring comparisons of cost on the basis of incremental statistics is decidedly dangerous. It is felt it is much safer and clearer to figure complete costs of power for each set-up using graphic methods covering the whole range of loads. The writer has seen figures prepared by two different parties on the same situation which showed a complete reversal due to a slight variation in incremental cost. If total costs had been employed, there could not possibly be any difference between the two. When working with incremental statistics, it is only too easy to lose sight of the effect upon total costs when the variation is carried any distance from a single load point.

On the other hand, the industrial management usually recognizes only operating and maintenance cost as representing the cost of power and lumps its power-plant investment costs in with other plant investment. This results in making comparisons

which are totally useless. No comparison of cost between public-utility and industrial-power service can be worth much unless made on the basis of both fixed and operating charges fairly determined. This second obstacle, therefore, sums up as a failure of both parties to determine prices for incoming and outgoing power that divide the profits reasonably between the public utility and the industrial. There are a great many industrials in which a combination plant owned and operated in some joint manner between the public utility and the industrial would pay the best returns on the investment. There are likewise a great many plants in which no such solution will be profitable, but no real growth can be expected in combination service by industrial and public-utility joint operation until real costs are fully recognized and there is full willingness on both sides to divide the profits fairly. To this end we need standard methods of accounting for costs which will be accepted by both groups and some method of evaluating reliability which is based on other things than opinions.

JULIUS G. BERGER.[12] This important subject of combined heat and power supply in industrial plants is one to which the writer has devoted many years, and he would cite a typical case where a manufacturing plant was about to increase its size considerably owing to another interest coming in with more capital. One of the questions to be decided was as to how steam and electricity needed to be supplied under the new conditions.

Two arguments were set forth by parties having something to sell—one recommended entire purchase of current, the other advised adding several more engines like those installed and making all electric power on the premises.

After a test of the existing power plant and a set-up of the steam and the electric current required in the future, a careful analysis was made of the problem, which was about as follows:

1 Make all possible power in the old plant; buy the requirements beyond that
2 Add a turbo-alternator to the old plant for heat-balance purposes; buy the requirements beyond that
3 Buy all the power, discard the old generating equipment, and motorize some steam-driven compressors with synchronous motors
4 Make all power in an extraction turbo-alternator plant
5 Make all power in a uniflow-engine generator plant.

The comparative figures then became about as follows:

Method	1	2	3	4	5
Comparative investments	$78,000	$111,000	$71,000	$449,000	$497,000
Comparative operating costs	170,000	161,000	171,000	177,000	183,000
Lowest investment, zero; others are relative increase	7,000	40,000	0	378,000	426,000
Lowest operating cost, zero; others are relative increase	9,000	0	10,000	16,000	22,000

This plainly indicated that a heat balance was most desirable, but still to make use of the old equipment and to keep the investment down, a method between 1 and 2 was adopted with very satisfactory results.

The writer has touched on only one phase of the problem that Mr. Ryan has so ably presented, to indicate that such studies lead to the obtaining of economical operating conditions at the lowest investment consistent with good practice.

It is hoped that the Society will continue to keep this subject before its meetings as an aid to awaken industry to the fact that it can not only often reduce its production cost in the power plant, but help to conserve our natural resources.

[12] Consulting Mechanical and Electrical Engineer, Newark, N. J. Mem. A.S.M.E.

C. F. HIRSHFELD.[13] The writer is heartily in accord with the views presented by Mr. Wood and with the arguments which he uses to support those views. In fact, the writer has been working and studying along such lines himself for several years and has reached similar conclusions independently.

There is no question that industrial competition is becoming rapidly more keen, both nationally and internationally. Such competition necessitates the study of everything that can possibly reduce production costs, even though the possible reduction per unit of output be exceedingly small. This is the driving force which activates those engaged in productive industry. A very similar motive, quite differently expressed, activates those engaged in the public-utility business. Here there is the continuing desire to be able to reduce the price charged for services. This reduction can occur only as the cost of producing units of such services can be reduced.

These motives are driving toward a common solution—a solution along such lines as indicated by Mr. Wood. But it must not be thought that this solution is completely or basically new. It rests, in the end, on that old friend, diversity. The public utilities have utilized diversities of various sorts, first in restricted areas and later over large areas, until now we have networks interconnecting such vast areas as are indicated by the author in his paper. We have not yet utilized all the previously recognized diversities along such lines, but we are daily coming closer to the time when we shall have done so; even now the pickings are becoming very lean. Interconnection with industrials, as described by the author, brings in a new set of diversities which can be used to the advantage of both the industrials and the customers of the utilities. It appears, therefore, as the next logical step in the steady march of ever-decreasing cost based upon the utilization of diversity.

Stripped of all complicating considerations, it is this idea that leads the writer to believe in the inevitable progress of the art in the direction indicated by Mr. Wood. But this progress will not be made without great difficulty. There are many complicating factors, psychological, physical, and economic, and it must not be concluded that, now that we see the light, we can all effect such interconnections overnight. Much study and much education are required before much progress can be made.

The time allotted for individual discussion does not permit even a cursory enumeration of the more important difficulties. This is regrettable because a comprehensive statement of them is necessarily the first step toward their solutions. However, one very important one may be mentioned profitably at this time.

A public utility is, among other things, a sort of trustee for the public that it serves. That public profits directly as the utility handles its business wisely and properly. Now consider a situation in which it can be shown that the utility might construct a plant on or near the property of an industrial primarily for the service of that industrial, say with electric power, process steam, and hot water. Assume that it can be shown that such action would result in cheaper service to the industrial than it could itself supply and also in cheaper service to the public through the utilization of otherwise unusable diversities.

Obviously all such results rest upon the assumption that the industrial is going to remain in business long enough to retire the investment made specifically for its service by the utility. Probably they rest on the further assumption that the industrial will maintain at least a certain average percentage of productive activity during that period. It is obvious that the utility when contemplating such a course is faced with the question of how far it is justified in gambling the necessary investment on the assumed

[13] Chief of Research Department, Detroit Edison Company, Detroit, Mich. Mem. A.S.M.E.

continuing solvency and business activity of a single large industrial.

In these days of rapidly changing industrial conditions and industrial products this is no insignificant consideration. The writer does not mean that it is to be regarded necessarily as an argument against interconnection with industrials. He does insist that it is an item which must come up for very serious consideration in any case in which such interconnection involves a large and specialized investment on the part of the utility. And there are many other equally difficult problems involved in this movement.

Z. G. Deutsch.[14] The papers by Mr. Wood and Mr. Ryan have dealt with apparently the same problem. The authors of both papers have clearly and forcibly demonstrated that, if we have sand in one hand and a rathole in the other, our procedure becomes obvious. They have further indicated that in the past this procedure has not been so obvious because, if one may be permitted to continue with the metaphor, the sand has been in the possession of one group, whereas the rathole has been held by another.

It is gratifying to hear that in a few isolated instances these two groups have been able to get together satisfactorily. As pointed out, there is still an enormous field for further work of this nature. It is felt, however, that extension of this work does not alone wait for a financial understanding together with an equitable means of bookkeeping, but that more often the difficulty is the amortization of present invested capital in power-producing equipment, which even though obsolete from the standpoint of utmost thermal efficiency, is still in good physical condition.

As an example of this, the writer knows of one mill which burns approximately 50,000 tons of coal per month and is capable of producing on the order of 40,000 firm kw. as by-product power. In this particular case the local public utility is unable to, or at least is prohibited from arranging for the purchase of this power due to certain contractual relationships; and furthermore the mill is not particularly interested in investing an enormous sum to replace a boiler plant which was a model of efficiency and size about twenty years ago and which has, of necessity, been kept in reasonably good physical condition. The administrators of this mill, and one has heard this as a common expression of other mill administrators, merely say, "But we are not in the power business."

Mr. Ryan's paper refers frequently to 100 per cent elimination of condenser losses and to 100 per cent thermal efficiency as long as the "exhaust steam" can be utilized by process. From the standpoint of the central-station engineer the industrial plant with a large heating or process load certainly does have the appearance of eliminating the condenser loss. However, this is only true in a few cases where the heat in the process steam is actually put into the product and shipped away with the product. Most generally an overall heat balance of a mill which uses large quantities of process steam will show that the heat content of the raw material is about equal to, or if anything more than, the heat content of the finished product. This includes, of course, all chemical heats of combination. The thermal balance of the mill referred to, which burns about 50,000 tons of coal per month, shows that this 50,000 tons of coal has been added to the atmosphere and to the source of cooling water almost quantitatively.

In an almost innumerable number of cases evaporator installations are built to use very low-pressure steam, and in this respect they are entirely comparable to a central station which has a boiler plant operating at 25 or 50 lb. pressure and condensing at about 20 in. vacuum. An evaporator is directly comparable to a boiler and condensing generator cycle.

Almost invariably equipment can be installed which will permit the evaporator to operate on steam of any assignable pressure and temperature, and this installation would substantially decrease the consumption of fuel in the same way that high-pressure boilers for turbines do. This last analogy is an example of a general case where new capital will reduce the consumption of process steam. Industrial plants also suffer tremendously from a lack of analysis of the consumption of the process steam. The writer knew of one instance where a chemical solution was being prepared in one portion of a large plant. This solution was concentrated up to about 50 deg. Baumé and was pumped to various processes within one-quarter mile of the point of preparation. Each of these processes then diluted the solution down, sometimes as low as 34 deg. Baumé, because it was more convenient to use in that particular strength.

By many other examples it can be shown that the consumption of process steam can be enormously reduced, often to such an extent that a central-station analysis together with an acceptance of the industrial process steam requirement may lead to a misexpenditure of money.

Therefore, in making a study for the purpose of recommending a unification of power-generating facilities between an industrial plant and a public utility it is recommended that the heat balance of the industrial plant itself be carefully studied to make sure that the portion of the low-head heat required in process cannot be obtained through some means of recirculation. Mr. Ryan himself once said to the writer that when the average mill is in trouble as to its steam supply there are two general things that can be done. The first is to buy and install more boilers; the second, which can much more frequently be carried out, is to reduce the current steam consumption to fit the existing boiler plant.

In a few words, the writer's recommendation is that the engineer engaged in this type of study work should be careful not to pass the buck of the condenser loss too easily to his industrial power-plant brother.

Author's Closure by Mr. Wood

The author agrees with Mr. Conrad in his suggestion that utility companies should plan for the establishment of a manufacturing district adjacent to large generating stations. Such a plan would reduce to a great extent the cost of transmission and distribution. It is this element of cost of power service that has trended upward while generating costs have continuously been lowered. Such a plan of interconnection would meet Mr. Taggart's criticism as losses of transmission and transformation would be greatly reduced and process steam sent to manufacturing plants. However, it is the existing diversified industrial plants that may be reached by the utility system that offer the greatest potential possibilities of interchange of power.

One of the most gratifying results of the presentation of Mr. Ryan's and the author's papers is the interest aroused among engineers who are actively interested in power problems of industrial plants. The more fully these problems are discussed and mutually understood the better for all concerned. We are all prone to magnify the importance of our relationship to such matters, but it is well to occasionally get the other fellow's point of view.

As a result of these discussions there is one point on which some agreement obtains; viz., the wastefulness of duplication of investment.

Author's Closure by Mr. Ryan

The author is glad to accept Mr. Kates' correction in regard to the efficiency of Diesel engines. However, with Mr. Kates'

[14] Consulting Engineer, Syracuse, N. Y. Jun. A.S.M.E.

figure of 7500 B.t.u. per kw-hr. rejected by the Diesel, there is a marked saving in thermal efficiency over a non-condensing steam prime mover which exhausts to the atmosphere.

The facts brought out in the discussion in regard to the practicability of using treated water in high-pressure boilers are most encouraging to the industrial-power engineer.

The prospect of becoming common carriers instead of manufacturers of power, as predicted by Mr. Randall, may not be very gratifying to the public utilities at the present time. Nevertheless, if Mr. Randall's prediction should be fulfilled, it is possible that the utilities would make more money with less investment than they do now. The author has heard the statement made by a prominent financier that the most profitable electric-light property in New England never owned a power plant.

Mr. Deutsch's comment on the desirability of investigating the process-steam consumption of an industrial plant, before designing a new power plant to provide that steam, is sound advice. By changes in process equipment or in methods of operation, it is frequently possible to save more fuel and more money than can possibly be saved by any power development.

Most of the statements made in this discussion, and in other papers concerning the cost of industrial power, are based on the erroneous assumption that one B.t.u. is worth as much as any other B.t.u. in an industrial process. The power engineer knows that this is not true in his own business, and that a B.t.u. at 700 deg. fahr. is worth almost infinitely more to him than one at 70 deg. fahr. This is just as true in almost any industrial process. It is common practice to credit a non-condensing prime mover with the heat content of its exhaust in direct proportion to the heat content of the live steam at the throttle. This basis of cost accounting does not represent the production cost of exhaust steam, and it establishes a price for exhaust steam which exceeds its value to the user.

It is probably on some such unsound basis of cost accounting that some astoundingly low costs of industrial electric power are obtained.

The discussion of Mr. Wood's paper and the author's has been most gratifying in that it shows an awakening respect for the public utilities' problem on the part of the industrial-power engineers, and likewise it displays an appreciation of the opportunities for cooperation with industrial plants on the part of the public utilities' engineers. Much as the engineers in these two fields of power engineering would like to maintain separate and distinct functions, the laws of economics compel a merging of their problems to produce solutions which best serve the interests of our country and of mankind.

FSP-53-26b

Engineering Aspects of Interchange of Power With Industrial Plants

Establishment of Means and Facilities for Utilizing Massed Resources to Serve the Power Requirements of Individual Industries

By B. F. WOOD,[1] NEW YORK, N. Y.

Economical operation can be effected and overhead costs greatly reduced in a great many cases where public-utility or other steam-generating or hydro power plants find it possible to hook up with large industrial operations. The advent of high steam pressures has largely increased these possibilities and extended the engineering problems involved in their proper solution. The engineer must seek the maximum overall economy with minimum investment having regard for reliability of service, and then work out a plan for the undertaking that will be to the equal advantage of all parties at interest.



INTERCHANGE of power may be secured by means of some plan of interconnection. The term "interconnection" carries the thought of tying together the transmission or other electrical facilities of two or more power systems.

Let us, for a moment, review the accomplishments of interchange of power by interconnection between public-utility systems.

About 25 years ago, in the United States, isolated or independent plants for the supply of electrical service to a community began to give way to the grouping of adjacent systems by the tying together of the transmission systems. This was primarily done for protection of service in emergencies and could be successful only when each company had a mutual regard for the common interest. As the advantages of interchange of power became better understood and appreciated, independent operating companies were brought under a common financial control through holding companies. Interconnection, more intelligent load dispatching, the use and development of the most economical power sources, standardization and interchangeability of equipment, better system planning, and many other advantages logically followed the extension of holding-company control.

Progress in interconnection has reached a point where the principal electric power systems between the Mississippi River and the Atlantic Coast and from the Gulf of Mexico to and across the Canadian boundary line may now interchange power. This territory comprises an area of approximately 900,000 sq. mi., about twice the size of Germany, France, and England combined.

Within a relatively short time interconnecting links will be extended to include other power systems west of the Mississippi, making the area of the then connected systems in excess of 1,500,000 sq. mi., in which there is a population of approximately 98,000,000. Some 20 major power systems and many more of smaller size serve this territory, in which there is a total of about 20,000,000 kw. in power-generating capacity. Each of the major power systems comprises a transmission network within itself, interconnecting from 400,000 to 2,000,000 kw. in power-generating facilities with interconnecting ties having capacities ranging from 25,000 to 125,000 kw. This great spread of interconnection has resulted in benefits to the power systems in the following respects:

1 Greater leeway in the choice of plant sites
2 Decrease in generating capacity due to diversity
3 Decrease in spare generating capacity
4 Increase in size of generating units, resulting in lower unit cost
5 Staggering of new construction
6 Making hydro capacity firm capacity
7 Making possible the use of more economical units.

The choice of plant site for a new generating station is influenced by three major considerations, namely: center of load, availability of condensing water, and the delivery and cost of fuel.

Interconnection tends to eliminate the restrictions due to load. As illustrating this statement, one may cite cases of such stations as Stanton, Deepwater, Trinidad, Toronto, Philo, Gorgas, and many others. All of these stations supply power into a transmission network of considerable extent, thus permitting selection of plant site to secure minimum operating cost because of cheaper fuel and increased available capacity because of an adequate supply of condensing water.

One advantage secured by interconnecting is the decrease in capacity due to diversity. The magnitude of this diversity may perhaps best be illustrated by the following cases:

1 The Southeastern system interconnects the Alabama-Mississippi systems (325,000 kw.) with the Georgia-South Carolina systems (225,000 kw.) and the Tennessee system (136,000 kw.). Diversity in loads adds about 40,000 kw. effective capacity to the combined system, about 5 per cent of the total.

2 Niagara-Hudson system connects the Buffalo System (900,000 kw.) with three systems in the eastern part of the

[1] President, Allied Engineers, Inc. Mem. A.S.M.E. Graduated from University of Arkansas and entered the employ of the Complete Electrical Construction Co. as Chief Engineer. For 15 years he had charge of the construction of steam and electric power stations for the Pennsylvania Railroad. In 1913 he became Vice-President and Chief Engineer of the United Gas and Electric Corporation. Four years later he entered business for himself, first as B. F. Wood, Inc., and later as Wood, Hulse, Yates Co., Inc. This company combined with Republic Engineers to form Stevens and Wood, Inc., which combined with the Dixie Construction Co. and changed its name to Allied Engineers, Inc.

Presented at the Second Plenary World Power Conference, Berlin, Germany, June 16 to 25, 1930. Also contributed by the Power Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

State (375,000 kw.). Load diversity adds between 30,000 and 50,000 kw. to the combined systems, or about 3 to 5 per cent of the total.

3 Pennsylvania-New Jersey interconnection ties Philadelphia Electric Company (950,000 kw.), the Public Service Company of New Jersey (860,000 kw.), and the Pennsylvania Power & Light Company (831,000 kw.). Load diversity adds over 300,000 kw. to the combined system, or nearly 12 per cent of the total.

It becomes possible, therefore, to reduce the total plant investment by 5 to 10 per cent because of system diversity when adequate transmission facilities are provided to distribute to the entire system the necessary reserve or standby capacity.

A further saving results in the decrease in spare equipment necessary to maintain reliability of service. Spare generating equipment in capacity equal to the largest single unit on one system becomes adequate spare for the interconnected system.

Increase in the size of generating units, which becomes possible for plant installations on interconnected systems, makes a substantial lowering of unit costs. A station, either hydro or steam, containing large units may be developed on the system or portion of a system, and later a station developed at another location, thus alternating or staggering the construction program to bring in only the additional capacity properly to meet the growth in load.

The possible saving in capital investment may be illustrated by the Toronto Station. This station in its first construction stage consisted of two 35,000-kw. units, which were installed at an average cost of $128 per kilowatt of capacity. In the next stage two additional 35,000-kw. units were added, making the average cost for the 140,000 kw. of installed capacity $114 per kilowatt. Had this station been constructed as a complete unit, under an interchange agreement, the generating equipment, instead of four 35,000-kw. units, would probably have been two 70,000-kw. units, which could have been constructed at a cost not in excess of $100 per kilowatt. The total cost of the output, including fixed charges on capital investment, operation, and maintenance, would have been 15 to possibly 20 per cent less for the station of larger generating units.

It is perhaps necessary to make some explanation of the statement that interconnection makes hydro capacity firm capacity. The usefulness of a hydro plant is limited to the ability of the system to absorb the capacity and energy that the stream flow as regulated by the available storage may develop. During low stream-flow periods the available capacity of run of river hydro plants may be supplemented by either storage hydro or steam plants, and the capacity of run of river hydro plants then becomes firm or useful capacity.

Many of the larger systems in the interconnected systems referred to are served by hydro as well as steam-generating stations, and to the extent that the systems are not so served the tendency is to interconnect with systems having generating facilities of different characteristics. Steam and hydro stations on a system supplant each other because of different characteristics.

The economy of combining such generating facilities results from the fact that the available output of the hydro station may be more fully utilized. There is thus provided a larger system into which the hydro energy may be poured, making it possible, during the high-water periods, to run the hydro plant to full capacity on base load, and during low-flow conditions the full capacity may be used during the short peak period, placing the economical steam stations on base load.

Large blocks of hydro energy available during high-flow seasons may be utilized to replace steam-generated energy, while during low-flow seasons when but little hydro energy is available the full hydro-plant capacity may be used daily for short periods on the system peaks. In this manner, every drop of water up to the installed hydro capacity may be effectively utilized. Such complete utilization, however, is only possible when the hydro plants are connected into systems having a minimum night or week-end load requirement sufficient to absorb the flood-flow hydro capacity and energy. With large hydro projects such load is rarely available except by extension of the interconnected system over a considerable geographic area without reference to political lines of division.

There remains, however, another condition that must be satisfied to secure the full advantages of the physical possibilities. The administrative direction must be in harmony with the economical advantages of the interconnected systems, a condition fully satisfied through ownership or lease of the hydro plant. In those cases where the ownership of the hydro plant, particularly if on streams where complete seasonal or yearly regulation is not obtained, lies with other interests, a mutually clear understanding of the common interests must obtain. It is extremely difficult if not impossible to express in a contract all conditions to secure the full utilization of all available power. The utility companies transmitting and disposing of the energy must say how and when the hydro plant is to be operated. The operations of the hydro as well as other generating stations must be under the immediate direction of the system load dispatcher unhampered by other restrictions.

The Conowingo (Susquehanna River) development of the Philadelphia Electric Company illustrates this point. The stream flow varies yearly as well as seasonally between wide limits (minimum 3000 sec-ft. to a maximum of 750,000 sec-ft.) and the day-to-day variation is frequently of considerable magnitude. The installed capacity of 250,000 kw. is used during high water at nearly 100 per cent load factor to carry the base load of the Pennsylvania-New Jersey interconnected systems on which the minimum load is approximately 460,000 kw. During low stream-flow conditions this full capacity is used for short periods on peak. In this plan of operation the total annual available output of some 1,350,000,000 kw-hr. is poured into the interconnected systems and profitably utilized. Without this mutually cooperative arrangement a considerable portion of usable hydro energy would be lost to the disadvantage of large numbers of domestic and industrial customers, and, in fact, is actually so lost on large undertakings where the administrative operation is by different authorities.

Interchange of power by means of the physical interconnection of different systems is thus the employment of massed resources. The unification of financial and administrative control extends the advantages of massed resources to other features, such as resources of capital and organized personnel. Larger and better projects may be developed and more intensive and careful study by a skilled organization may be devoted to the improvement of service and betterment of operating results.

The great development of and economic advantages gained to the utility systems of the United States during the past two decades by interchange of power through massed resources have been discussed at some length because, for similar reasons, it is believed that a great opportunity is offered for cooperation between industrial plants and utility systems. The next two decades should witness even greater strides in extending the interchange of power between industrial plants and utility systems through massed resources.

Let us now consider interchange of power with industrial plant; and in this discussion it is the purpose to think of interconnection as the means employed to secure the advantages of massed resources of power production.

Within the area over which this network of interconnected

utility systems extends, nearly 200 industrial concerns have, during the past few years, established new plants, with boiler equipment designed to operate at a pressure of 300 lb. or higher. Nearly two-thirds of this number installed power-generating equipment aggregating approximately 500,000 kw. of capacity. The advent of high pressure has made it possible for the industrial plant to get some or all of its power requirements from the steam that is subsequently used for process or other heating operations. There is, beyond doubt, a very much larger number of industrial plants in operation at less than 300 lb. pressure supplying the multiplicity of power and heating requirements of industrial enterprises. An industrial plant often requires process steam as well as electric power, and frequently some form of waste heat is available. Steam and electric power requirements of an industrial plant do not usually coincide, either hourly, daily, or seasonally, and these requirements do not, except in a general way, synchronize with the liberation of waste heat or the possible production of surplus electric power. A great field of opportunity is here presented for cooperation between utility and industrial companies to bring about a massing of resources of power production.

To secure the maximum degree of reliability and economy of power service and the fullest utilization of any waste heat or usable surplus power some arrangement must be made to bring about a massing of the industrial and utility companies' resources of power production. This is hardly possible if the utility company limits its activity solely to the furnishing of electric service, particularly if the utility company adheres rigidly to an established schedule of rates. A discussion between utility and industrial people frequently starts with a mutual distrust of each other's cost statements, which leads to dickering and may end in the industrial company determining to be wholly independent in its entire power service. Recently, a broader and more liberal view has prevailed, and progress is being made in bringing about an agreement to the mutual advantage of both interests. To some extent, this change of attitude may be due to a better understanding of unity of interests.

Among the means that may be employed to secure the advantages of massed resources of power production, with respect to utility and industrial plants, there may be mentioned the following:

1 The interconnection of the electric circuits securing advantages of protection of service, diversity of load conditions, economy of operation, etc.

2 The establishment of jointly owned plants and facilities, supplying to the industrial plant electric, steam, or other power service, utilizing the utility system as the reservoir from which to draw or into which any and all surplus electric power may be poured.

3 The building and operation of plants and facilities by a utility company near the industrial plant, supplying such power service as required by the industrial plant, making up any deficiency from or taking any surplus electric power on the utility company's power system.

In the first case there are the possibilities of all the advantages of interchange of electric power presented by interconnection between different utility systems with the further advantage, usually, of a different character and greater degree of diversity of load.

It is, however, the last two cases that present the greatest possibility for mutually advantageous arrangement between industrial and utility plants and that make the engineering studies to secure maximum economic results most interesting.

In the establishment of jointly owned plants or the allocation of a portion of the plant facilities to the service of the industrial plant, the total investment may be minimized. Many plant facilities such as real estate, railroad sidings, coal- and ash-handling equipment, etc., need not be duplicated. Spare equipment may serve as spare for both the industrial and utility. The organization for operation and maintenance may be spread over both operations.

A good illustration of the second case is the Deepwater Station, located on the New Jersey side of the Delaware River, just opposite Wilmington, Del. This station is jointly owned by the American Gas & Electric Company and the United Gas Improvement Company, with a portion of the plant allocated to the services of the E. I. du Pont de Nemours & Co.

The first two companies are important utility companies, with a large and rapidly growing electric light and power business. The du Pont company is one of the largest industrial companies, manufacturing widely diversified products and having a substantial and increasing need for both process steam and electric power. The combined additional electric-power requirements of the two utility companies was in the order of some 100,000 kw. demand and 400,000,000 to 600,000,000 kw-hr., annually while the industrial company wanted 200,000 to 400,000 lb. per hour of process steam at 180 lb. pressure. Mutual discussion between the three companies led to the conclusion that they could combine their needs to the economic advantage of all concerned. The power companies organized a company to build and operate a power station for their joint interests and to enter into a contract to furnish steam and electric service to the du Pont company.

After a careful study of plant investment and operating costs the power companies adopted a station designed to operate with a boiler pressure of 1400 lb. The initial installation includes six boilers, each of a capacity of about 330,000 lb. of steam per hour. Two such boilers are allocated to the service of the du Pont company, leaving two boilers, one standard and one reheat boiler, for each of the utility companies. The station is laid out on the unit plan with one standard and one reheat boiler in each of the utility sections and two standard boilers in the center or du Pont section. The initial turbine installation consists of two cross-compound units, each of a capacity of 55,000 kw. The high-pressure elements of these compound units are identical in design with the 12,500-kw. unit in the du Pont section. The exhaust from the du Pont unit, however, which at maximum load amounts to 530,000 lb. per hour, goes directly to evaporators which furnish 400,000 lb. of steam per hour at 180 lb. pressure.

Steam from the evaporators passes through live-steam superheaters bringing the temperature up to 440 deg. fahr. before it is sent out on the du Pont lines. The condensate from the evaporators is returned as feed to the boilers, making a closed cycle. The du Pont company returns or makes up the condensate of its steam supply, and to the extent that the return is at high temperature less heat is required in the evaporators.

The essential features of the contract between the utility companies and the du Pont company are as follows:

1 The du Pont company pays in monthly instalments a sum representing a reasonable return on the investment in that portion of the plant facilities allocated to their service.

2 It pays the proportionate share of all direct operating and maintenance costs, including fuel.

The du Pont company is thus entitled to all electric power generated by the steam that passes through its high-pressure turbine unit, and should its need for process steam be in excess of that required to produce the necessary supply of electric power, the excess power flows into the utility system and is credited to the du Pont company. On the other hand, should its need for electric power at any time exceed the production of the generating unit, power will be drawn from the utility company's system.

The second element of the du Pont company's monthly charge is determined by the ratio that the total heat supplied to its generating unit bears to the total heat supplied to all generating units. As a result of this arrangement it is given full access to all station-operating records and may even purchase or designate other sources of fuel if it believes lower operating costs would result.

The combined full utility and industrial load will require the operation of five boilers, leaving the sixth as spare for all companies. Furthermore, either utility company may, at any time, take any unused capacity of the other, paying only the proportionate operating and maintenance costs. Thus, to the extent that their respective loads do not synchronize, each company has the privilege of fully loading the unused capacity of the other without increasing its share of capital expense. An incentive is thus created that should result in keeping the station as nearly fully loaded at all times as possible. Should this occur, the net effect will be to the benefit of all parties, as in this manner the unit station-heat costs will be less because the total cost will be spread over a larger output.

This is believed to be one of the largest instances, if not the first instance, in this country of utilizing the massed resources of utility and industrial companies for power production. The successful working out of this arrangement should open up many new opportunities for the engineer in the problem of interchange of power between utility and industrial plants.

As an example of the third class—that is, of the utility company owning and operating the plant and facilities—may be cited the case of a steel company wishing to change its rolling mills and other steam-engine-driven equipment to electric-motor drive. The steel company produces finished products in an amount of approximately 90,000 gross tons per month. It was found that, after making use of coke-oven and blast-furnace gas in heating furnaces and other metallurgical operations, sufficient gas was available to supply the steel company's electric-power requirements after complete mill electrification. The steel company's electric-power requirement will then be about 300,000,000 kw. annually, with a maximum demand of 54,000 kw. The available waste gas will produce, over and above the steel company's requirements, a surplus of some 40,000,000 kw-hr. annually. The steel company either would have to install complete spare equipment or relay its electric-power service from the utility company's power system, thus either materially increasing its annual capital charges or adding to its operating charges for reserve or standby service.

An industrial company such as the steel company is usually not prepared to make large capital investments, except under pressure of necessity, unless the annual return on capital investment is greatly in excess of that required in the diversified power service of a utility company. This situation led to the suggestion that the utility company build, own, and operate the power plant and facilities, purchasing the waste heat from the steel company at substantially the equivalent cost of other fuel, and furnish the steel company with its entire electric-power requirements on a cost basis to yield a proper return on the investment.

The utility company could build a station designed to operate at 1400 lb. pressure, instead of 400 lb. as contemplated by the steel company, providing no spare generating equipment. The utility company could relay the steel company's service with the spare equipment required for its other service. Such a plant, in addition to supplying the steel company's requirements, could generate some 100,000,000 kw-hr. annually at a cost that could advantageously be poured into the utility company's system. The net result will be to furnish the steel company's entire electric-power requirement at a cost comparable with any cost to itself and to save the steel company an investment of something over $6,000,000 in power-station facilities and to increase the annual return on the monies to be spent by the steel company on its electrification program from about 20 per cent to about 25 per cent. While as yet no actual contract has been consummated to carry out this program the general principles have been accepted pending the development of details. The resources will be massed, no usable product wasted, and capital investment conserved.

A case now under consideration is that of a large manufacturer using some 3,000,000,000 lb. of steam per year for process and heating and at the same time buying and generating electric power up to a maximum demand of 42,000 kw. The manufacturer is faced with the necessity of abandoning one of two large boiler plants, and being thus faced with rehabilitating his steam and power supply, can effect substantial gains by building a high-pressure boiler plant and generating by-product power. Here again even greater overall results can be obtained by the public-utility company building the new plant and serving both the steam and electric-power requirements, primarily because the utility company can make a materially greater use of the investment in plant facilities.

Another case recently under consideration was a group of stockyard and packing-house companies whose plants are located close together. Their combined maximum demand for steam is almost 2,000,000 lb. an hour, now served from eight relatively low-pressure boiler plants, while they generate or purchase about 130,000,000 kw-hr. a year. Estimates of a central high-pressure steam station producing by-product power, including, of course, fixed charges on the new investment, show very substantial economies over the present operating costs, but the point to emphasize is that the overall saving is a maximum when the public-utility company is associated with the new enterprise, as this permits minimizing investment in spare or reserve plant capacity; it permits maximum economical use of all by-product power; and it supplies, in the most economical manner, electric power which at times is required in quantities greater than is available from the steam then used.

The economic advantage of expanding high-pressure steam through engines or turbines to make by-product power before the steam is used in process work is being lessened by two factors, namely: the constantly improving thermal performance of modern large turbine installations and the higher cost of boilers, evaporators, and other equipment required for by-product power production with high steam pressures.

Only a few years ago, a station thermal performance of 20,000 B.t.u. per kw-hr. was regarded as excellent. Many large central-station plants are now being designed for approximately 12,000 B.t.u. By-product power may be obtained for about 5000 B.t.u. per kw-hr., but in high-pressure plants evaporators are usually necessary to insure scale-free water in the boilers, and as much as 6000 B.t.u. per kw-hr. may be required. The ratio has thus been changed from about 4 to 1 to 2 to 1. If to obtain by-product power it becomes necessary to make large capital investment or if the rate of annual fixed charge is high, the economy of an independent high-pressure plant to supply the service to an industrial company is thus greatly reduced. The economic results of a station supplying combined service to an industrial plant and a utility system may, however, be quite improved and in any event should be better than is possible if each develops independently.

The principles of contractual relations between the industrial and power company are simple, but their application may be complex.

In principle the industrial should pay the power company a reasonable rate of return on all investments made on its account and also pay its proper share of all operating costs. In

practice it is often desired to reduce the payments to unit rates for steam and electric power.

While it is relatively simple to ascertain the probable costs for any given set of operating conditions, it is difficult to determine unit rates which will be equitable at all probable ranges and conditions of electric load and steam flow.

For instance, if by-product power from steam be involved, then the combined unit cost of saving both steam and electric power is a minimum when the investment in steam capacity is fully utilized and when the electric-power requirement is just that of the by-product power available from this amount of steam. Any unbalanced conditions from this exact relation between the quantity of steam and of electric power will increase the combined unit cost. Hence, the unit rates for each should bear some relation to the quantity of the other, and in the limit be equitable on the one hand for furnishing steam only a portion of the time and on the other hand for furnishing electric power only a portion of the time. This requires a very careful survey of all probable operating conditions and a clear understanding between the engineers of both companies.

Many other examples of the interchange of power with industrial plants might be cited. There are undoubtedly thousands of opportunities for the establishment of means and facilities to utilize massed resources to serve the power requirements of industrial plants. The advent of high steam pressures has largely increased these possibilities and extended the engineering problems involved in the proper solution. The engineer must seek the maximum overall economy with minimum investment, having regard for reliability of service, and then work out a plan for the undertaking that will be to the equal advantage of all parties at interest. These problems are of special interest because of the many new and unusual factors involved.

[NOTE: Discussion of this paper jointly with that of W. F. Ryan by Edgar J. Kates, Charles W. Conrad, James M. Taggart, Guy B. Randall, R. J. S. Pigott, C. F. Hirshfeld, and Z. G. Deutsch will be found following the Ryan paper.]

FSP-53-27

Combined Burners for Firing Waste Fuels

By R. C. VROOM,[1] NEW YORK, N. Y.

Waste by-products of oil refining, formerly difficult of disposal, are now burned in special plants to furnish electric and steam power. This is made possible by the design of high-capacity burners, some liberating 100,000,000 B.t.u. per hour. The author describes a wide-range mechanical atomizer of the circulating type and the results of an installation at the Louisiana Station at Baton Rouge of the Louisiana Steam Products, Inc. The burners are successfully firing crude or lubricating (acid) sludge, soda bottoms, neutralized sludge, acid tar, wax tailings, and flux bottoms, in addition to fuel oil.

IN certain industrial processes, and in particular in oil refineries, there are obtained various combustible by-products or residues which can be used as fuel provided that suitable combustion equipment is employed. In the past this utilization has been confined largely to those plants in which such materials were produced. In some cases their combustion has proved so difficult that it has been impracticable to consume them in such a manner. As a result they sometimes have collected in considerable quantities, and their disposal has represented a serious problem. In an article by Dr. A. E. Dunstan, chief chemist of the Anglo-Persian Oil Company, he says when referring to petroleum coke: "In the majority of refineries no immediate use for the coke could be found, and there arose again the question of storage space, almost acreage, as was the case with acid sludge."

Within recent years waste fuels have become of increasing importance, and not only more effective efforts to burn them have been made in the plants where they are produced, but often they are given some treatment to minimize their undesirable characteristics, and in this treated form the better grades are being furnished to outside plants.

In some instances public-utility companies have found it economical to construct a steam station adjacent to an industrial plant in order to make use of waste fuels, in which case the steam station supplies to the industrial plant process steam and electric current which the latter previously had produced itself.

The combustion of residual or waste fuels requires special attention, as fuels which heretofore have not been considered commercial, now are occupying a position of increasing importance.

In oil refineries the use of such fuels has been coupled with the necessity for a great increase in heat liberation per furnace, per unit of furnace volume, and per burner, due to the fact that these fuels often must be used for firing larger units requiring a higher rate of heat absorption per unit of heating surface. Also, modern refining processes frequently are sensitive to furnace conditions and necessitate a greater use of radiant-heat transfer with a minimum of excess air. In such cases absolute control of flame shape and other furnace conditions is essential.

The use of a few flexible and economical high-capacity burners for fuel firing, in contrast to a great number of smaller units, is in accordance with the modern trend of design for fuel-firing and other plant equipment. Burners are available in sizes to liberate up to 100,000,000 B.t.u. per hour when burning practically any liquid, gaseous, or pulverized residual fuel, without any sacrifice in economy or flexibility, and with the ability to give absolute control over the furnace conditions.

Often such burners are used with highly preheated air. In oil-still installations, where there have been great advances in the methods employed, the use of preheated air has been developed beyond the range ordinarily found in power plants, temperatures approaching 1000 deg. fahr. sometimes being employed. Such temperatures of course require the use of insulated panel burner register fronts and special metals, not only for the register parts, but for air preheaters and other equipment in contact with unusually hot gases.

The higher air temperatures increase the forced-draft pressure necessary at any given fuel rate per burner, and while the use of highly preheated air speeds combustion, the most careful burner design is required. Not only must air pressures be maintained within reason, but at the same time the correct flame shape and the other desirable characteristics which the burner had when operating at lower temperatures must not be sacrificed. Maximum turbulence is essential, as each particle of fuel must come in intimate contact with the greater volume occupied by the air necessary to burn it.

Not only is it necessary to have individual burners which are capable of handling standard as well as waste fuels at high capacity, but it is essential to have flexible combined units for two or three fuels, these being capable of obtaining maximum capacity with any one of the fuels. Fig. 1 shows part of a boiler room having six 3400-hp. boilers, each equipped with 20 combined gas and oil burners similar to the design shown in Fig. 4. In another boiler room of this station are eleven 1500-hp. boilers, each equipped with 15 of these burners, the total number of burners of this type in the plant being 285.

The word "combined" as used in this connection must be taken in its proper sense, or else the full advantage of a combined burner will not be indicated. Until a few years ago combined burners generally meant a multiplicity of burners for liquid, gaseous, or pulverized fuels "combined" in the same furnace. What is meant in this paper by a "combined burner" is a burner in which any one of three forms of fuel, liquid, gaseous, or pulverized, may be fired through the same throat opening into the furnace, and in which use is made of a common passage for the combustion air.

Combined units are advantageous for various reasons. In certain plants they are needed due to the fact that prices of the different types of standard fuels may fluctuate widely from time to time, and the use of such equipment enables the plant to burn that fuel which is the most economical. In some industries there are obtained from the various processes numerous waste fuels which are of such a character as to have practically no market value, but the supply of such fuels is irregular or inadequate to carry the entire plant load continuously, with the result that provision also must be made for burning standard fuels.

The advantages of combined burners over independent burners for the various types of fuels may be summarized as follows:

---

[1] Chief Engineer, Peabody Engineering Corporation. Assoc-Mem. A.S.M.E. After graduation from Stevens Institute of Technology in 1922 with a degree of M.E., the author spent a short period of service in the Engineering Department of the Western Union Telegraph Company. He then entered the employ of the Peabody Engineering Corporation, being engaged in problems related to the combustion of liquid, gaseous, and pulverized fuels.

Presented before a meeting of the Metropolitan Section, New York, N. Y., Jan. 20, 1931, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

1 The ease of operation due to a smaller number of burners.

2 The simplicity of furnace and air-duct construction and air control, due to the location of all burners in a single group in a common windbox, so that all fuels are fired through one furnace wall and with one set of throat openings into the furnace. The latter is of particular importance when water walls are used.

3 The ability to keep in operation at all times all of the burners firing any one furnace, thus eliminating the necessity for protecting the burners not in service.

4 The ability to light one fuel quickly from the other, permitting a practically instantaneous change in fuel without

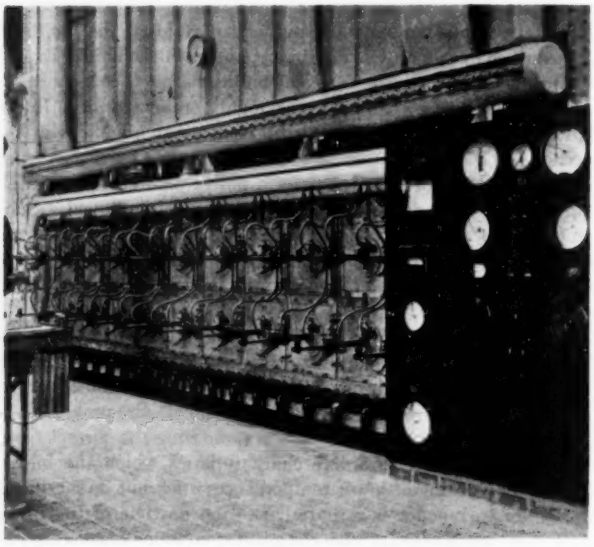


FIG. 1 PART OF BOILER ROOM EQUIPPED WITH COMBINED GAS AND OIL BURNERS

loss of output from the unit being fired. This feature makes a standby fuel instantly available.

5 The ability to burn pulverized fuels of very low volatile content or fuel containing considerable water, as ignition can be maintained, when not otherwise possible, by also burning simultaneously a small quantity of another fuel such as oil or gas.

Fig. 2 shows a convenient and reliable form of high-capacity combined burner for firing three fuels. It is a true combined burner in that all three fuels may be burned simultaneously. It is not necessary to remove a part of the burner for firing one fuel in order to insert a part of the burner for firing another fuel. The combustion air for the fuel being fired sweeps over and cools those parts of the burner for firing the other two fuels. A changeover can be made almost instantly.

The gaseous fuel is admitted to the furnace throat in an annular stream issuing from the inner periphery of the ring-shaped gas chamber immediately adjacent to the furnace throat opening. The gas impinges on the rapidly rotating column of air from the curved blades of the air register.

The pulverized fuel and carrier air are admitted tangent to the inner periphery of the rifled coal-burner body located coaxially and adjacent to the gas chamber. This stream is picked up and mixed with the secondary air from the register rotating in the same direction.

The liquid fuel issues from the wide-range mechanical-atomizer tip in a diverging conical sheet which just clears the furnace-throat surface. All the rotating combustion air from the register must pass through this sheet.

Fig. 3 shows a similar combined burner for lower capacities. In this burner the coal is admitted tangentially into a ringlike, volute-shaped coal chamber and delivered to the furnace throat through an annular slot in the inner periphery of the chamber. The gas chamber is located between the coal chamber and the furnace front plate.

In the commercial application of these combined burners for firing waste fuels numerous combinations of gaseous, pulverized, and liquid fuels of various types are encountered.

In addition to natural gas, many kinds of industrial gases are being fired by such units, among these being refinery-still gases. Such gaseous fuels sometimes have a heat content per cubic foot as high as 2200 B.t.u. When burning these gases, it is sometimes difficult to obtain the best-appearing furnace conditions, due to the presence of a large percentage of the heavier hydrocarbons, and also because there may be present considerable quantities of raw distillate in suspension. Of course, whenever possible, the raw distillate should be properly removed in a separator as its irregular entrainment in the gas continually changes the heat liberation of the burner and makes steady furnace conditions impossible. This is accentuated when operating with low excess air. It is obvious that for the proper combustion of such fuels, rapid and complete mixing of the fuel and air is essential.

Among the waste pulverized fuels being fired by these burners are various forms of petroleum coke. Depending upon the process from which the coke is derived, it may contain from 4 to 12 per cent of volatile matter and varying percentages of sulphur and ash. However, the ash content is normally low and its fusing point high, although in certain instances fuels containing ash having a fusing point as low as 2000 deg. fahr. are encountered, and proper provision must be made to handle these. Refinery coke is often less friable than ordinary bituminous coal, but it is burned readily in pulverized form unless its composition approaches that of a pitch.

When burning a low-volatile fuel particularly in a water-cooled furnace, the combined burner is very desirable as it provides

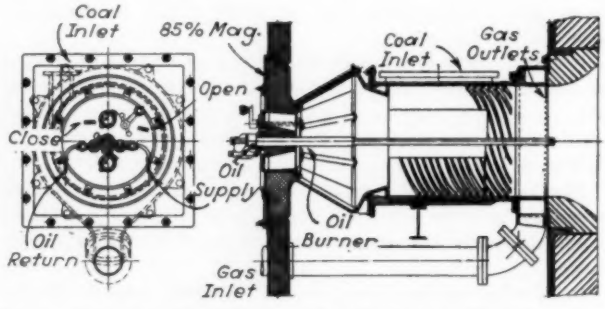


FIG. 2 COMBINED COAL, OIL, AND GAS BURNER OF HIGH CAPACITY

an auxiliary fuel which is available immediately, and in such a way that the control and operation of the plant are in no way complicated. In burning such pulverized fuels, it is necessary that proper combustion equipment be used, or else delayed ignition will occur and the combustible loss due to unburned carbon in the fly ash will be excessive.

The liquid fuels which are encountered often are of high specific gravity and contain a considerable amount of solid matter, carbonaceous or otherwise, in non-colloidal suspension. Mechanical atomization long has been recognized as the most economical method of burning liquid fuel, but not until recently has it been considered practicable for burning fuels of this nature.

Apparently one of the reasons for the previous inability to handle acid sludge in particular, and refinery sludges in general, in mechanical atomizers, was because capacity control

by the use of inlet pressure variation was employed. Furthermore, it had been taken for granted that the mechanical atomizer must handle the fuel under exactly the same conditions as the steam atomizer, and due to the high standard of operation expected from the mechanical atomizer, it was at the same time required to produce far better results before being considered successful.

It is well known that ordinary fuel oil cannot, in all cases, be handled with the mechanical atomizer with as little treatment as is required for the steam atomizer, and it is generally admitted that the slight additional preparation is warranted by the superior furnace conditions and better net economy with the mechanical burner. The proper procedure for economical sludge firing with mechanical burners for such fuels is therefore obvious.

Extensive tests indicated that with a little preparation refinery sludges could be burned satisfactory in the Peabody wide-range mechanical atomizer of the circulating type. As a result of these tests, an installation was made at the Louisiana Station of Louisiana Steam Products, Inc., at Baton Rouge, La., and the operating results have more than borne out the experimental data. The burners at this point are successfully firing crude or lubricating (acid) sludge, soda bottoms, neutralized sludge, acid tar, wax tailings, and flux bottoms, in addition to fuel oil. Combined burners are employed, as it also is necessary to fire pulverized and gaseous fuels.

The circulating type of wide-range mechanical atomizer is firing grades of asphalt pitch, which, while solid at atmospheric temperatures, melt at temperatures up to 125 deg. fahr., depending upon the source. When such pitches are delivered from the process to the burners at temperatures varying from 300 to 600 deg. fahr., their viscosity is sufficiently low to enable them to be atomized properly. The use of fuels delivered at the burners at such elevated temperatures has made it necessary to use special materials for the atomizer parts subject to pressure. Such atomizers also are firing other liquid residual fuels encountered in the petroleum industry.

The ability of this design of mechanical atomizer to handle such fuels successfully is due largely to the wide-range feature of control. This feature ordinarily is used to obtain a continuous and positive control (it is not a "high-low" burner) over a wide range of burner capacity, without reducing the fineness of atomization and without the necessity for changing atomizer tips. This is of particular importance in a plant with a widely fluctuating load when firing into a "black" furnace where the method of changing the supply pressure to take care of load range leads to coarser atomization at the lower capacities. Due to the existence at these lower capacities, of a lower heat liberation per unit of furnace volume, furnace conditions tend to become poorer under the best of circumstances, and coarse atomization seriously accentuates the trouble. When such conditions are coupled with increased burner capacities and the use of waste fuels which require the maximum fineness of atomization at all capacities to produce the best results, and which are far more sensitive to variation of supply pressure than ordinary fuel oils, it can be seen that the ability to operate over a wide range without affecting the atomization is almost essential.

In the case of heavy liquid fuels containing a considerable amount of foreign matter in non-colloidal suspension, the wide-range feature providing circulation has the very important added value that all of the fuel entering the burner is not necessarily sprayed. In this way a continuous circulation of fuel through the atomizer tip of the burner can be maintained at all times, part of the fuel supplied entering the furnace and part being returned to the tank. All dead ends are eliminated, the atomizer tip is cooled by the excess amount of circulating fuel, and the velocities can be kept sufficiently high to prevent such solid matter as is present from settling out and plugging. Tendency to carbonize in the atomizer tip is greatly reduced due to the additional cooling effect of the circulating fluid.

The mechanical-atomizing burner as an economical liquid-fuel-combustion unit has been discussed in various papers by E. H. Peabody, and this quality was brought forcibly to the attention of the engineering profession by data contained in the

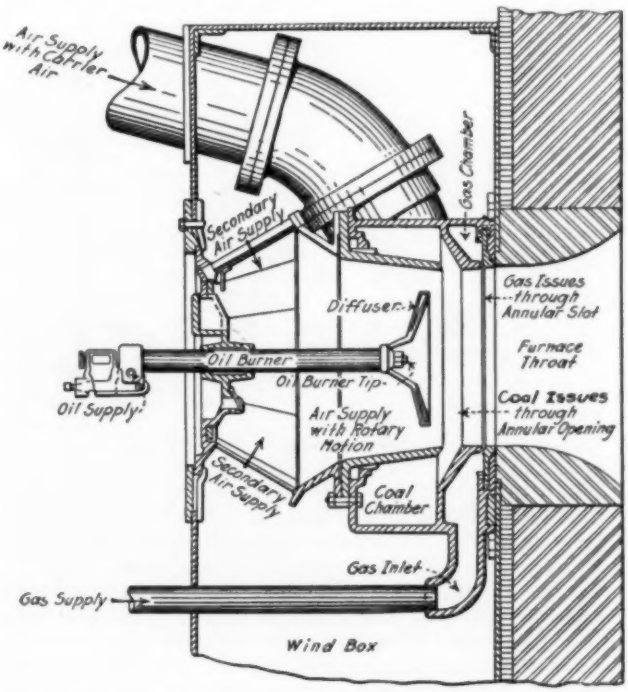


FIG. 3 COMBINED COAL, OIL, AND GAS BURNER OF LOWER CAPACITIES

paper on "Recent Developments in Oil Burning," presented at the forty-third convention of the National Electric Light Association, at Pasadena, Calif., May 18–22, 1920, by D. S. Jacobus and N. E. Lewis. While it is true that improvements have been made in both steam and mechanical atomizers since that time, the same general conclusions still apply as to the superiority of the mechanical atomizer. Whether the steam atomizer be of the flat-flame or round-flame type, there remains the objection that the amount of steam used for atomizing reduces the net efficiency below that possible with the mechanical burner and lowers the dew point of the exit gases. Furthermore, in the case of round-flame steam atomizers, the conclusion that the steam atomizer is less subject to stoppage than the mechanical atomizer, due to its larger passages, cannot be fully justified.

The wide-range mechanical atomizer of the circulating type can be changed to a steam atomizer merely by changing the atomizer tip. In this form one atomizer has fired successfully approximately 9000 lb. per hour of acid sludge. Gas atomization by the use of high-pressure refinery gas also has been used to produce corresponding heat liberations, but as an atomizing agent, gas is not quite so effective as steam. However, as this gas is often available at high pressure and must be disposed of, the quantity of atomizing gas is of relative unimportance, and every bit of atomizing agent acts as fuel contrary to conditions obtaining when using steam for this purpose. In the case of steam the most desirable furnace appearance often is obtained with an abnormal amount of atomizing steam, and if the operator is not careful, the quantity may reach an excessive figure.

Combined burners of the type shown in Fig. 4 have been purchased to fire two 600-lb. pressure boilers at the new steam station of the Humble Oil and Refining Company at Bayton, Tex. Each burner will fire approximately 4000 lb. per hour of fuel oil, and each boiler is to produce 350,000 lb. of steam per hour. In addition to this, there will be burned various refinery sludges and natural gas. These burners are combined gas and oil burners. They are similar to the three-fuel burner shown in Fig. 1, but Jackson & Moreland, of Boston, are the engineers for this project.

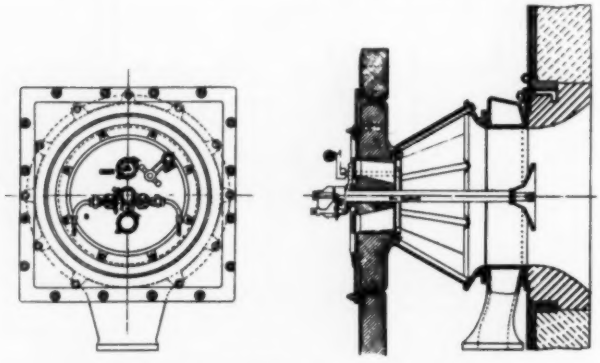


FIG. 4 COMBINED GAS AND OIL BURNER

An excellent example of a station in operation where various waste fuels are used to full advantage is the Louisiana Station. There combined burners similar to those shown in Figs. 2 and 4 are firing successfully various kinds of refinery sludges, petroleum coke, natural gas, and fuel oil, under four boilers, each having a capacity of 350,000 lb. of steam per hour, the boiler feed being treated Mississippi River water. Maximum operation represents about 850 per cent of rating based on the boiler heating surface. Stone & Webster Engineering Corporation, of Boston, are the engineers for this project.

This station is a typical public-utility plant, utilizing the waste fuels from a refinery and supplying to the refinery process steam and electric current, which previously the latter had produced in its own plant. It was described briefly in a paper presented before the Boston section of the A.I.E.E. on Nov. 18, 1930, by H. J. Klotz, Stone & Webster Engineering Corporation, and will be further discussed by him on Feb. 11, at the National Fuels Meeting in Chicago.

James F. Muir also has written a very interesting article[2] regarding the fuels at this plant. As this station is the first of its magnitude to be arranged for waste-fuel consumption in combined burners, its design and operation are of great interest to combustion engineers, and without a discussion of its fuel problems this paper would be incomplete.

The proportion of the heat supplied by the different fuels varies considerably, but averages about as follows, the fuel oil, which is used only for standby purposes, being neglected: 50 per cent natural gas, 33⅓ per cent refinery sludges, and 16⅔ per cent petroleum coke. About 85 per cent of the sludges contain from 30 to 50 per cent of dilute sulphuric acid, which gives them an actual acid content of about 12 to 20 per cent. Approximately one-half of the remaining 15 per cent of the sludges is strongly alkaline.

Some of the sludges fired at the Louisiana Station are more or less readily "burnable," but lubricating (acid) sludge, proper, has in many cases caused considerable difficulty, and the equipment heretofore developed in the refineries for burning it has been designed primarily for the purpose of disposing of the fuel and not with the main object of obtaining efficient combustion. The combustion of the sludge usually has been the most convenient means of disposal, and the mere ability to burn it has been considered an event.

This sludge is extremely variable in its characteristics, depending not only upon the initial crude stock, but also upon the method of treating that stock, including the amount and type of oil used for fluxing, as well as the method of agitating the flux with the sludge. Sludge often contains a large amount of suspended carbonaceous material, and one acid sludge may be practically solid and another rather fluid. It varies from day to day and from hour to hour. Some sludges have a heating value as high as 17,500 B.t.u. per lb., while others have a heating value as low as 8000 B.t.u. per lb. This wide variation is accounted for largely by differences in the amount of flux which must be added to make the material flow and by variations in the weak acid content. This latter often will run as high as 40 per cent. One sample of acid tar (which is considerably more fluid than the acid sludge) contained 54 per cent weak acid, i.e., acid and water, of which 38 per cent represented the acidity, which means that the tar contained 21 per cent of actual sulphuric acid.

The gravity of acid sludge may run from 5 to 14 deg. Baumé. The viscosity of the material cannot be determined, as the heavy and light components settle out in the viscosimeter, giving readings which are not at all representative of the true internal friction.

Lubricating acid sludge usually must be treated. This treatment ordinarily consists of fluxing with oil or tar to reduce its viscosity. Sometimes a sludge cannot be made "burnable" by fluxing in this fashion and must have a caustic material added to it. This process changes it from acid to neutralized sludge. The proper viscosity cannot always be obtained by additional heating, as some sludges tend to foam at a temperature slightly above 200 deg. fahr.

Sludges usually are kept in agitation in suitable lead- or brick-lined tanks by jets of steam or jets of steam and air. Quite possibly some method of mechanical agitation could be used instead of steam, but up to the present time this has not been done. In some plants it has been found impracticable to use tank heating coils, as the sludge carbonizes on the outside of the coils, reducing the heat transfer. Furthermore, agitation seems desirable, such as is now accomplished by the steam jets.

The preparation of the sludge varies with its character and u...ers in various refineries. No definite rules can be given for its treatment. As to whether it is properly fluxed or sufficiently agitated, only visual observation of experienced operators can determine. About the only definite statement which can be made about it is that a properly prepared sludge will be smooth and not too granular.

Improper treatment will cause coking. Any coke particles which are precipitated from the sludge tend to coagulate and form larger lumps of coke. The acid in the sludge appears to hold the coke in a sort of semi-colloidal suspension, and in certain cases, even if the sludge be boiled with heavy acid after the coke particles are precipitated, they will not again be absorbed by the fluid, but when boiled with a neutralizing agent, in some instances they can be made to return into suspension.

While the sludge itself is highly corrosive, due to its sulphuric-acid content, it often seems to be more so when mixed with steam, as is indicated by the wear on steam-atomizer nozzles. This may be accentuated by high velocity and the increased corrosive effect due to the presence of the steam at high temperature and to radiation from the furnace.

In firing boilers with acid sludge or acid tar it is necessary to keep the exit-gas temperatures above the dew point in order to prevent acid corrosion of the flues and certain portions of the air-heater surface. At the Louisiana Station no acid fuels are burned

[2] *Power*, Jan. 13, 1931.

when the boiler capacity falls below a value giving a final gas temperature below about 275 deg. fahr.

With some sludges slag is deposited on the furnace bottom and on the boiler tubes, similar to the deposit obtained when firing pulverized fuel with relatively low-fusing-point ash.

The alkaline sludges such as soda bottoms and neutralized sludge do not attack ferrous metals, so that the usual materials are satisfactory for piping and atomizer parts. Also, these sludges are generally more easily fired and less variable in their characteristics than the acid lubricating sludge. Brickwork difficulties when burning alkaline sludges sometimes are experienced. Furthermore, with alkaline sludges salt may be deposited on the boiler tubes, superheater, and air preheater. This may result in excessive draft losses unless ample soot-blowing equipment is provided.

In the installation at Baton Rouge, each burner is designed to handle approximately 2200 lb. per hour of liquid sludge or 1600 lb. of fuel oil. In addition to the acid sludge, acid tar, neutralized sludge, and soda bottoms, there also are burned wax tailings, flux bottoms, natural gas, petroleum coke, and fuel oil.

The liquid fuel piping on each boiler front consists of two sets of headers, one of steel for the neutral or alkaline fuels and one of brass for the acid fuels. All burners handling liquid fuels are piped to fire fuel oil which may be delivered in the steel header. The two bottom rows are piped to take their supply either from the acid or the alkaline header, while the top row takes its supply only from the alkaline header. Each burner is equipped to fire gas.

Each of two of the four boilers is equipped with 18 combined gas and liquid fuel burners in three horizontal rows of six each. The two lower rows of burners handle acid sludge. The top row handles neutral or alkaline fuels. All burners, of course, can be used for fuel oil or gas. The sludge supply header is

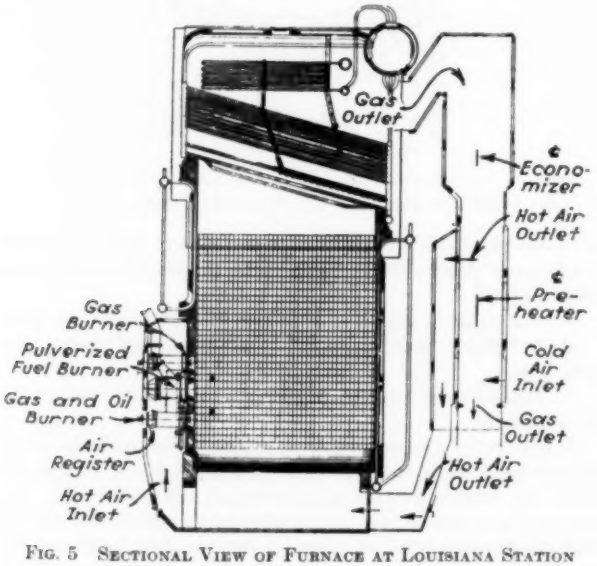


FIG. 5 SECTIONAL VIEW OF FURNACE AT LOUISIANA STATION

connected to the circulating pipe leading back to the tank to eliminate dead ends in the supply header and to prevent the deposition of coke therein.

Each of the second two boilers is fitted with 17 burners. There are six combined gas and liquid fuel burners of the usual size in the bottom row. In the second row are four of these combined burners and also two Toronto type combined pulverized-coke and gas burners in the center of the row. In the top row are five combined burners. The two lower rows are for handling acid tar, oil, and gas, and the top row for alkaline fuels, oil, and gas.

Gas is always fired in some of the burners. This is because the amount of sludge is insufficient to provide for the total steam output. Also previous refinery practice seems to indicate that it is better to fire gas and sludge together, although this is not always done. The burners at Baton Rouge fire through the vertical front wall of the furnace, and no ignition arches are provided. The interior surface of the furnace is water-cooled, part of the

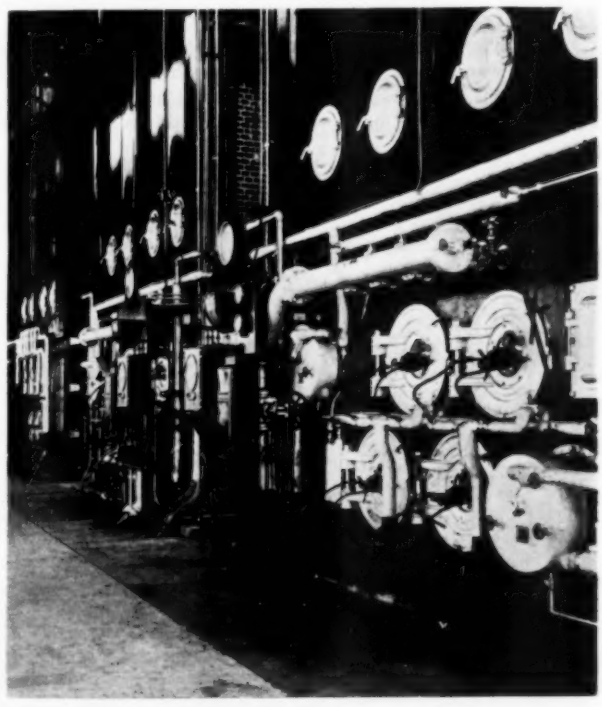


FIG. 6 PLANT IN WHICH DUBB'S RESIDUUM AND STILL GAS ARE FIRED

(The change from one fuel to the other is made many times a day.)

cooling tubes being covered with refractory material. Fig. 5 is a sectional view of the furnace.

As might be expected in a plant of this magnitude where the conditions are so new and the quantities so variable, some operating difficulties have been encountered. However, the author can say from his experience that they have been surprisingly few and far less than had been anticipated.

In so far as the combustion and furnace conditions are concerned, results are satisfactory. It sometimes happens when lighting "off" burners in handling a new batch of sludge that the first atomizers installed will plug due to material which has collected behind the stop valves connecting the burner to the fuel header. However, once in operation, the burners will stay clean for periods comparable with those between cleanings in a plant using ordinary fuel oil.

The question of the ideal material for burner tips has not been settled definitely. The difficulty of this problem will be realized when it is stated that, in one plant using steam atomizers, brass tips have been eaten away in 48 hours. The tips originally supplied seem to have given the best service, although this is not so good as is obtained when firing fuel oil. The burner tips are being used as a convenient "laboratory" in which to test materials for other parts of the plant subject to corrosion, such as pump liners, heater tubes, valve trim, etc. Substantial progress has been made, and new materials are in sight which give promise of being entirely satisfactory.

When it is considered that sometimes pulverized coke, sludge, and gas are being fired simultaneously in one furnace, it is obvious that the combustion problem is not a simple one. The fact that the operation has been carried on so successfully is a tribute, not only to the designers and builders of the plant, but also to the operators.

This plant is the only one of many plants where combined burners are being used for firing waste fuels. In other plants waste fuels which have been successfully fired in Peabody wide-range mechanical atomizers and gas burners are as follows:

1 Dubb's residuum, containing up to 30 per cent of base settlings and water and nearly 5 per cent of sediment by extraction.

2 Still gas, containing up to 10 per cent of hydrogen sulphide.

Both of these fuels (Dubb's residuum and still gas) are fired in the same plant, the change from gaseous fuel to liquid fuel being made many times a day. Fig. 6 shows the plant in which these fuels are fired.

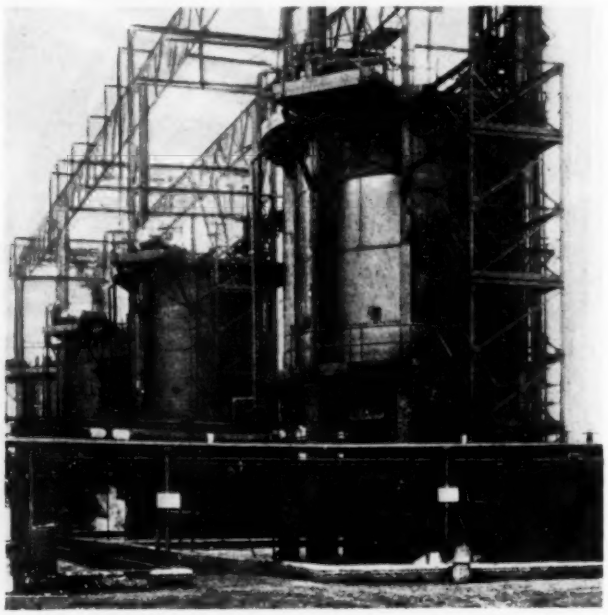


FIG. 7 VERTICAL HEATERS FIRED FROM TOP WITH COMBINED GAS AND OIL BURNERS

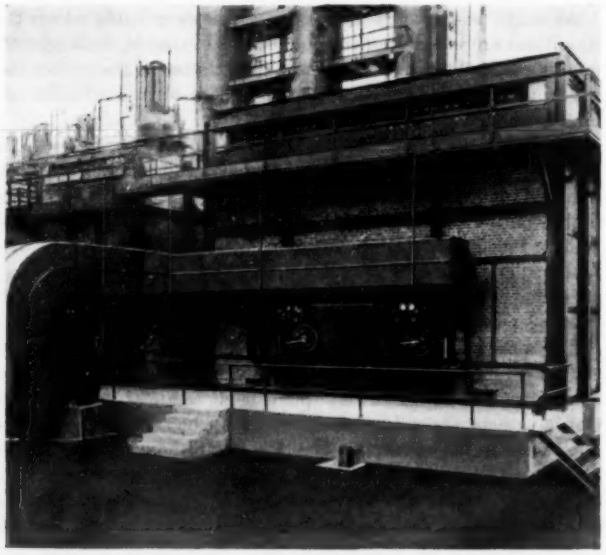


FIG. 8 SECTIONAL VIEW OF FURNACE FIRED FROM BOTTOM WITH SINGLE COMBINED STILL-GAS AND YARD-OIL BURNER

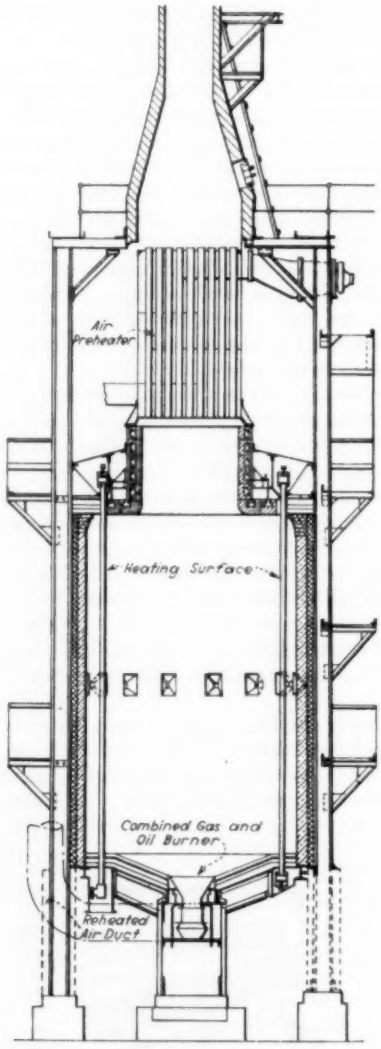


FIG. 9 COMBINED GAS AND OIL BURNERS FIRING A TUBE STILL

3 A mixture of gas oil and Holmes-Manley cracker bottoms.

4 A 125-deg. melting point asphalt pitch, delivered at 700 deg. fahr.

5 Distillate containing a considerable amount of solid matter in suspension.

It is extremely interesting to note that the oil refineries which usually have available large quantities of relatively inexpensive fuel, and which therefore are sometimes not particularly concerned about saving fuel, are turning to the circulating type of wide-range mechanical atomizer for their modern boiler and still installations. Control of heat distribution and furnace conditions and the saving of water for atomizing steam (an item of great importance to those plants so located that extensive feedwater treatment is required) are contributing factors.

Combined burners are being used for firing units of various types. Some of the newer types of oil stills are vertical cylinders as large as 25 ft. in diameter, the walls being formed by vertical tubes containing the fuel to be processed. Some of these are being fired from the bottom and some others from the top. The former method is now more widely employed.

Fig. 7 is a view of a bank of vertical heaters fired from the top with combined gas and oil burners. Fig. 8 is a sectional view of such a furnace fired from the bottom with a single combined still-gas and yard-oil burner of the type previously described. This burner liberates 40,000,000 B.t.u. per hour. When burning gas, two traverses at right-angles in a plane perpendicular to the axis and taken at 1-ft. intervals showed no CO and a CO_2 variation from 10.8 to 12 per cent, indicating the evenness of furnace conditions and the manner in which this single burner distributes the furnace gases. Fig. 9 shows combined gas and oil burners firing a tube still.

In closing, let it be said that the refinery engineers were presented with an enormous problem when they had to dispose of these fuels, and the fact that for years they have been burning them is a testimonial to their ingenuity. They have contributed a vast amount of information, and if this had not been available, modern installations of waste-fuel-burning plants would be far behind their present state of progress. No one can have had any contact with the oil-refining industry in this country without developing a profound respect for the intricate problems involved and the effective manner in which these are being solved.

The fact that refinery engineers have constructed many plants equipped with combined burners for the disposal of their waste fuels is an evidence of their foresightedness.

HYD-53-1

Federal Relations to Water-Power Development

By FRANK E. BONNER,[1] WASHINGTON, D. C.

With the amazing advances made in the production and distribution of electrical energy during the past decade, it is of interest to examine the part played by the Federal Government. This is particularly true of the water-power field where, through vast land holdings and constitutional authority in respect to navigation, the National Government exercises control over a large portion of the country's potential resources. During this period the administration of the law regulating the utilization of power sites involving Government interest has reposed with the Federal Power Commission. Although developments of many of the large rivers have been delayed by legislative embargoes and political controversy, the Commission has authorized the construction of many important projects, and about 2,600,000 horsepower of capacity is now operating under license from the Commission. The proper division of authority between the Government and the states has raised some complex problems which are pressing for solution.



IT IS UNNECESSARY to recount for the members of this Society the unparalleled progress made by the electric industry of this country during the past decade. The mechanical engineer has rendered such important aid in bringing about the astonishing economic and technical advances in supplying our people with cheap and dependable electrical service, that these achievements are already well known to your profession. The preeminent position that has been attained by the United States in the utilization of electrical energy for turning the wheels of industry and lightening the labor burden of the homes and the farms, makes it of some interest to consider what assistance or encouragement the Government may have contributed toward this result.

The primary relation of this industry is with the states. Under the system of regulated public utilities through which most of the electric energy is marketed in this country, the agencies of the state governments have highly important functions in the elimination of destructive competition, while at the same time safeguarding the public from unreasonable rates. The merit of this system has been amply demonstrated by the progressive lowering of the charges to consumers on the one hand, and the growth and stability of the industry on the other. The beneficial effects are especially conspicuous in the existing period of world-wide economic depression, when many basic industries are suffering from the effects of over-expansion and wasteful competition.

It has been recognized that electrical power must of necessity be consumed in the immediate vicinity of its place of production, and the matter of regulation has been viewed as a local problem properly within the authority of the states. With the growth of system interconnection, some problems of an interstate nature are beginning to arise, and proposals for their solution through imposition of Federal activity contemplate a minimum of interference with the existing machinery of the state agencies.

It is evident that the contacts of the Federal Government with the manufacturers and distributers of electrical energy have been confined mostly to those which are common to all industries. One prominent exception is in connection with the development of water power. Here through a combination of two factors the Government finds itself a shareholder in many potential water-power sites. These two factors consist of the Federal responsibility for navigation and the ownership of public land.

U. S. Government the Largest Real-Estate Holder in the Country

The fact is sometimes overlooked that the Government is by far the largest real-estate owner in the country. These holdings are concentrated chiefly in the western states, where vast areas of the original public domain have been reserved for national forests, national parks, Indian reservations, and other purposes. There are also millions of acres comprising the unreserved public domain for which no use, sufficiently profitable, has yet been found to justify private acquisition under the public land laws. Three-quarters of all the country's available water-power resources are found in the eleven most western states and, as might be expected, many of the potential sites involve United States land in greater or less degree. The national forests, which blanket the most of the great mountain ranges, harbor great power resources and have been an important factor in maintaining land useful for water-power purposes in public ownership. Even in these reservations, however, it is rarely found that a development may be made wholly on Government land. Through the operation of the public land laws prior to the establishment of the policies under which water power and other lands were reserved, the Government's holdings have been reduced to scattered and fragmentary tracts which serve to give merely a minority interest in many important developments. The Government, however, with its sovereign powers

---

[1] Executive Secretary, Federal Power Commission. Mem. Am. Soc. C.E. For the past 21 years Mr. Bonner has served continuously in the Government service. Upon completing his engineering training at the University of Montana in 1909, he entered the engineering department of the U. S. Forest Service and with that agency served as district engineer in charge of the engineering activities in several of the western field districts. For a time he had charge of the engineering work at the Research Laboratory at Madison, Wis., and later served several years as assistant chief engineer of the Forest Service at Washington, D. C., in the direction of water-power administration, highway construction, and other activities. In 1922 he was assigned as district engineer of the California District and in charge of the Federal Power Commission's activities of that region. He served in that capacity until transferred to his present position on July 1, 1929.

Contributed by the Hydraulic Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

may not be disposed of through the ordinary process of condemnation like other small land holders, and therefore, even though its stake in the ownership of numerous sites is small, it is in a position to dictate the terms upon which the entire development may be allowed to proceed. In the exercise of this unlimited authority the Government becomes an important factor relative to many western projects which otherwise would be controlled solely by the state concerned.

Federal Jurisdiction Over Navigable Streams

The interests of navigation which supply the basis for the second element of Federal jurisdiction over certain water-power projects are derived from the fundamental law. The Constitution makes no reference to water power or navigation as such, but Section 8 of Article 1 embraces the following phrase:

> The Congress shall have power . . . to regulate commerce with foreign nations, and among the several States, and with the Indian Tribes.

By these few words the states surrendered to the central Government vast power under which many forms of Federal activity have grown up, including the control of the waterways used or useful for the transportation of persons or property in interstate commerce. The definition of navigable waters subject to the control of Congress has been the subject of much controversy. A long line of court decisions has failed to lay down a precise rule by which the jurisdictions of the state and national governments may be segregated with assurance. Whether a stream is navigable or not seems to be a mixed question of law and fact. On this point one of the clearest statements of the Supreme Court is that found in the decision in the case of United States vs. Holt State Bank (270 U. S. 49, 56) which said:

> The rule long since approved by this Court in applying the Constitution and Laws of the United States is that streams or lakes which are navigable in fact must be regarded as navigable in law; that they are navigable in fact when they are used, or are susceptible of being used, in their natural and ordinary condition, as highways for commerce, over which trade and travel are or may be conducted in the customary modes of trade and travel on water; and further that navigability does not depend on the particular mode in which such use is or may be had—whether by steamboats, sailing vessels, or flatboats—nor on an absence of occasional difficulties in navigation, but on the fact, if it be a fact, that the stream in its natural and ordinary condition affords a channel for useful commerce.

While uncertainty will arise relative to the navigable status of individual streams, it is evident that many of the country's waterways are subject to Federal jurisdiction, and by the act of March 3, 1899, Congress decreed that "any obstruction not affirmatively authorized by Congress to the navigable capacity of any of the waters of the United States is hereby prohibited." This law has been construed to apply not only to all structures placed in waters actually navigable, but also to such obstructions on non-navigable tributaries which have the effect of substantially impairing the use of "navigable waters" lower down on the stream.

From the foregoing, it is apparent that even though the separate states possess the right to control the use and development of waters within their borders, the National Government with its land ownerships and responsibility to navigation holds a basis for participation in the authorization of many water-power projects. Under existing law the administration of these interests is vested with the Federal Power Commission created in 1920. Prior to 1920 authorization for projects encroaching upon Government land or in navigable waters was obtained in various forms and from various agencies. Quite a number were sanctioned by direct grants from Congress. Many others were authorized under general legislation through permits issued by whatever Government agency happened to have jurisdiction over the property involved. These permits were of a revocable nature, and the difficulty of financing the cost of large works on leaseholds subject to bureaucratic vagaries led to a demand for legislation which would serve to promote the development of water resources.

Purpose of Federal Water-Power Act

After many years of discussion the Federal water-power act was passed in 1920. It aimed to provide a uniform policy under which all of the permissive actions of the Government might be managed. It authorized the granting of licenses which are in the nature of limited leases for 50 years. Development by private capital or public agencies was encouraged to the fullest extent practicable by limiting hampering restrictions and administrative fees to a minimum. Moreover the licensee was assured a stable tenure by the guarantee of reimbursement for the cost or fair value of improvements in case the lease should not be renewed upon its termination. The law thoroughly recognizes the principle that the exercise of Federal jurisdiction should be subordinated to that performed by the states in their rightful fields.

In preference to setting up an entirely independent agency for carrying out the functions of the law, Congress chose to have the work continued by the technical staffs of the departments. The necessary correlation of effort was accomplished by composing the Commission of the heads of the three departments concerned and providing it with a small headquarters staff. Pursuant to recommendation by the President, the last Congress authorized the substitution of full-time commissioners in order that the cabinet-officer members might be permitted to give undivided attention to the management of their departments.

The law has now had ten years of trial, and it seems appropriate that inquiry be made about what has been accomplished and whether it is serving to encourage water-power development as intended. As mentioned heretofore, this decade has been characterized by enormous growth and expansion in the electric industry. In this period the consumption of electrical energy has increased 137 per cent, while the population has grown only 16 per cent. Today we are using more than twice as much power per capita as we were ten years ago. The rapidly expanding demand has created the need for huge new sources of production, and many noteworthy water-power plants have been constructed.

Activities of Federal Power Commission

A substantial proportion of the new construction has been carried forward under the provisions of the Federal water-power act. Altogether the Commission has granted licenses for 449 projects, but only 107 of these are in the so-called major classification which embraces the developments larger than 100 hp. The rest relate to minor projects and segments of transmission lines crossing Government land. The geographical distribution of the licensed projects shows great concentration in the western states. California alone accounts for more than a third of the 449 projects, while the three Pacific Coast states together have just about half of the total number. Less than 10 per cent of the projects authorized are situated east of the Mississippi River. In most of the western cases the Government's jurisdiction arises from land ownership, while the limited number in the East are due almost entirely to navigation interest. Although few in number, the projects related to navigation are generally of large size.

The licenses of chief importance are those covering the 107 major projects. These range in magnitude from small plants of several hundred horsepower constructed to serve isolated

mining developments or small communities, to great public-utility developments of several hundred thousand horsepower. The giant generating stations at Niagara, at Conowingo, and on the Sierra Nevada streams of California furnish conspicuous examples of the large modern stations carried out under the existing Federal legislation. Obviously, many of the proposed projects involve questions of a highly controversial nature, but in handling applications the Commission has moved with such diligence and dispatch that no complaint has been made that construction has been unduly delayed on any legitimate project or that essential interests of the public have not been adequately safeguarded. During the past year the Commission has granted licenses which authorize the installation of more than 1,000,000 hp. It is anticipated that the immediate construction of these projects will contribute measurably toward the improvement of employment conditions and the stimulation of industrial development in the localities affected. During the same period, 30 applications were rejected pursuant to the Commission's endeavors to prevent favorable sites from falling into the hands of speculative enterprises which are unprepared to proceed with construction.

Aggregate Capacity of Water-Power Plants Under License

The aggregate capacity of plants under license which have been completed and placed in operation now amounts to about 2,600,000 hp. This comprises approximately 25 per cent of all the water-power capacity, and slightly less than 6 per cent of the total generating capacity in public-utility service. The Commission has no jurisdiction in respect to water-power developments other than the projects under license, and thus it is apparent that the scope of its authority in respect to the electrical industry as a whole is comparatively limited. Contrary to popular impression, the Commission has almost no jurisdiction in the field of rate regulation. The law specifically limits the exercise of such functions by the Commission to those licensed projects not controlled by state regulation, and since practically all the states embracing licensed projects are supervising public-utility rates, there has been little room for the Commission to engage in this business.

Since the authority of the Commission is by no means exclusive in respect to the licensed projects, the administration of the law has required close correlation with the activities of the state agencies. In general, these relations have been entirely harmonious and mutually helpful, but at times there has been complaint that the Commission was stretching its authority unduly. These complaints have related mostly to jurisdiction over projects located entirely on private land along non-navigable tributaries, where the Commission has assumed that the operation of the project works might serve to impair the use of navigable waterways lower down on the stream. The Commission's powers over such developments are rather vague and indefinite under the law, and a number of the states have challenged the Government's constitutional authority. They assert that the assumption of jurisdiction in such cases constitutes a serious invasion of the rights of the states, and interferes unnecessarily with developments of strictly local concern. For guidance in the matter the Commission has recently obtained an interpretation of the law from the Attorney General of the United States. This opinion holds that the Commission, in such cases, may grant licenses embodying terms restricted to those necessary for the protection of navigation. The states do not contest or object to reasonable exercise of the Government's functions in safeguarding navigation, and water-power licenses designed for this specific purpose will probably be acceptable to them. By thus harmonizing the respective jurisdictions of the State and Federal governments so that each may be effective in its proper field, a threatening shadow on the operation of the law will be removed.

With the extensive construction of new hydroelectric projects recently, it is somewhat surprising to find that the proportion of the energy requirements now generated from water power is practically the same as it was ten years ago. We are still producing only slightly more than a third of the country's public-utility needs from water power, and manufacturing the rest in fuel-burning stations. When the factors bearing on the situation are examined, some doubt is occasioned as to whether water power will be able to maintain as large a share of the increasing business of the future. We are in a period of great shifting of fundamental economic conditions. The tremendous production of cheap fuel supplies, the low cost of their distribution, and the remarkable improvements in machinery used for their transformation into electrical energy, have largely dispelled the popular notion about the advantages of water power. Even in the regions of abundant water-power resources there is a marked increase in the dependence placed on fuel-generated energy. Other factors which are restricting wider use of the water-power resources include the high cost of transmission to distant load centers, the deficient water supplies due to severe and repeated droughts, the increase of obstructive litigation, the tendency to make water-power developments the object of political controversies, and the imposition of special tax burdens on this particular form of power generation. It is evident that under present conditions the development of water power is faced with the keenest sort of competition by steam plants, and notwithstanding the encouragement extended by the Federal law the undertaking of new projects will depend largely upon a full measure of public cooperation.

Discussion

Guido H. Marx.[2] To the student of the conservation movement in our country familiar with the issues involved in the long campaign which culminated in Public Act No. 280 of the 66th Congress, known as the Water Power Act, this paper—emanating from the Executive Secretary of the Federal Power Commission, the executive officer primarily concerned with seeing that the act's provisions protecting the rights and interests of the general public are effectively exercised—is more than a disquieting one; it is one which should stir him to pro est and action.

To discuss in detail the departure of its philosophy from that of the act which the Power Commission is supposed to enforce would require space equal to that occupied by the paper itself. It is desirable, therefore, to limit th's discussion to two points only of the many which should be taken up.

(1) The author says, "It has been recognized that electrical power must of necessity be consumed in the immediate vicinity of its place of production," and upon this he rests his argument for state regulation with a "minimum interference" by Federal authority. To engineers possessing even the most elementary knowledge of present-day electrical transmission of power the statement is known to be the direct antithesis of the actual fact, and the argument based upon it is as invalid as the foundation is worthless. A large part of the admitted failure of state regulation of public utilities can be laid directly at the door of lack of coincident jurisdiction.

In a pamphlet issued by the Guarantee Trust Company of New York (surely not a Bolshevistic source) under date of Sep-

[2] Professor of Machine Design, Stanford University, P. O. Stanford University, Calif. Mem. A.S.M.E.

tember 15, 1920, entitled "Our Vast Unused Energy," and containing a reprint of the act in question, there is this interpretation of the regulatory powers conferred upon the commission:

> Public service enterprises are subjected to control of service and rates by state commissions where such exist, and where such a state commission is not in existence, the Federal Power Commission is authorized to regulate and control service, rates, and the issuance of securities. When the power or any part of it enters into interstate or foreign commerce, the rates of service must be reasonable, just, and non-discriminating. Briefly, this involves a degree of control in the field of power development and distribution between states comparable to that exercised over railways by the Interstate Commerce Commission.

In the present writer's mind the query arises whether the attitude of relinquishment shown in the paper by this executive officer does not amount to nonfeasance.

One minor question in passing; one wonders what is meant by the phrase "to prevent favorable sites from falling into the hands of speculative enterprises which are unprepared to proceed with construction." Is this the method employed by the Commission to encourage that sturdy individualism we have been hearing so much about? Among the unenlightened there might be some who would construe the words to imply that those not affiliated with the great pecuniary interests, popularly known as the "power trust," may more hopefully expend their efforts of initiative in some other line of "private enterprise."

(2) The discussion of the relative economic merits of water or fuel power which closes the paper, with an attempt to minimize the importance of the water-power issue, can scarcely mislead a group of engineers who are accustomed to analyze the differences between permanent real values and temporary pecuniary values. It is little less than tragic that a Federal administration from whom engineers, in general, had hoped so much in achievement and prestige should have no more dependable sources to inform them of the temper of the people, not to mention their will as expressed in their executed law. Some of the consequences can be seen in the results of the recent elections as shown in New York, Pennsylvania, Tennessee, Nebraska, and elsewhere.

NOTE: Following the presentation of Mr. Bonner's paper and the reading of the letter from Professor Marx, an oral discussion developed. The consensus of opinion expressed was strongly in support of the principles advocated in Mr. Bonner's paper. It was deemed desirable that the impartial and unbiased viewpoint of the engineering profession on these questions be made available to governmental bureaus desiring it, and to the public. This led to the preparation of the following resolution.

> *Resolved*, That the Hydraulic Division recommend to Council that it sponsor the formation of a committee to be known as the Engineers Water-Power Policy Committee, and that the committee be composed of five members, two of which are to be appointed by the American Engineering Council, one by The American Society of Mechanical Engineers, one by the American Institute of Electrical Engineers, and one by the American Society of Civil Engineers.

[The resolution was presented to the Council by Ely C. Hutchinson, and the Council voted to transmit it to the American Engineering Council with the suggestion that a committee corresponding to the one mentioned in the resolution be set up in the American Engineering Council to deal with this matter. Similar action was taken by the American Society of Civil Engineers and the American Institute of Electrical Engineers.—EDITOR.]

Calibration of a Large Nozzle

By JOHN R. DuPRIEST,[1] MINNEAPOLIS, MINN., AND JAMES H. POLHEMUS,[2] PORTLAND, ORE.

In testing large centrifugal pumps it is difficult to determine the amount of water that is being handled. In many cases the quantity is so great it is almost impossible to determine it by the ordinary field weighing system. The use of standard test nozzles offers a simple means of determining the capacity of large pumps, but the nozzles must be accurately calibrated. The authors made a test of three pumps by laying about 1000 ft. of pipe behind a dredge and attached standard nozzles to the end of the pipe line for calibration purposes. The quantity of water flowing was determined by the salt-solution method.

IN the testing of large centrifugal pumps one of the difficult tasks is to determine the amount of water being handled by the pump. In many cases the quantity is so great that it is practically impossible to determine it by any ordinary field weighing system.



J. R. DuPriest

There are several methods used to determine the capacity of large pumps, and each method has advantages and disadvantages, depending on the conditions under which the tests must be made. Standard test nozzles offer a very simple means of determining the capacity of large pumps, but the nozzles must be very accurately calibrated in order to insure trustworthy results.

Fig. 1 shows dimensions of a set of nozzles which have been used for many years for the testing of large hydraulic dredge pumps and which are considered standard for such work.

Some time ago the authors were engaged in research work to collect data on large dredge pumps preparatory to building the large Diesel-electric dredge *Clackamas* for the Port of Portland. When this work was under consideration the question arose as to the methods to be used for testing the pumps, and naturally the nozzle method was suggested on account of its simplicity.

Since the work to be done was of great importance, it was highly desirable that the results be as accurate as it was possible to obtain from field tests; therefore every phase of the work was very carefully studied.



J. H. Polhemus

One of the first questions therefore to be considered was the accuracy of the nozzle method for such tests.

Investigation showed that a coefficient of discharge of 0.94 had been established for this set of nozzles for all operating conditions and had been applied to many tests over a long period of years. No one seemed to know just when this coefficient was established or by whom.

The authors were not just satisfied as to the accuracy of this coefficient and decided to make an effort to recalibrate the set of nozzles. From the design of the nozzles one would hardly expect a very high coefficient, and also it did

TABLE 1 NOZZLE COEFFICIENT DATA OF DREDGE "TUALATIN," TEST NO. 1

Pump speed, r.p.m. (1)	Pipe-line velocity, ft. per sec. (2)	Nozzle pressure, lb. per sq. in. at gage (3)	Static head on nozzle, ft. at c.l. (4)	Velocity head at base of nozzle, ft. (5)	Total head on nozzle, ft. at c.l. (6)	Discharge coefficient C (7)
		Nozzle Diameter, 8 In.				
185	3.65	20.5	48.65	0.21	48.86	0.892
215	4.29	28.1	66.15	0.29	66.44	0.862
245	4.93	37.5	87.85	0.38	88.23	0.857
275	5.56	48.0	112.25	0.48	112.73	0.857
314	6.39	63.0	147.05	0.63	147.68	0.862
		Nozzle Diameter, 10 In.				
190	5.59	19.3	45.85	0.49	46.34	0.864
215	6.41	25.3	59.75	0.64	60.39	0.865
245	7.39	34.0	79.75	0.85	80.60	0.863
275	8.38	43.8	102.45	1.09	103.54	0.863
306	9.37	55.0	128.35	1.36	129.71	0.863
		Nozzle Diameter, 12 In.				
195	7.90	18.0	42.85	0.97	43.82	0.867
215	8.75	22.3	52.75	1.19	53.94	0.866
245	10.0	30.0	70.55	1.55	72.10	0.857
275	11.3	38.6	90.35	2.03	92.38	0.852
298	12.2	45.8	107.25	2.39	109.64	0.850
		Nozzle Diameter, 14 In.				
200	9.89	16.0	38.25	1.51	39.76	0.839
220	10.9	20.0	47.45	1.85	49.30	0.830
240	11.9	24.1	56.85	2.20	59.05	0.826
260	12.9	28.6	67.35	2.58	69.93	0.825
288	14.2	35.4	83.05	3.14	86.19	0.818
		Nozzle Diameter, 16 In.				
205	12.3	12.9	31.05	2.37	33.42	0.875
225	13.5	16.0	38.25	2.84	41.09	0.862
245	14.7	19.6	46.45	3.36	49.81	0.852
260	15.6	22.5	53.25	3.78	57.03	0.843
276	16.5	25.5	60.15	4.24	64.39	0.842
		Nozzle Diameter, 20 In.				
160	11.8	3.0	8.19	2.09	10.28	0.945
185	13.3	4.6	11.90	2.74	14.64	0.905
210	14.9	6.6	16.93	3.45	20.37	0.864
235	16.5	8.9	21.85	4.25	26.10	0.846
264	18.5	11.0	26.65	5.32	31.97	0.856

[1] Professor of Mechanical Engineering, University of Minnesota. Mem. A.S.M.E. Mr. DuPriest was graduated in 1901 from Virginia Polytechnic Institute with B.S. in E.E. In 1901–1905, in charge of mathematics and shop work in high schools at Marion, S. C., and at Clearfield, Pa.; 1905–1906, drafting and design with General Electric Company, Ingersoll-Rand Company; 1906–1909, designer on steam and gas engines with the C. & G. Cooper Co., Mt. Vernon, Ohio; 1909–1911, chief engineer, Columbus Machine Company, Columbus, Ohio; 1911–1920, instructor at Cornell University and at University of Wisconsin, and professor at University of Idaho and Rensselaer Polytechnic Institute; 1920–1921, consulting engineer, Ellicott Machine Corporation, Baltimore, Md., and was sent abroad to study designs of special machinery; 1921–1927, consulting engineer for the Port of Portland, designing the main pump on the new Diesel-electric dredge *Clackamas;* 1927 to date, head of Department of Mechanical Engineering, University of Minnesota.

[2] General Manager and Chief Engineer, Port of Portland. Mr. Polhemus was graduated from Leland Stanford, Jr., University in 1910. He became connected with the Port of Portland in 1920 after having been the general manager and chief engineer of the Coos Bay Ship Building Company from 1917 to 1920, during which time this company built twelve ships for the United States Shipping Board. Prior to 1927 he served as an assistant engineer for the U. S. Engineer Department for some six years. During the last three years he was in charge of the river and harbor improvement work in southern Oregon, an interesting work in that district at that time being the dredging for the Coos Bay ocean bar with a seagoing dredge. This was the first successful bar dredging of this nature on the rough exposed North Pacific coast harbors. In his present position he has designed, built, and operated some of the largest and most modern hydraulic pipe-line dredges in this country.

Contributed by the Hydraulic Division and presented at the Semi-Annual Meeting, Birmingham, Ala., April 20 to 23, 1931, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

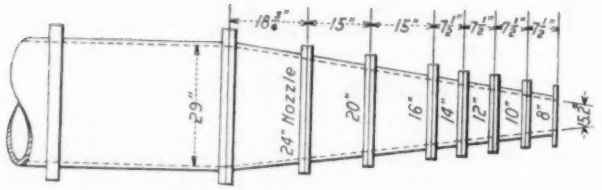


FIG. 1 PORT OF PORTLAND TEST NOZZLE

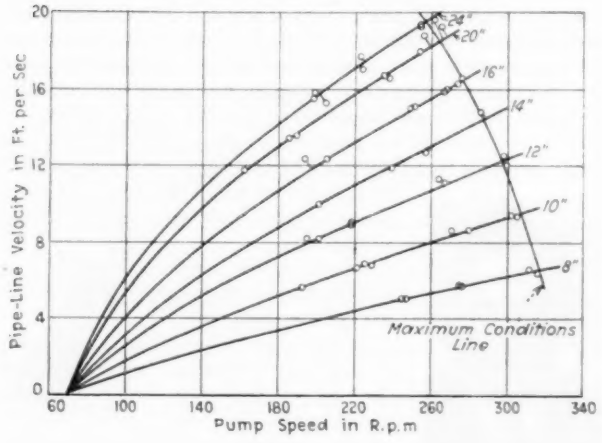


Fig. 2

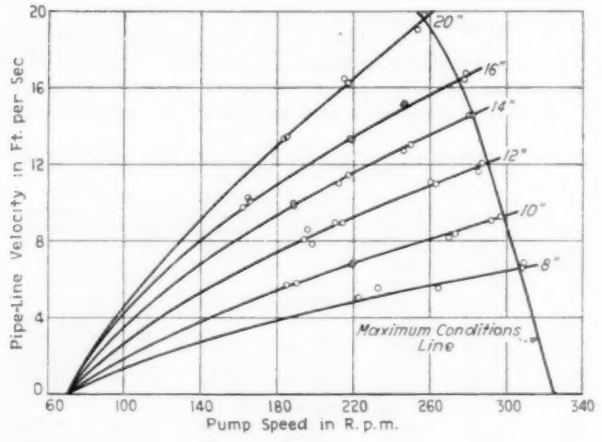


Fig. 3

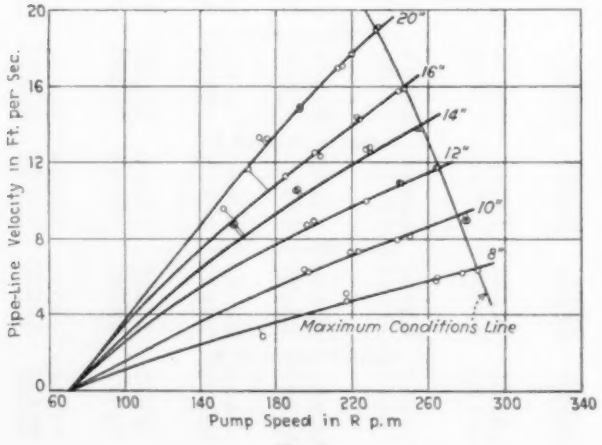


Fig. 4

TABLE 2 NOZZLE COEFFICIENT DATA OF DREDGE "TUALATIN," TEST NO. 2

Pump speed, r.p.m. (1)	Pipe-line velocity, ft. per sec. (2)	Nozzle pressure, lb. per sq. in. at gage (3)	Static head on nozzle, ft. at c.l. (4)	Velocity head at base of nozzle, ft. (5)	Total head on nozzle, ft. at c.l. (6)	Discharge coefficient C (7)
			Nozzle Diameter, 8 In.			
155	3.1	13.1	31.45	0.15	31.60	0.900
195	4.2	24.1	56.75	0.27	57.02	0.909
235	5.1	35.7	83.55	0.40	83.95	0.904
275	5.9	48.9	114.05	0.54	114.59	0.902
308	6.5	63.2	147.15	0.66	147.81	0.875
			Nozzle Diameter, 10 In.			
185	5.5	19.3	45.75	0.47	46.22	0.850
215	6.6	27.0	60.15	0.68	60.83	0.888
245	7.6	35.4	82.75	0.90	83.65	0.871
275	8.5	44.9	104.55	1.12	105.67	0.866
299	9.2	54.2	126.25	1.32	127.57	0.854
			Nozzle Diameter, 12 In.			
195	8.1	18.5	43.85	1.02	44.87	0.880
220	9.2	24.5	57.75	1.31	59.06	0.872
245	10.3	30.8	72.25	1.64	73.89	0.862
270	11.3	37.9	87.30	1.96	89.27	0.866
289	12.0	44.0	102.55	2.24	104.79	0.851
			Nozzle Diameter, 14 In.			
190	9.8	14.5	35.65	1.49	37.14	0.851
215	11.1	19.4	45.85	1.90	47.75	0.855
240	12.3	24.5	57.65	2.36	60.01	0.850
265	13.6	30.2	70.75	2.87	73.62	0.847
281	14.4	34.3	80.25	3.24	83.49	0.844
			Nozzle Diameter, 16 In.			
165	10.1	8.0	19.66	1.57	21.23	0.893
190	11.6	11.5	27.75	2.07	29.82	0.866
215	13.0	15.3	36.55	2.64	39.19	0.850
245	14.8	20.1	47.55	3.40	50.95	0.848
274	16.5	25.4	59.75	4.25	64.00	0.846
			Nozzle Diameter, 20 In.			
185	13.5	4.0	10.46	2.81	13.27	0.964
205	15.2	5.9	14.85	3.59	18.44	0.924
225	16.9	8.0	19.68	4.43	24.11	0.900
240	18.2	9.6	23.37	5.15	28.52	0.890
256	19.6	11.6	27.97	5.96	33.93	0.880

TABLE 3 NOZZLE COEFFICIENT DATA OF DREDGE "TUALATIN," TEST NO. 3

Pump speed, r.p.m. (1)	Pipe-line velocity, ft. per sec. (2)	Nozzle pressure, lb. per sq. in. at gage (3)	Static head on nozzle, ft. at c.l. (4)	Velocity head at base of nozzle, ft. (5)	Total head on nozzle, ft. at c.l. (6)	Discharge coefficient C (7)
			Nozzle Diameter, 8 In.			
215	4.6	32.0	75.05	0.33	75.38	0.870
235	5.1	39.1	91.45	0.41	91.86	0.871
250	5.5	45.0	105.05	0.46	105.51	0.870
265	5.8	51.2	119.25	0.52	119.77	0.870
286	6.3	61.0	141.95	0.62	142.57	0.865
			Nozzle Diameter, 10 In.			
195	6.2	23.0	54.25	0.60	54.85	0.882
215	6.9	28.2	66.25	0.75	67.00	0.889
235	7.6	34.2	80.15	0.89	81.04	0.884
255	8.3	41.0	95.75	1.06	96.81	0.880
277	9.0	49.2	114.75	1.26	116.01	0.875
			Nozzle Diameter, 12 In.			
200	8.8	21.5	50.85	1.21	52.06	0.892
220	9.7	26.5	62.35	1.46	63.81	0.885
235	10.3	30.4	71.35	1.66	73.01	0.877
250	11.1	34.8	81.45	1.90	83.35	0.880
267	11.8	40.0	93.55	2.16	95.71	0.876
			Nozzle Diameter, 14 In.			
155	7.6	11.5	27.75	0.90	28.65	0.759
180	9.4	15.0	35.85	1.37	37.22	0.825
205	11.0	19.6	46.45	1.88	48.33	0.847
230	12.5	25.3	59.65	2.43	62.08	0.848
257	14.1	32.0	75.05	3.08	77.33	0.856
			Nozzle Diameter, 16 In.			
165	10.3	9.4	22.95	1.65	24.60	0.850
185	11.6	11.7	28.25	2.10	30.35	0.862
205	13.0	14.8	35.35	2.62	37.97	0.862
225	14.4	18.5	43.85	3.12	26.97	0.857
248	16.0	23.1	54.55	3.97	58.52	0.856
			Nozzle Diameter, 20 In.			
175	13.1	4.2	10.95	2.67	13.62	0.925
190	14.7	5.3	13.45	3.36	16.81	0.935
205	16.2	6.8	16.95	4.06	21.01	0.924
220	17.7	8.5	20.85	4.85	25.70	0.911
234	19.2	10.3	25.05	5.71	30.76	0.905

FIGS. 2, 3, AND 4 (LEFT) PIPE-LINE-VELOCITY CURVES FOR TESTS OF THREE PUMPS
(Nozzle 1089 ft. behind dredge.)

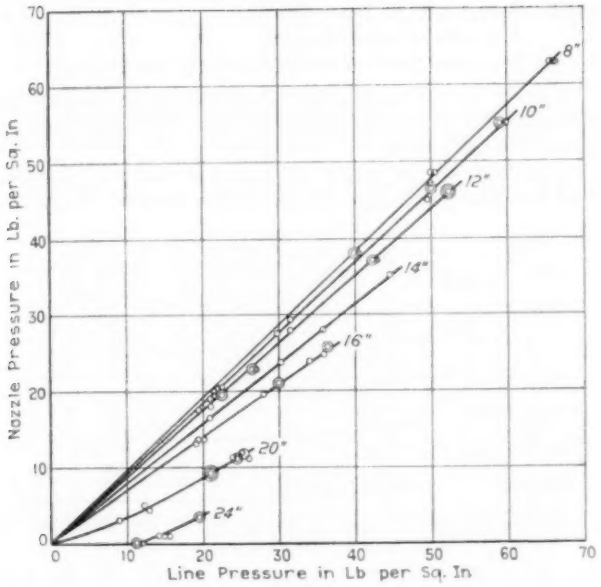


Fig. 5

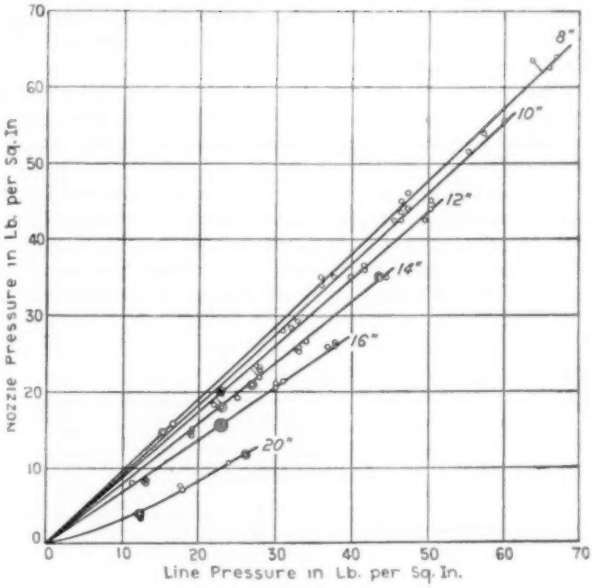


Fig. 6

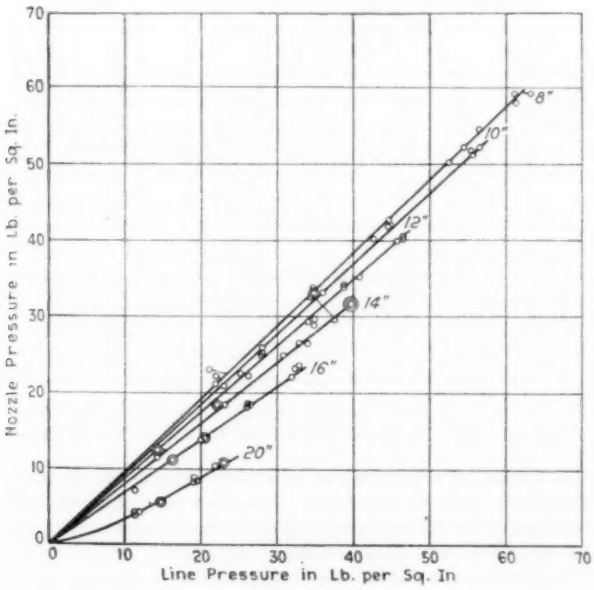


Fig. 7

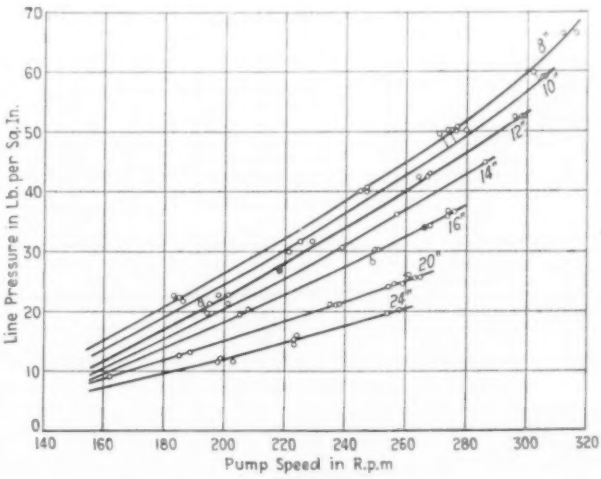


Fig. 8

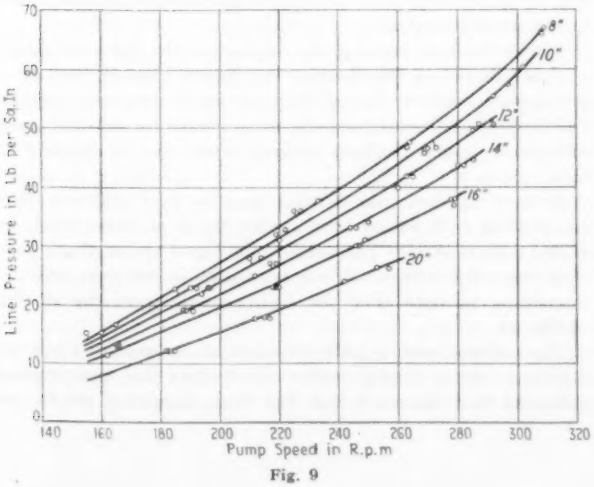


Fig. 9

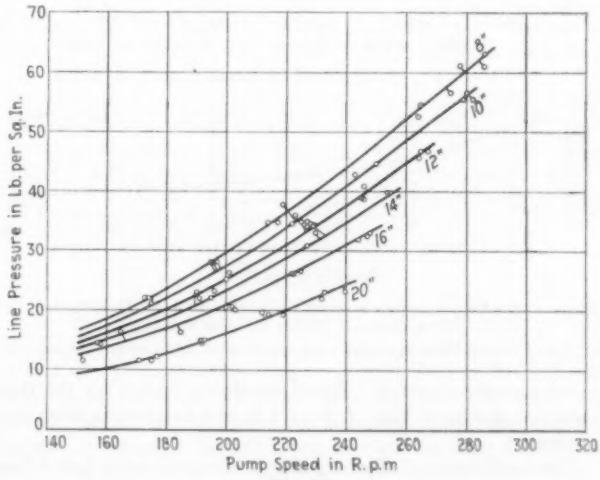


Fig. 10

FIGS. 5 TO 10 TOTAL HEAD ON THE NOZZLE TAKEN FROM PRESSURES READ JUST AHEAD OF THE NOZZLE AND LINE PRESSURES JUST BEHIND THE DREDGE
(Nozzle 1089 ft. behind dredge.)

not seem reasonable that the same coefficient would hold for all nozzles of the set during wide variations in pressure and while L/D was changing so much.

Finally it was decided to lay about 1000 ft. of pipe behind the dredge and attach the nozzles to the end of the pipe line for calibration purposes. This procedure was used during complete tests of three pumps, and the results are given in this paper.

The quantity of water flowing was found by determining the pipe-line velocity by the salt-solution method, and it is believed that the results found in this way are as accurate as could be obtained under the circumstances. Data from these tests plot up in families of curves, which enables errors of observation to

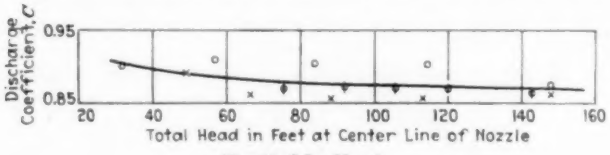


Fig. 11, 8-In. Nozzle

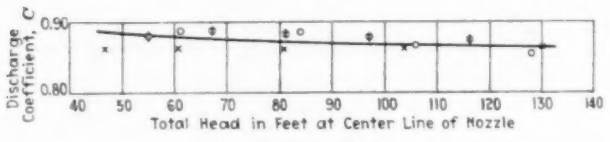


Fig. 12, 10-In. Nozzle

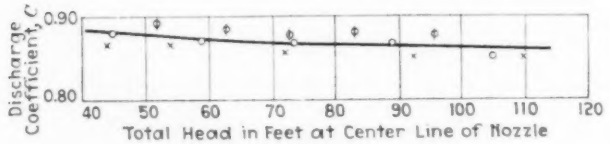


Fig. 13, 12-In. Nozzle

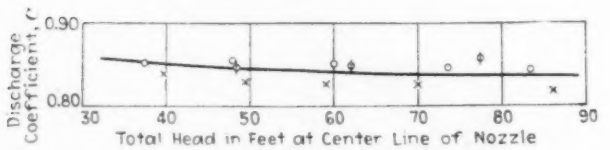


Fig. 14, 14-In. Nozzle

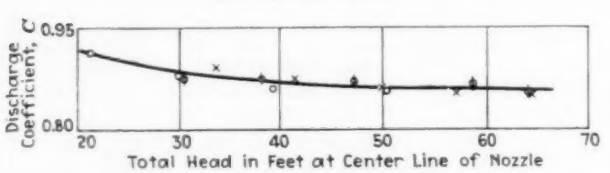


Fig. 15, 16-In. Nozzle

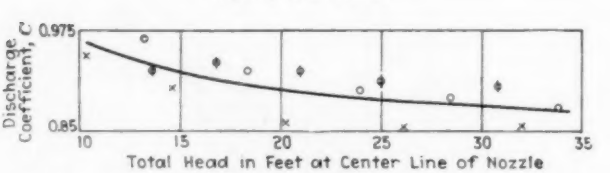


Fig. 16, 20-In. Nozzle

FIGS. 11 TO 16 CURVES FOR THE COEFFICIENT OF DISCHARGE AS A FUNCTION OF HEAD ON THE NOZZLE

(For a 15-deg. conical nozzle attached to pipe 29 in. inside diameter.)

be very easily detected. Pipe-line-velocity curves for the three tests are shown in Figs. 2, 3, and 4, and the results seem consistent.

The coefficients of discharge for the nozzles were found from the equation

$$Q = CA\sqrt{2gh}$$

where Q = cu. ft. of water per sec.
A = area of nozzle in sq. ft.
h = total head on nozzle at center line in ft.

Tables 1, 2, and 3 give observed and computed data for calculating values for C, and these data were obtained from curves shown in Figs. 2 to 10.

Total head on the nozzle was found as follows: Pressures were read just ahead of the nozzle, and line pressures just behind

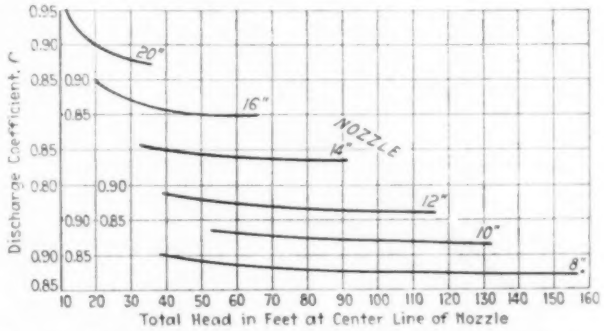


FIG. 17 NOZZLE COEFFICIENT CURVES DRAWN TO THE SAME SCALES

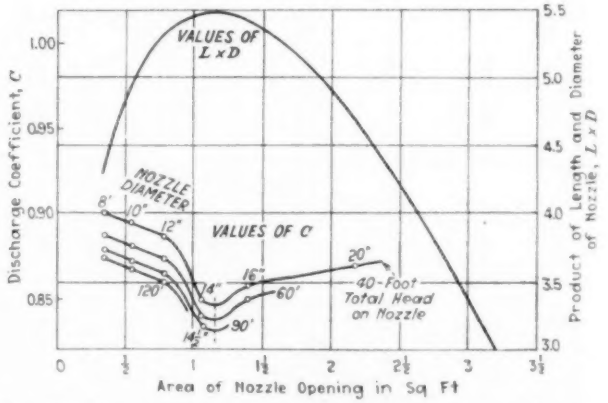


FIG. 18 COEFFICIENTS OF DISCHARGE PLOTTED AGAINST AREA OF NOZZLES FOR CONSTANT DISCHARGE HEADS ON THE NOZZLES

the dredge. These pressures were plotted up as shown in Figs. 5 to 10 to smooth out errors. Pressures were then read from the curves for chosen speeds, and to these values were added velocity-head and gage-elevation corrections.

Curves for the coefficients of discharge as a function of head on the nozzle for each nozzle are given in Figs. 11 to 16, and Fig. 17 shows curves for all nozzles plotted to the same scales, but displaced vertically.

The method of drawing the curves for the different nozzles was as follows: A pencil curve was drawn through each set of points for a nozzle as though only one set of data was available; values were then read from the three curves for selected heads, and the averages of these readings were used in plotting the final curves.

As was expected, the different nozzles gave different coefficients since L/D varied over a wide range and also the heads varied a great deal. The coefficients found are all much lower than the value 0.94 which has been used in the past, and it is interesting to note that the 14-in. nozzle gave the smallest coefficient.

The authors were a little surprised at the results when first obtained, but a careful review of all data and computations indicated that the work was free from numerical errors, and

further study of the question made the results seem more reasonable.

A nozzle placed on the end of a pipe will act in quite a different way from a nozzle attached to the side of a tank. If the length of a nozzle attached to a tank approaches zero, the result is an orifice in a thin plate, and the coefficient will be near 0.6, whereas with the set of nozzles attached to a pipe, of the same diameter as the largest nozzle connection, as the nozzle is shortened the

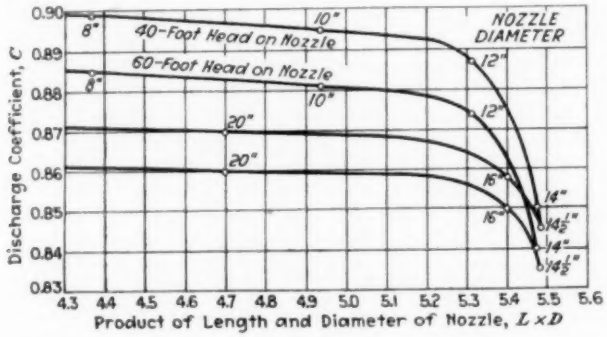


FIG. 19 DISCHARGE COEFFICIENT, C VERSUS $L \times D$
(For a 15-in. conical nozzle attached to a pipe 29 in. inside diameter.)

final result will be an open-end pipe with a coefficient approaching unity.

Inspection of Fig. 1 shows that the design of the set of nozzles is such that each nozzle is a frustum of a cone and that the exit side of each nozzle has a very sharp edge which would be expected to give a low coefficient of discharge. The nozzles cannot be changed to improve the bad exit conditions if they are to be assembled to form a set as in Fig. 1.

After finding by experiment that the lowest coefficient of discharge was for the 14-in. nozzle, further study was undertaken to determine the reason for this result and to find out if possible just where the lowest value might fall. Curves were plotted, as in Fig. 18, showing coefficients of discharge against area of nozzles for constant discharge heads on the nozzles, and it appeared that the minimum value for the discharge coefficient would occur for a nozzle slightly larger than 14 in.

With constant head on the nozzles, the only variables are L and D; therefore some relation was sought between these dimensions which might have a critical value when the coefficient was a minimum. After studies were made along several different lines, it was found that when the product of $L \times D$ was plotted against the area of the nozzle, the curve shown in Fig. 18 was obtained, which appeared to have a critical value just beyond the 14-in. nozzle point.

This result suggested an analysis by the calculus, and it was then found that the maximum value for $L \times D$, regardless of the nozzle angle, always occurred at a point where the diameter of the nozzle was one-half the diameter at the largest end.

Now since $L \times D$ was found to have a critical value and also the coefficients of discharge showed critical values, curves were plotted between these two sets of data, giving the very interesting results shown in Fig. 19.

The authors have no explanation just now to offer for these apparently unusual results, but it is believed, if sufficient data were available and all factors affecting loss through a nozzle were taken into consideration, that a satisfactory analysis of this problem could be made.

As mentioned, the coefficients found for this set of nozzles are much lower than the value 0.94 which has been used in the past; however, the data were very carefully observed, and in working up the results care was taken to smooth out errors of observation, and also numerical work was thoroughly checked. It is believed, therefore, that the results given are substantially correct for the conditions under which the nozzles were calibrated.

It should be noted here that in making comparisons with dredge-performance data, which have been obtained by using a value of $C = 0.94$, capacities and efficiencies will be too high for all such tests.

Another point of interest in this connection is that it is commonly stated that any nozzle is equivalent to a definite length of pipe line under all conditions of operation.

The curves shown in Fig. 20 give pipe-line lengths against static pressures for various size nozzles, and it was found that

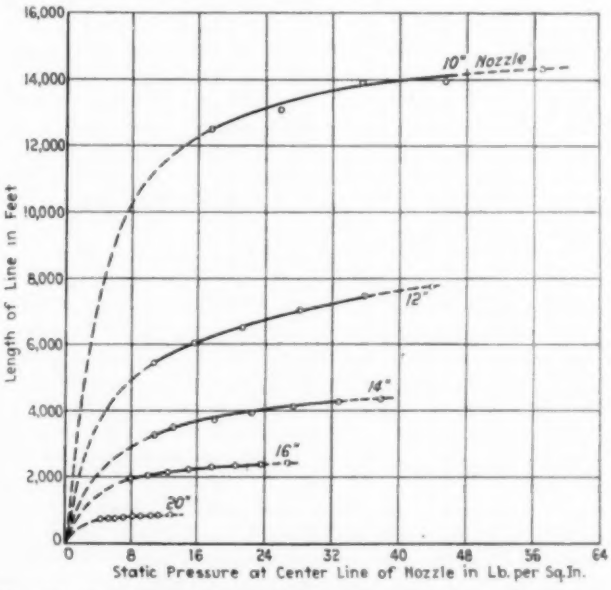


FIG. 20 PIPE-LINE LENGTHS PLOTTED AGAINST STATIC PRESSURES FOR VARIOUS SIZE NOZZLES

the length equivalent to a given size nozzle increased as the static pressure at the center line of the nozzle increased.

The authors realize that the nozzle discussed in this paper is of such a large size that very few engineers will have occasion to make use of the data and results presented. It is hoped, however, that this discussion will create some interest in the general subject of nozzles for measuring the flow of water and perhaps lead other engineers to follow up this work on perhaps smaller nozzles of similar design with the result that something of real value may soon be added to the meager literature on this subject.

The nozzle offers a very simple means of measuring the flow of water, and it would seem that it would be well worth while to develop a few standard sizes of the type discussed in this paper which could be calibrated and which would then serve as an accurate and rapid method of determining the flow of water over very wide ranges of operating conditions.

Discussion

SANFORD A. MOSS.[3] The authors are to be congratulated on the great gain in accuracy that they have secured by use of an actually determined coefficient of discharge, instead of the single uncertain value of 0.94 previously in use. No doubt the existing conical nozzles gave the best solution of the authors' problem. It seems, however, that a better method is possible, which will give still more accurate results, and that any future

[3] Mechanical Engineer, Thomson Research Laboratory, General Electric Co., West Lynn, Mass. Mem. A.S.M.E.

problem of this type will be better solved by the more accurate method.

This method is the use of a nozzle with well-rounded approach, and calibration by means of an impact tube traverse in the jet discharged from the nozzle. As an actual fact, enough data exist with respect to nozzles with well-rounded approach to give results accurate to less than 1 per cent, without any calibration whatever. Such nozzles with well-rounded approach have discharge coefficients which are very close to 0.99 for the conditions of the authors' problem, and an easily made calibration would give actual variation of values between about 0.985 and about 0.995.

The jet discharged from a conical nozzle usually has some contraction, so that there is a point of minimum diameter shortly beyond the nozzle end. If the authors have any photographs of the actual appearance of the jet, it would be interesting to include them in the paper, and there is no doubt that they would show this contraction. A proper nozzle with well-rounded approach ends with a throat having a parallel portion of sufficient length to properly size the jet, usually about half the diameter, so that the jet leaving the nozzle has the exact diameter of this parallel portion. It is furthermore to be assumed that the curve of the nozzle just preceding the parallel portion is very gentle and blends into the parallel portion smoothly. A curve with a radius equal to 4/3 times the nozzle diameter, and tangent to the parallel portion, gives a satisfactory profile. This curve need only extend for a length equal to $^1/_2$ the nozzle diameter preceding the parallel portion, and there may be a smaller radius preceding, such as $^1/_2$ the diameter. A nozzle practically of such a shape was used by the Bureau of Standards in some measurements of air flow (Bureau of Standards Research Paper, No. 49).

A series of such nozzles, each with a different throat diameter, could be independently bolted to the end of the 29-in. pipe. The cost of such a series, each to be used independently on the end of the 29-in. pipe, would be but little more than the cost of the set of successive cones which the authors actually used. The coefficients mentioned apply to cases where the nozzle diameter is about $^1/_2$ the pipe diameter or less. However, taking account of the contraction which must have existed with the conical nozzles, a 15-in. nozzle with well-rounded approach would probably have discharged the same quantity of water as the 16-in. nozzle that the authors used. This would leave only the 20-in. nozzle as being outside the range of the coefficients mentioned and with which fairly accurate results could be obtained without calibration. However, even a 20-in. nozzle with well-rounded approach could be used if complete calibration was made. With such nozzles with well-rounded approach, the nozzle pressure is most conveniently found by an impact tube in the jet discharged from the nozzle, situated at the center of the jet and a short distance ahead of the plane of the nozzle end. Such an impact tube is very easily located and gives the total nozzle pressure, including velocity head, without requiring any computation of the approach velocity in the pipe. The jet discharged by such a nozzle with well-rounded approach has a constant velocity throughout the cross-section, except for a narrow rim extending in a short way from the outer surface of the jet. The calibration of such a nozzle is made by making a traverse, with a very small impact tube, of this rim, finding the differential pressure between a point at a known distance from the edge of the jet and the impact tube in the center of the jet. This method was first used with nozzles by John R. Freeman, and it is described in a number of papers in the Transactions of the American Society of Civil Engineers.[4]

[4] Trans. A.S.C.E., no. 426, vol. 21, Nov., 1889; no. 479, vol. 24, June, 1891; and vol. 13, p. 382.

This method is also described in a paper in the Transactions of the American Society of Mechanical Engineers.[5] The same method was used in the Bureau of Standards paper quoted.

The authors may fairly be asked if such a type of nozzle and such a calibration would have been more accurate and easier to execute than the method they used, of course assuming that a proper set of nozzles was available.

J. M. Spitzglass.[6] The data disclosed by the authors bring to light a very interesting point in the behavior of fluid flow through a converging nozzle. The test nozzle of constant angle shown by the authors has the form of the approach side of the Herschel venturi. Here in the United States the term "venturi" designates a tube starting with a converging approach of a constant angle leading to a throat and terminating with a receding angle smaller than the approach. In European countries, especially in Germany, the approach side of the venturi is a well-rounded nozzle, so that the term "venturi" in that case has reference only to the receding part of the venturi, while the approach part is the same as any other well-rounded nozzle.

When the writer visited in Germany in 1927, he was surprised to find that the German engineers deviated from the Herschel type of venturi construction. Upon inquiry, he was told that such construction brings about a more uniform coefficient. A mathematical analysis of the case proved that the rounded approach of the same roughness will result in a smaller and more uniform drop of pressure through the approach to the throat of the venturi or nozzle. Since the friction drop through the approach is what reduces the value of the coefficient in proportion to the square root of that drop, any variation in this drop of pressure results in variation of the value of the venturi coefficient.

When the paper was read, the writer immediately realized the explanation for the results which the authors thought were unusual. In the analysis by the calculus the authors found "that the maximum value for $L \times D$, regardless of the nozzle angle, always occurred at a point where the diameter of the nozzle was one-half the diameter at the largest end." The same analysis carried farther indicates that, for a given flow through a nozzle, a maximum drop of pressure occurs when the diameter of the nozzle is one-half the diameter at the largest end. The product of $L \times D$ is also a maximum at the same point, as it should be, because the product $L \times D$ represents the effective rubbing surface or the criterion of frictional resistance in turbulent flow. The total drop through a constant angle converging nozzle is a summation of the increments of the product of $L \times D$. As we integrate these increments for a variable length of approach we find that the summation increases as L is increased to a point where D is equal to one-half of D_1. As D is further decreased, the summation gives a reduced total drop of pressure through the approach even though the length L is considerably increased.

It is of still further interest to note that this disclosure is in full contradiction to the statement made in paragraph C-65, page 67, of the 1931 issue of the American Society of Mechanical Engineers research publication, "Fluid Meters." The paragraph reads:

> When we come to applying this method, the first difficulty we encounter is that the available data refer to venturis of various diameter ratios between 2 and 3 and that there are not enough on any one design to be worth much by themselves. But it turns out upon examination that for a given value of D_2S_2, the value of C is very little affected by variations of the diameter ratio R between

[5] "The Impact Tube," paper 1554, 1916.

[6] Vice-President and Consulting Engineer, Republic Flow Meters Company, Chicago, Ill. Mem. A.S.M.E.

the above limits. This is not surprising; for the major part of the resistance which makes the actual discharge fall below the computed theoretical value must certainly occur in or near the throat, where the speed is highest and the ratio of circumference to cross-section greatest. And it is very probable that if D_2S_2 remained constant, lengthening the entrance cone and increasing the pipe diameter, so as to raise the diameter ratio to 4 or 5, would have very little influence on the value of the discharge coefficient. We therefore ignore the differences of diameter ratio and treat all the venturis on which data are available as if they belonged to the same series and were all geometrically similar to one another.

It is realized, of course, that the nozzle tested by the authors will have lower coefficients than a venturi with the smaller angle of approach and terminating with the rounded throat at the end. The fact remains, however, that the actual drop through the approach reaches a maximum at one-half ratio of throat diameter to pipe diameter. It appears, therefore, that for any given set of conditions the coefficient of the venturi will have a minimum value at the point where the throat diameter is one-half of the pipe diameter, increasing in both directions from that point.

Authors' Closure

As to Dr. Moss' interesting discussion, the authors realize that the set of nozzles discussed in this paper was not designed to give a high coefficient of discharge.

Nozzles made according to Dr. Moss' suggestions would give higher coefficients and also would probably be more accurate than the set discussed in this paper, but the authors' problem was to calibrate a set of nozzles which had been in use for many years so that data from previous tests could be compared with results obtained by other methods.

If a separate nozzle were made up for each size to be attached to the 29-in. pipe, they would cost considerably more than the ones now being used, and also it would require more time and labor for changing nozzles, as well as more space for handling. Large dredging equipment is costly to operate, and any unnecessary time spent during test periods is expensive.

The nozzles are used to determine pump characteristics under different operating conditions and with different runners, and it makes little or no difference what the coefficient of discharge is for any given nozzle if it is accurate. Nozzles of the type discussed in this paper are about as easy to make and use as any set that could be devised, and when accurately calibrated will give good results.

As to Mr. Spitzglass' interesting contribution, the authors have searched all accessible literature and have found nothing concerning nozzles of the type discussed in this paper. Many engineers with whom they have talked have expressed doubt over the findings presented in this paper, although they could find nothing wrong with the data or results, and it is very gratifying for Mr. Spitzglass to give an explanation for the results obtained. It is hoped that more engineers may be interested enough in this subject to do further research work along this line and check the results here presented.

HYD-53-3

A Method for the Standardization of Centrifugal Pumps

By JOS. S. STEPANOV,[1] HARRISON, N. J.

In the author's opinion, the combined application of the preferred-numbers idea and the law of similitude provides an effective means of eliminating unnecessary variety of sizes and of simplifying tool and machinery equipment in any field of industrial production.

In the application of this method to the standardization of centrifugal pumps, the author develops a set of geometric series conforming to the preferred-numbers table. The formation of the series, and their interconnection and use for treating the standardization problem, are illustrated by numerical tables, charts, and diagrams.

Standardizations of hydraulic and mechanical parts of centrifugal pumps are tied together by the use of the law of similitude, resulting in the formation of a set of interconnected geometric series applicable to both cases.



I—INTRODUCTORY

ENGINEERING of late has been making a more and more intensive as well as extensive use of scientific methods in the solution of its problems.

In recent years, and particularly during and since the World War, the problem of waste elimination has been given great attention. The standardization of industrial products, thus preventing the manufacture of an unnecessarily great variety of sizes and simplifying the tool and machinery equipment, has been an important part of the waste-elimination problem, and the subject of a thorough study in this country and in Europe.

As a result of such study it has been established that the development of sizes, to be economically and psychologically satisfactory, should follow a certain natural law of size variation which mathematically is expressed by a geometric series.

Such series of "preferred numbers" have been adopted by German and French industries and informally approved by the American Standards Association and recommended to our industries for a period of trial in practice.[2] A thorough survey of the problem[3] affords substantial evidence showing that the geometric series is most likely to be the natural law of size variation.

Working on the problem of standardization of centrifugal pumps, the author, before he had become cognizant of the preferred-numbers concept, was led to adopt the geometric series as the natural law of size variation. This fact is further evidence that a certain psychological law in human nature directs investigators. Most likely it is identical with the law of Weber-Fechner, which can be formulated as follows: "To produce sensations varying in intensity as an *arithmetic series*, the irritations causing them should vary as a *geometric series*."

II—APPLICATION OF THE LAW OF SIMILITUDE TO CENTRIFUGAL PUMPS

The conceptions and mathematical equations expressing the application of the law of similitude to centrifugal pumps are numerous, and sometimes confusing. Therefore it seems necessary to state here certain invariants which will be used in this paper.

A general inspection of these invariants calls for the following remarks:

Although the units n, q, and p have the same names (unit speed, unit capacity, etc.) as n_l, q_l, and p_l, they express quite different things, n, q, and p being constant for a definite size at different speeds, whereas n_l, q_l, and p_l, are for different speeds and different sizes of the same type. (For symbols used see p. 10.)

	Invariants in terms of H	Invariants in terms of H and D	Invariants in terms of Velocity $\sqrt{2gH}$	Specific speed
Speed..	$n = \frac{N}{\sqrt{H}}$	$n_l = \frac{ND}{\sqrt{H}}$	$K_u = \frac{u}{\sqrt{2gH}}$	$n_q = \frac{NQ^{1/2}}{H^{3/4}} = n\sqrt{q} = n_l\sqrt{q_l}$
Capacity	$q = \frac{Q}{\sqrt{H}}$	$q_l = \frac{Q}{\sqrt{H \times D^2}}$	$K_c = \frac{c}{\sqrt{2gH}}$	$n_{sp} = \frac{NP^{1/2}}{H^{3/4}} = n\sqrt{p} = n_l\sqrt{p_l}$
Power..	$p = \frac{P}{H^{3/2}}$	$p_l = \frac{P}{H^{3/2} \times D^2}$	.....	$m_s = N^2P = n_{sp}^2 H^{3/2}$

On another hand, the pairs n_l, K_u, and q_l, K_c, in spite of their different structures and different names, express the same physical relationship, differing in numerical values by a constant factor.

Capacity Q will be used in gallons per minute, therefore in

---

[1] Research Engineer, Worthington Pump and Machinery Corporation. Mem. A.S.M.E. Mr. Stepanov was graduated from the Imperial Institute of Technology, Petrograd, Russia, in 1908. He then started with the Petrograd Metal Works Co. as designer and was later assistant to the chief designer of its steam-turbine department (1908–1913). Later he was with the Russian Baltic Shipbuilding and Engineering Co., Russia, as chief designer and later chief engineer for the mechanical department in building torpedo-boat destroyers, cruisers, and other craft for the Russian Navy (1913–1918) and with the Machine Works Ilmarine Co., Esthonia, as manager, in building small oil and steam engines (1918–1924). After coming to the United States Mr. Stepanov was temporarily engaged in design and tests of ship propellers with the Akimoff Propeller Co., Philadelphia, and in design and standardization of dry-quenching equipment with the International Combustion Engineering Corp., N. Y. For the last two years he has been engaged in research and development work with the Worthington Corporation.

[2] "Table of Preferred Numbers," A.S.A. Z17-1927.

[3] "Size Standardization by Preferred Numbers," C. F. Hirshfeld and C. H. Berry, *Mechanical Engineering*, December, 1922, p. 791.

Contributed by the Hydraulic Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

the denominator of the n_{sq} formula there appears a constant factor 21.2.

The law of similitude as applied to hydraulic machines involves certain assumptions, one of which is of special importance for the "mechanical" specific speed, N^2P, and the limitations caused thereby will be treated later in the chapter on mechanical standardization.

III—SET OF GEOMETRIC SERIES, THEIR FORMATION AND INTERCONNECTION

1 *Impeller-Diameter or D-Series.* This series is basic for the development of all the other series. It is the result of the assumption that sizes of the impeller diameters, as well as all hydraulically controlling dimensions, of a set of geometrically similar pumps should vary according to a geometric series.

Let us call the ratio of two successive terms of such a series $\lambda = \frac{D_n}{D_n - 1}$ or "length scale" of the pump series

Then the nth term of this D-series is $D_n = D_0\lambda^n$. The impeller diameter D_0 of the model pump being the same for all of the

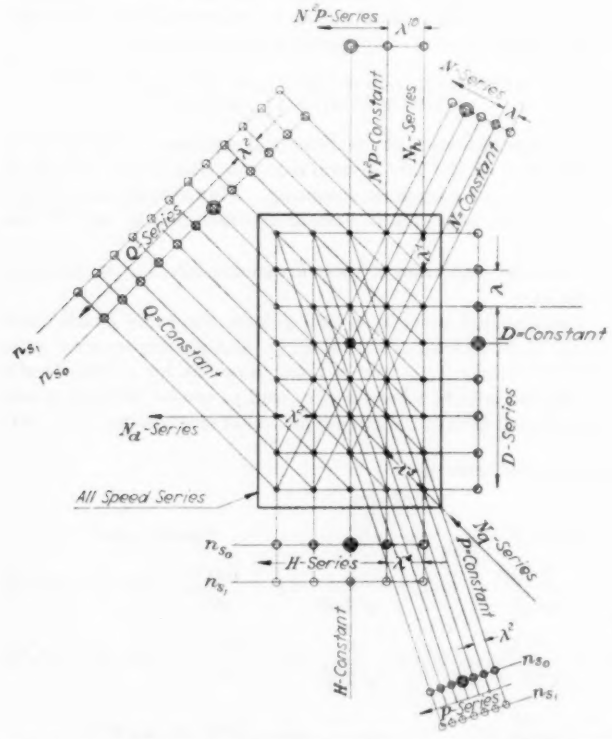


FIG. 1 SCHEME FOR THE INTERCONNECTION OF THE GEOMETRIC SERIES

D-series, can be omitted in further development, and the general term of the D-series reduces to λ^n or: $\lambda^{-n} \ldots \lambda^{-2}, \lambda^{-1}, \lambda^0, \lambda^1, \lambda^2 \ldots \lambda^n$, in which n can have negative as well as positive numerical values.

The number of variables involved in the problem makes it necessary to deal with the following ten series:

D-series	N-series	N_h-series
Q-series	N_q-series	N_α-series
H-series	N_p-series	PN^2-series
P-series		

The functional interconnection of all these series may to a certain extent be visualized by means of the scheme shown in Fig. 1. In this scheme the different rows of circles represent the different series as shown: Q, H, D, P, N, and PN^2. Each of these variables develops one series, whereas speeds N (included in the rectangle, Fig. 1), form the additional series N_α, N_q, N_p, and N_h, according to the row of the constant along which they proceed (D = const., Q = const., P = const., H = const.).

It may be also seen that in this arrangement any set of the corresponding terms of all series can be easily located, as, for instance, those shown on the scheme by the double circles.

Using this scheme we shall proceed with the formation of the various series.

2 *Capacity Series at H = Const. or Q-Series.* Proceeding through the successive terms Nh (H = const.), we pass through the corresponding successive terms of the Q- and D-series. From

$$q_l = \frac{Q_n}{\sqrt{H}D^2_n} = \frac{Q_{n-1}}{\sqrt{H} \times D^2_{n-1}}$$

it follows that

$$\frac{Q_n}{Q_{n-1}} = \frac{D^2_n}{D^2_{n-1}} = \lambda^2 \text{ for } \frac{D_n}{D_{n-1}} = \lambda$$

The term ratio of the Q-series is λ^2, and its general term λ^{2n}.

3 *Speed Series at H = Const. or N_h-Series.* The terms of this series run parallel to the D- and Q-series (Par. 2.). From

$$n_l = \frac{N_n D_n}{\sqrt{H}} = \frac{N_{n-1}\, D_{n-1}}{\sqrt{H}}, \quad \frac{N_n}{N_{n-1}} = \frac{D_{n-1}}{D_n} = \lambda^{-1}$$

The term ratio of the N_h-series is λ^{-1}; the series runs in a direction opposite to that of the D-series and its general term is λ^{-n}.

4 *Head Series at Q = Const., or H-Series.* Proceeding through the terms of N_q-series (Q = const.) we pass through the corresponding successive terms of the H- and D-series. From

$$q_l = \frac{Q}{\sqrt{H_n} \times D^2_n} = \frac{Q}{\sqrt{H_{n-1}} \times D^2_{n-1}}$$

it follows that

$$\frac{\sqrt{H_n}}{\sqrt{H_{n-1}}} = \frac{D^2_{n-1}}{D^2_n} \text{ or } \frac{H_n}{H_{n-1}} = \frac{D^4_{n-1}}{D^4_n} = \lambda^{-4}$$

The term ratio of the H-series is λ^{-4} the series runs in a direction opposite to that of the D-series, and its general term is λ^{-4n}.

5 *Speed Series at Q = Const. or N_q-Series.* The terms of this series run parallel to the D- and H-series (Par. 4). From

$$n_l = \frac{N_n D_n}{\sqrt{H_n}} = \frac{N_{n-1}\, D_{n-1}}{\sqrt{H_{n-1}}}$$

it follows that

$$\frac{N_n}{N_{n-1}} = \frac{\sqrt{H_n}}{\sqrt{H_{n-1}}} \times \frac{D_{n-1}}{D_n} = \lambda^{-3} \text{ for } \frac{D_{n-1}}{D_n} = \lambda^{-1}$$

and

$$\frac{\sqrt{H_n}}{\sqrt{H_{n-1}}} = \lambda^{-2} \text{ according to Par. 4}$$

The term ratio of the N_q-series is λ^{-3} and it runs in a direction opposite to that of the D-series, and its general term is λ^{-3n}.

6 *Speed Series at D = Const., or N_α-Series.* This series

develops parallel to the Q- and H-series, D being constant. From

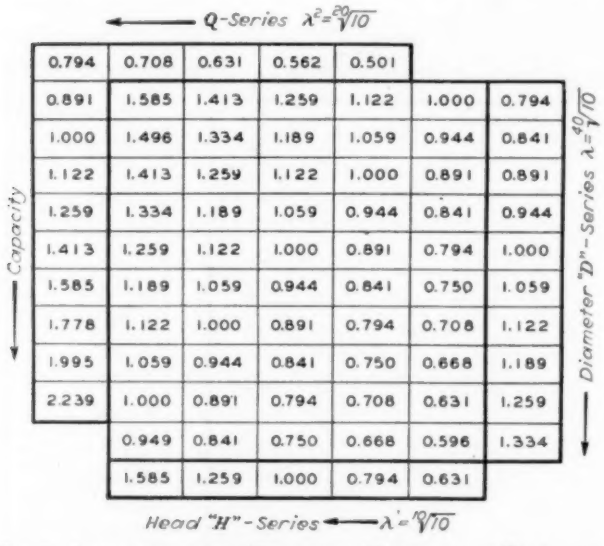
$$\frac{N_n D}{\sqrt{H_n}} = \frac{N_{n-1} D}{\sqrt{H_{n-1}}}$$

it follows that

$$\frac{N_n}{N_{n-1}} = \frac{\sqrt{H_n}}{\sqrt{H_{n-1}}} = \sqrt{\lambda^{-4}} = \lambda^{-2}$$

The term ratio of the N_d-series is λ^{-2}, it runs in the same direction as the H-series (λ^{-4}), and its general term is λ^{-2n}.

7 *Speed Series at Q, H, and D Variable, or N-Series.* Each



0.794	0.708	0.631	0.562	0.501		
0.891	1.585	1.413	1.259	1.122	1.000	0.794
1.000	1.496	1.334	1.189	1.059	0.944	0.841
1.122	1.413	1.259	1.122	1.000	0.891	0.891
1.259	1.334	1.189	1.059	0.944	0.841	0.944
1.413	1.259	1.122	1.000	0.891	0.794	1.000
1.585	1.189	1.059	0.944	0.841	0.750	1.059
1.778	1.122	1.000	0.891	0.794	0.708	1.122
1.995	1.059	0.944	0.841	0.750	0.668	1.189
2.239	1.000	0.891	0.794	0.708	0.631	1.259
	0.949	0.841	0.750	0.668	0.596	1.334
	1.585	1.259	1.000	0.794	0.631	

FIG. 2 ARRANGEMENT OF PREFERRED NUMBERS TO REPRESENT D-, Q-, H-, N-, N_h-, N_d-, AND N_q-SERIES ACCORDING TO SCHEME SHOWN IN FIG. 1

(The range is arbitrary and can be extended in any direction.)

term of this series represents a certain row of constant N, i.e., those common terms of the N_h- and N_d-series which have the same numerical values.

The term ratio of N_d is λ^{-2} and that of N_h is λ^{-1}. It is evident that if we pass one step along the N_d-series (λ^2) in the direction of decreasing terms and two steps along the N_h-series $(\lambda^{-1})^2 = \lambda^{-2}$ in the direction of increasing terms, we shall meet a term of the same numerical value. This gives the location of the row of N = const.

The ratio of the terms represented by each N = const., in other words, the term ratio of the N-series, is λ^{-1}, the same as for the N_h-series, for all successive rows of N = const. pass successively the terms of the N_h-series.

8 *Power or P-Series.* Proceeding through the successive terms of N_h we pass through the corresponding successive terms of the P-, D-, and Q-series, H being constant. From

$$p_I = \frac{P_n}{H^{3/2} D_n^2} = \frac{P_{n-1}}{H^{3/2} D_{n-1}}$$

it follows that

$$\frac{P_n}{P_{n-1}} = \frac{D_n^2}{D_n^2{}_{-1}} = \lambda^2$$

The term ratio of the P-series is λ^2, and its general term is λ^{2n}.

9 *Speed Series at Constant P, or N_p-Series.* Proceeding through the terms of this series we pass through the corresponding successive terms of the Q- and H-series, P being constant. As the ratio of the H-series is λ^{-4}, the ratio of this series will be found as follows. From

$$n_{sp} = \frac{N_n P^{1/2}}{H_n^{5/4}} = \frac{N_{n-1} P^{1/2}}{H_{n-1}^{5/4}}$$

it follows that

$$\frac{N_n}{N_{n-1}} = \frac{H_n^{5/4}}{H_{n-1}^{5/4}} = (\lambda^{-4})^{5/4} = \lambda^{-5}$$

The N_p-series runs in a direction opposite to that of the D-series, and its general term will be λ^{-5n}.

10 *PN^2-Series.* Each successive term of this series corresponds to a certain term of the H-series, for PN^2 = constant is equivalent to H = constant ($PN^2 = n_{sp}{}^2 \times H^{5/2}$). The term ratio of this series is:

$$\frac{(PN^2)_n}{(PN^2)_{n-1}} = \frac{H_n^{5/2}}{H_{n-1}^{5/2}} = (\lambda^{-4})^{5/2} = \lambda^{-10}$$

The PN^2-series runs in a direction opposite to that of the D-series, and its general term is λ^{-10n}.

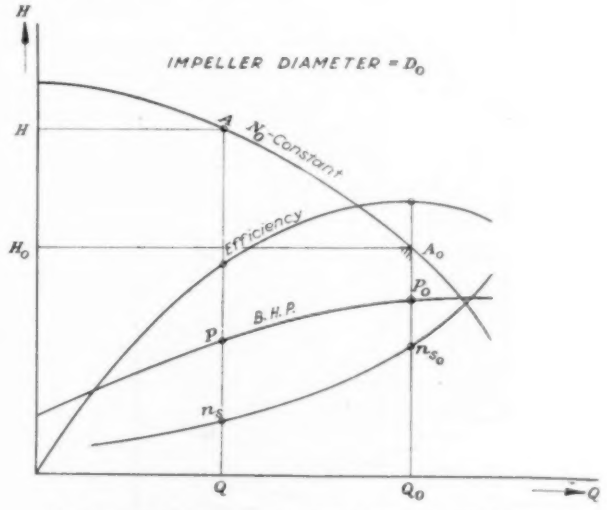


FIG. 3 RELATION OF THE RATING CURVE TO THE SERIES

Arranged in the order of the ratio powers, these various series are given in the following table:

Series	D	N	N_h	Q	N_d	P	N_q	H	N_p	NP^2
Ratio	λ	λ^{-1}	λ^{-1}	λ^2	λ^{-2}	λ^2	λ^{-3}	λ^{-4}	λ^{-5}	λ^{-10}
General Term...	λ^n	λ^{-n}	λ^{-n}	λ^{2n}	λ^{-2n}	λ^{2n}	λ^{-3n}	λ^{-4n}	λ^{-5n}	λ^{-10n}

Neglecting the plus and minus signs, which only show the directions of the series' development, we have only six series, with the ratios λ, λ^2, λ^3, λ^4, λ^5, and λ^{10}.

It is obvious that all the terms of all these six series are included in the first one, with ratio = λ; omitting certain terms of this series and taking only those corresponding to the power of λ, we can get any other series. This fact immensely simplifies the further treatment of the problem; for, in terms of the preferred-numbers idea, all these series are nothing else than different "orders of preference" of the same table of the preferred numbers.

IV—METHOD OF APPLICATION TO THE STANDARDIZATION OF CENTRIFUGAL PUMPS

1 *Numerical Values of the Series.* Tables of preferred numbers recommend the ratios: $\sqrt[80]{10}$, $\sqrt[40]{10}$, $\sqrt[20]{10}$, $\sqrt[10]{10}$, $\sqrt[5]{10}$.

It seems to the author that $\lambda = \sqrt[40]{10}$ can be adopted as ratio of the D-series. If, for certain types or certain services, the sizes appear to be too close to each other, λ can be easily changed to the next order of preference by simply omitting every alternate term from this series.

Preferred numbers having a ratio $\lambda = \sqrt[40]{10}$ are arranged in Fig. 2 according to the scheme shown in Fig. 1, excluding, however, the P- and N^2P-series.

Inspection of Fig. 2 shows that no numbers are used except from the series $\sqrt[40]{10}$, yet in the arrangement of the scheme all derivative series, with their ratios, appear in the expected places.

Another check on the table is the fact that any set of corresponding values of the H-, Q-, and N-series substituted in the expression of the specific speed $n_{sq} = N\frac{\sqrt{Q}}{H^{3/4}}$ should always give the value unity.

Similar tables can be arranged also for P and N^2P, following the indications of the scheme shown in Fig. 1.

2 *Application of the Preferred-Numbers Table to a Concrete Type of Pump.* Let us assume that the test curves of one of the sizes of a certain type of pump to be standardized are as given in Fig. 3. These show that a pump of impeller diameter D_0 at the speed N_0 delivers Q_0 against H_0 and calls for P_0, the maximum efficiency being right at this point of operation and the figured specific speed being

$$n_{sq0} = \frac{N_0 Q_0^{1/2}}{H^{3/4}}$$

Using the table of Fig. 2, we can construct another table simply by multiplying the terms of the capacity, head, speed, and diameter series by Q_0, H_0, N_0, and D, respectively.

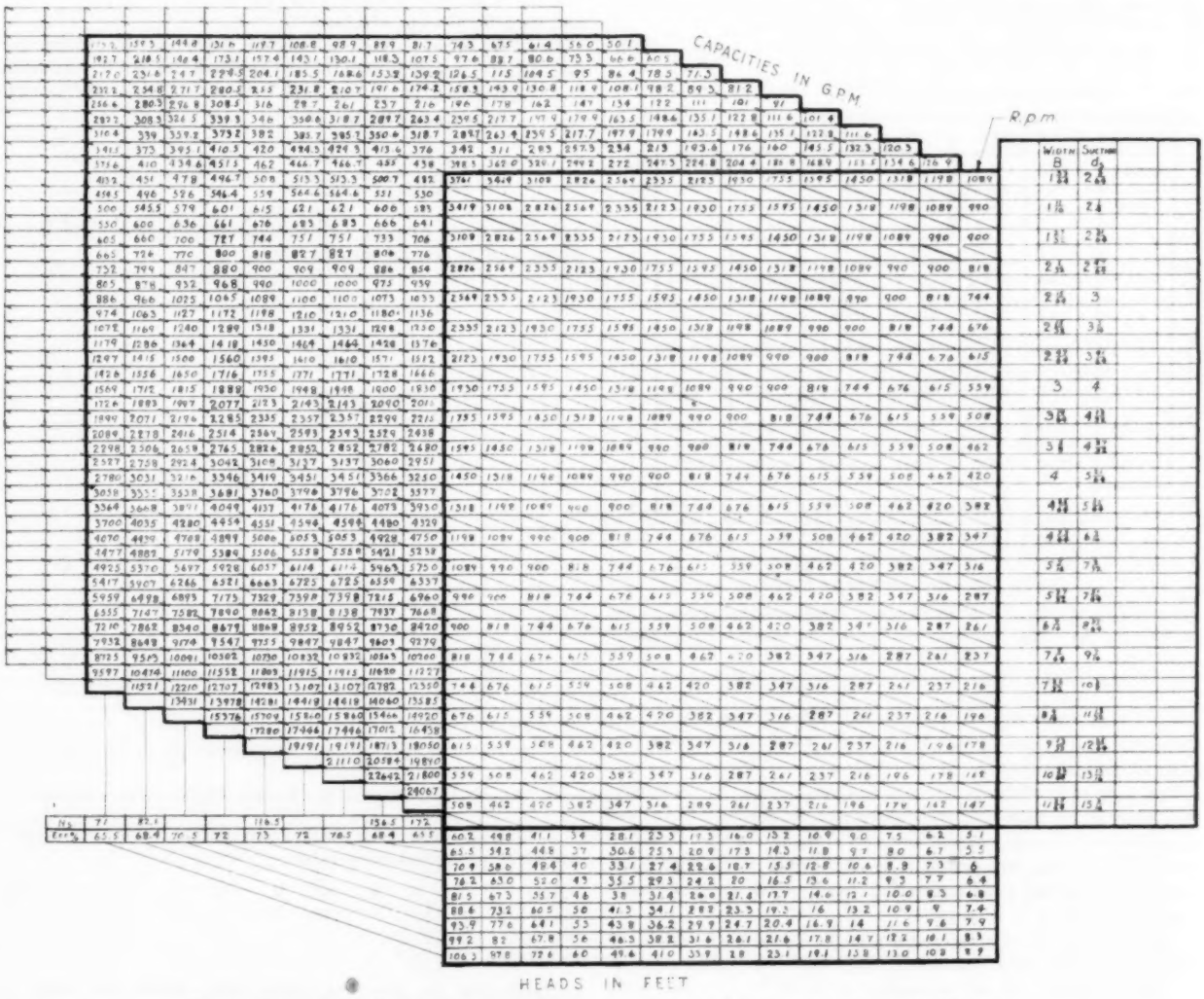


FIG. 4 TABULATION SHOWING STANDARDIZATION OF A PUMP ACCORDING TO AUTHOR'S METHOD

The specific speed for all sets of the corresponding N's, Q's, H's, and D's in such a table remain the same: $n_{sq0} = \frac{N_0 Q_0^{1/2}}{H^{3/4}} \times 1$, for, as was stated in the preceding paragraph, the corresponding values of the preferred numbers substituted in the specific-speed formula always give 1 as the result.

For another point A of the head-capacity curve, the same impeller D_0 at the same speed N_0 delivers Q and generates H at a specific speed n_s and a lower efficiency. Obviously, for this point the N- and D-series remain as for the previous case (point A), and only new Q- and H-series should be added by multiplying the capacity series (Fig. 2) by Q and head series by H.

In a similar way the Q- and H-series may be added for as many points of the head-capacity curve as desired.

Fig. 4 is a tabulation showing the standardization of a pump according to this method, the numerical values of the impeller D being omitted.

The blank spaces in the N- and D-series show that it was decided for the time being to actually develop only every alternate term of the series, i.e., to use the next order of preference.

The number of terms of all the series (Fig. 2) is theoretically unlimited. Conforming to the special given type, the range of the H-, Q-, D-, and N-series and even the order of preference should be selected according to the field to be covered, the demand of the market under consideration, and the lowest acceptable efficiency.

3 *Graphical Presentation.* The developed geometric series may be presented in convenient graphical form on logarithmic paper, as shown in Fig. 5, which includes all the series given in Fig. 1.

Using the scheme of Fig. 1 and keeping in mind that the logarithms of the terms of a geometric series form an arithmetical

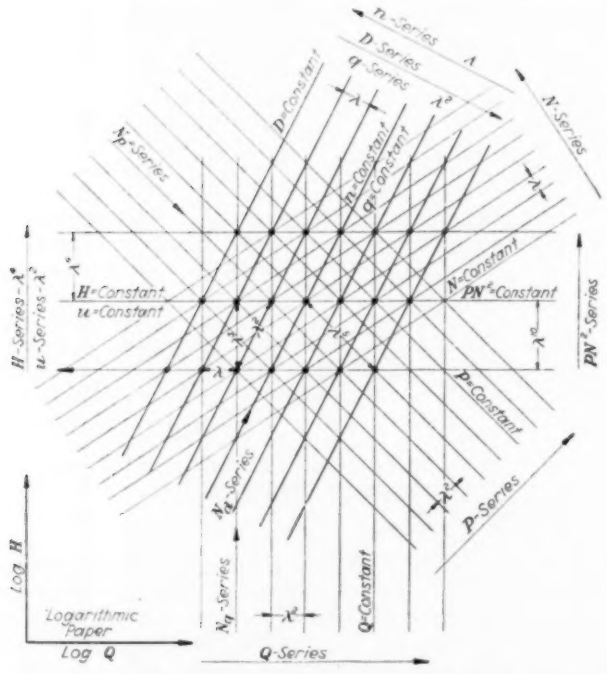


Fig. 5 Graphical Method of Presenting All the Series, on Logarithmic Paper

series, and therefore the distances between the terms of each series on logarithmic paper are equal, it seems to the author that no further explanations are necessary.

However, attention is called to the fact that certain other derivative series are presented on the same chart, namely,

(*a*) *Unit Speed or n-Series:* $n = \dfrac{N}{\sqrt{H}}$. The terms of this series are located along the D = const. line, for the same line is n = const. The ratio of this series is derived from the relationship $n = n_1/D$. As n_1 is constant for all sizes of the type, it follows that the ratio of this series is the same as for the D-series, i.e., λ^{-1}.

(*b*) *Unit Capacity or q-Series:* $q = \dfrac{Q}{\sqrt{H}}$. The same reasoning shows that the terms q = const. coinciding also with D = const., form a q-series with the ratio λ^2 for $q = q_1 D^2$, q_1 being constant for all sizes.

(*c*) *Peripheral Velocity or u-Series.* This series runs parallel to the H-series, H = const. being also u = const., for $K_u = \dfrac{u}{\sqrt{2gH}}$ is, like n_1, constant for all sizes. The term ratio of this series follows from the relationship:

$$\frac{u_n}{\sqrt{2gH_n}} = \frac{u_{n-1}}{\sqrt{2gH_{n-1}}} \quad \text{or} \quad \frac{u_n}{u_{n-1}} = \frac{\sqrt{H_n}}{\sqrt{H_{n-1}}} = \lambda^{-2}$$

4 *Simplification of Plotting.* The plotting of the chart is highly simplified if all regularities of the series relationship are

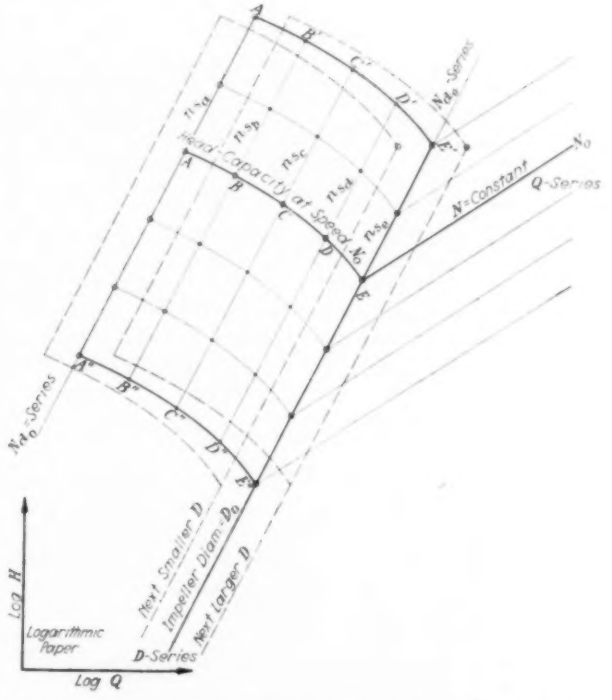


Fig. 6 Method of Plotting Field Covered by Impeller of a Given Size

fully utilized. The method of plotting is shown diagrammatically in Fig. 6.

It is not necessary to plot all the points given in the table of the numerical values, Fig. 4.

To plot the field covered by an impeller of size D_0 (model impeller), in the selected range of the head-capacity curve (range of efficiencies and specific speeds) $ABCDE$ at speed N_0, it is sufficient to plot only the points shown in Fig. 6 by circles.

This means the plotting of $Q_0 - H_0$ points along the N_{d0}-series, first for the highest specific speed n_{se}-points along $E' - E''$, and, second, for the lowest specific speed n_{sa}-points along $A' - A''$. This will give extreme points A and E of the selected range of the head-capacity curve at the speed N_0.

The intermediate points B, C, D of the same curve should be plotted, using the pairs of values Q and H for the corresponding values n_{sb}, n_{sc}, and n_{sd} at the same N_0. Thus the shape of the $ABCDE$ curve is determined. As the shape of the curve on the logarithmic paper does not change along all speeds (and all sizes), it is sufficient to transport this curvature between the points given on $A'A''$ and $E'E''$.

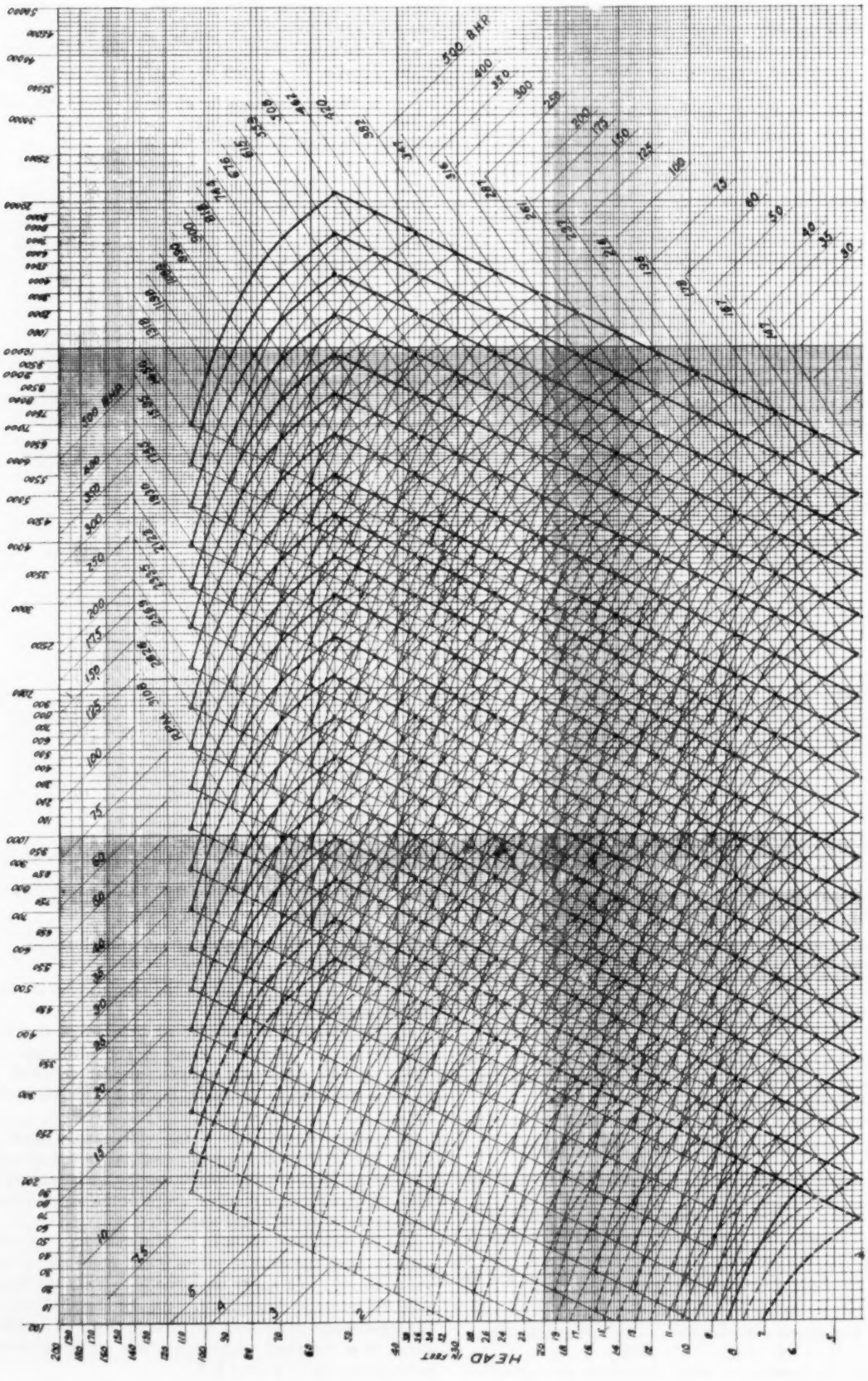


FIG. 7 TYPICAL EXAMPLE SHOWING STANDARDIZATION OF ONE OF A SERIES OF PUMPS FOR MANUFACTURE

The points of these head-capacity curves, such as B', C', D' or B'', C'', D'', corresponding to the points B, C, D of the model curve, are located on the n_s = const. lines $B'B''$, $C'C''$, $D'D''$ which are parallel to $A'A''$ and $E'E''$ and pass through the points B, C, D.

The intersections indicated by dots are corresponding to B-, C-, D-points, and can be checked by the numerical values given on the table (like the one in Fig. 4).

Thus the field covered by the D_0 impeller is plotted showing 6 head-capacity curves, corresponding to 6 different speeds N from the N-series.

For other sizes of impeller (D-series) it is sufficient to move the field $A'A''E''E'$ parallel to itself in the Q-direction by equal spaces, corresponding to the location of the D-const. lines, for instance, along the Q-series.

The corner point E' will move to E'_L for the next larger size, or to E'_s for the next smaller one.

The accuracy can be checked by using the numerical values from the table, as above.

Similarly, using the same method, the b.hp.-curves can be plotted, the difference being that the location of corresponding points on these curves should be determined by the pairs of numerical values of P and Q along the N_{s0}-series.

As example of such a plotting, the chart, Fig. 7, is given which presents the standardization of a type of pump. On this chart, for clarity's sake, the intermediate points of the curves, like B, C, D, in Fig. 6, are not emphasized by lines like $B'B''$, $C'C''$, and $D'D''$.

Corresponding to the numerical values of the table, Fig. 4, the D-series and N-series are of the next order of preference, any second term of the original series being omitted.

The numerical values of the D-series, being unnecessary for the purpose, are also omitted.

The b.hp.-lines, shown only at their ends, represent standard motor sizes, at the lowest efficiency of the pump under consideration, i.e., any condition point located on a certain line calls for maximum b.hp. indicated at the ends of this line.

V—USE OF THE METHOD FOR STANDARDIZATION OF DIFFERENT SERVICES

1—Specific-Speed Series

In the preceding pages the term "type" has been used to connote the constancy of the specific speed for all sizes of the particular class of pump being considered, i.e., all pumps of the series being geometrically similar.

To cover the field required by a certain kind of service, several "types" in this sense of the word are usually necessary, i.e., impellers of different specific speeds.

In deriving all the preceding series, the specific speed n_s has always been kept constant. This is equivalent to saying that only the impeller diameters D and speeds N have been varied to obtain all the variety of capacities and heads.

In practice it is frequently necessary to reduce the number of impeller diameters (casings) or speeds (drivers) to certain limits dictated commercially.

Therefore, variation of specific speeds, generally necessary to reduce variety of sizes and speeds, represents the only means of best utilizing certain expensive casings or certain motor speed (high-pressure multi-stage casings).

Specific-Speed Series at Constant Speed, or N_s-Series. All the speed series (N, N_h, N_d, and N_q) represent the variation of speed, with different variables being kept constant, needed to deliver a corresponding Q and generate a corresponding H, the specific speed being always constant.

The problem now is to deliver the same capacities Q and generate the same heads H, varying the specific speed n_s and keeping speed N constant. The specific-speed formula shows what will happen:

$$n_s = \frac{NQ^{1/2}}{H^{3/4}}, \qquad \text{or} \qquad \frac{n_s}{N} = \frac{Q^{1/2}}{H^{3/4}}$$

As Q and H are assumed to remain the same, evidently n_s/N should remain the same. This means: if n_s = constant and N forms a series with ratio λ^n, the result will be the same if N = constant, and n_s forms a series with the ratio λ^{-n}, for

$$\frac{n_{s0}}{N\lambda^n} = \frac{n_s\lambda^{-n}}{N_0}$$

This relationship justifies the following statements: *Speeds N and specific speeds n_s in all speed series are interchangeable*, with the change only in the direction of the variation, the ratios remaining the same. Lines of N = const. at variable speeds and constant specific speeds turn into lines of n_s = const. at variable n_s and constant N.

Let us investigate a field $ABCD$ (Fig. 8) covered by 4 impeller diameters and 8 speeds, all impellers being geometrically similar, i.e., n_s, n_l, q_l = const. Let us assume that all speeds are constant and that the N-series is replaced by the n_s-series directed as shown in Fig. 8. We then see that for AB impeller (as for each of the rest) there is only one speed, but two specific speeds are used. Unit capacities q and q_l remain the same as before, but unit speeds n and n_l vary:

$$\frac{n_n}{n_{n-1}} = \frac{N}{\sqrt{H_n}} \times \frac{\sqrt{H_{n-1}}}{N} = \sqrt{\frac{H_{n-1}}{H_n}} = \lambda^2$$

for both series, D being constant. This shows that n_s is changing only on account of change of unit speeds (n, n_l), unit capacities being constant.

Inspecting now the N_h-series, we see that along this series speeds are constant; i.e., corresponding terms of the D-series should also be constant, for these two series have the same ratio running in opposite directions, whereas the specific speeds in this direction is varying with ratio λ.

Along the same direction (AC) n is constant $\left(= \frac{N}{\sqrt{H}}\right)$ and $n_l \left(= \frac{ND}{\sqrt{H}}\right)$, but q and q_l form a series with λ^2. Therefore it can be stated that keeping D = constant in the direction n_{sh}, we should vary q and q_l at constant n and n_l.

As a final result it can be seen that all the field $ABCD$ can be covered by one impeller diameter at one speed, but using eight different specific speeds.

It is interesting to note here that the lines of constant specific speed do not always represent geometrically similar impellers: both the shape of the vanes (n_l) and the width of the impellers (q_l) are varying. This may indicate that the shapes of the operating curves are not necessarily the same, and this fact may call for a certain overlapping of the range of the separate impellers to cover completely the field under consideration.

The number of specific speeds can be reduced to any extent if a certain variation of impeller diameters and speeds is possible. For instance, a certain field (Fig. 9) equivalent to 9 parallelograms like one shown in Fig. 8 can be covered by 3 impeller diameters, 6 speeds, and 4 specific speeds. In the arrangement shown here, the same speeds and specific speeds are used for several impeller diameters and the same diameter is also used at several speeds and several specific speeds.

Such combinations of speeds (N), diameters (D), and specific

speeds (n_s) provide a flexible means of reducing the number of terms of any one of these series to one, correspondingly increasing the numbers of the terms of other series.

It can easily be seen that the problem is always how many terms of the D, N, or n_s-series should be omitted and how many used—in other words, what order of preference should be selected for each of these series.

Treating the actual problems of standardization of centrifugal pumps for different services covering different fields, it is usually necessary to reduce the number of impeller diameters, conforming to the number of casings, which latter should be reduced to a minimum, or to reduce the number of speeds, conforming to the standard speeds of the motor drivers.

In addition to the shape of the efficiency and the head-capacity curves, the lowest competitive efficiency, as affecting the operating range of the curves, is to be considered in the final selection from all possible combinations.

In the multi-stage pumps, the number of stages is an additional variable which, in combination with n_s-series, provides still further facilities for reduction of the number of impeller diameters and speeds required to cover a definite field of service.

Hydraulic engineers are familiar with the exceptional flexibility of centrifugal pumps. This feature is clearly expressed in the multitude of the series, in their various interrelations: 3 fundamental series (N, D, n_s) and their derivatives provide an almost unlimited variety of the Q and H.

The method developed in this paper furnishes a means of utilizing this feature intelligently for the elimination of an unnecessary variety of impeller diameters and specific speeds, and for unification of the chief dimensions, thus avoiding unnecessary variety of different parts.

Using when necessary different orders of preference, it is possible to establish *a single set of impeller diameters, a single D-series, for all kinds of services and designs, i.e., for the entire range of output* of a centrifugal-pump manufacturing company.

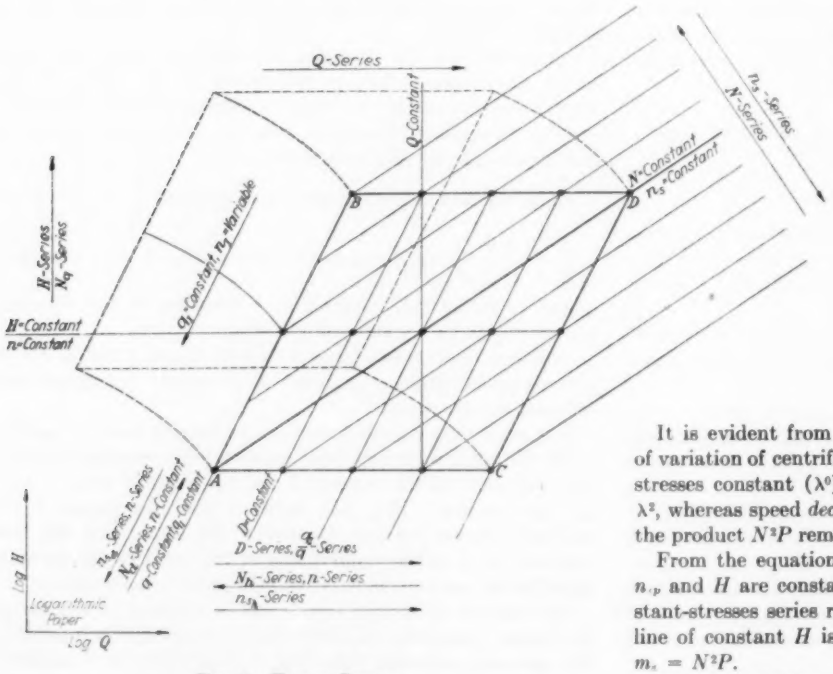


FIG. 8 THE n_s-SERIES

2—CONSTANT-STRESSES OR N^2P-SERIES AND MECHANICAL STANDARDIZATION

The specific speed in terms of power absorbed by pump is

$$N_{sp} = \frac{NP^{1/2}}{H^{5/4}}$$

It can be proved that the numerator $NP^{1/2}$ is a mechanical invariant of the centrifugal pump. Let us call it $m_s = N^2P = n^2_s H^{5/2}$. This invariant can be called also "mechanical specific speed" of a centrifugal pump, and is characteristic for the mechanical properties of similar pumps.

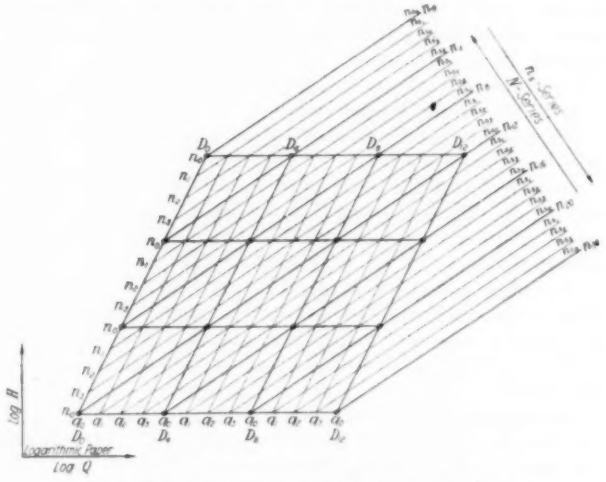


FIG. 9 COMBINATION OF THE D-, N-, AND n_s-SERIES

Using the same "length scale" as before, the ratios for the mechanical properties of geometrically similar centrifugal pumps will be as follows, for:

Sectional areas ... Surfaces	λ^2
Masses	λ^3
Velocities	λ^0-const.
Speeds	λ^{-1}
Accelerations	λ^{-1}
Mass forces	λ^3
Working pressures	λ^0-const.
Working forces	λ^2
Stresses	λ^0-const.
Power	λ^3

It is evident from these ratios that at the "length scale" λ of variation of centrifugal pumps, with velocities, pressures, and stresses constant (λ^0), power *increases* directly proportional to λ^2, whereas speed *decreases* directly proportional only to λ; i.e., the product N^2P remains constant.

From the equation $m_s (= n^2_{sp} H^{5/2} = N^2P)$ is constant when n_{sp} and H are constants, i.e., it indicates that the m_s- or constant-stresses series runs parallel to the H-series and that each line of constant H is the line of constant stresses, or constant $m_s = N^2P$.

The ratio of this series can be derived from the relationship:

$$\frac{P_n N_n^2}{P_{n-1} N^2_{n-1}} = \left(\frac{H_n}{H_{n-1}}\right)^{5/2} = \lambda^{10}$$

If $\lambda = \sqrt[60]{10}$, $\lambda^{10} = \sqrt[6]{10} = 1.778$. This means if we pass

from one term of the H-series to the next, the stresses will increase or decrease 1.778 times as much.

Certain limitations of the application of the mechanical invariant are due to the fact that the law of similitude neglects gravitational forces. For instances, the stresses due to the weight of the parts themselves in similar pumps do not remain constant, for weights increase with λ^3 and sections only with λ^2. The difference is caused by the fact that the acceleration due to gravity (g) remains constant, whereas the acceleration of the moving masses decreases with λ^{-1}. For the case of centrifugal pumps, *this deviation from the mechanical invariant is of importance only for the shaft deflections and critical speeds.* Let us therefore find the law of variation.

Deflections:

$$\frac{f_n}{f_{n-1}} = \frac{W_n}{W_{n-1}} \times \frac{L^3_n}{L^3_{n-1}} \times \frac{d^4_{n-1}}{d^4_n} = \lambda^3 \times \lambda^3 \times \lambda^{-4} = \lambda^2$$

Critical speeds:

$$\frac{N_{cn}}{N_{cn-1}} = \sqrt{\frac{f_{n-1}}{f_n}} = (\lambda^{-2})^{1/2} = \lambda^{-1}$$

This shows that the critical speeds decrease when the sizes of shafts, as well as of pumps, increase according to the D-series. But along the constant-stresses (m_s) or H-series or N_h series, the operating speeds, N_h-series, decrease in the same ratio (λ^{-1}): this means that along the constant-stresses series the *margin of safety against the critical speeds is also constant.*

The deflections increase with λ^2, whereas all dimensions increase with λ, and there is no compensation to cover this difference in the variation ratios. Therefore the following is of importance in designing the bearings: If the dimensions of the shaft are kept similar to all other dimensions of the pump, the *limit of deflection allowed by the type of bearings should not be exceeded.*

Thus we see that along the lines of constant stresses everything remains equally safe except the deflection. But when we move from one such line to the next, the stresses vary according to the stresses series in the enormous ratio λ^{10}, and this calls for special attention if the same pump is to be used for higher r.p.m.

Out of the above-stated relationship between the mechanical and the operating properties of centrifugal pumps, an interesting problem arises in the mechanical standardization of pumps.

As in the hydraulic standardization the range of the efficiency fluctuations (range of operating curve and range of operating speeds) is of paramount importance, so in the mechanical standardization the range of stresses (efficiency of utilization of the material) is a controlling factor.

To keep the *constant stresses* along all size series we should have not only as many sizes as there are impeller diameters, but for each impeller size as many mechanical dimensions (wall thicknesses, shaft diameters and lengths, etc.) as there are terms in the constant-stresses series.

On the other hand, if we do not change these dimensions from one line of constant stresses to another, it will mean a poor utilization of the strength of the material according to the range of heads (or terms of the N^2P series), the stress variations may be many times 1.778.

This is a case frequently encountered in engineering practice when certain contradictory requirements find their reconciliation in a properly selected compromise. This can be done by introducing certain series of constant stresses at constant dimensions.

Torque, or T-Series. The conception of unit torque and its formula $t_l = T/HD^3$ provide a means of forming a torque series and finding its term ratio. Proceeding through the terms of the N_h-series (H = const.) we pass through the corresponding successive terms of the T- and D-series.

From

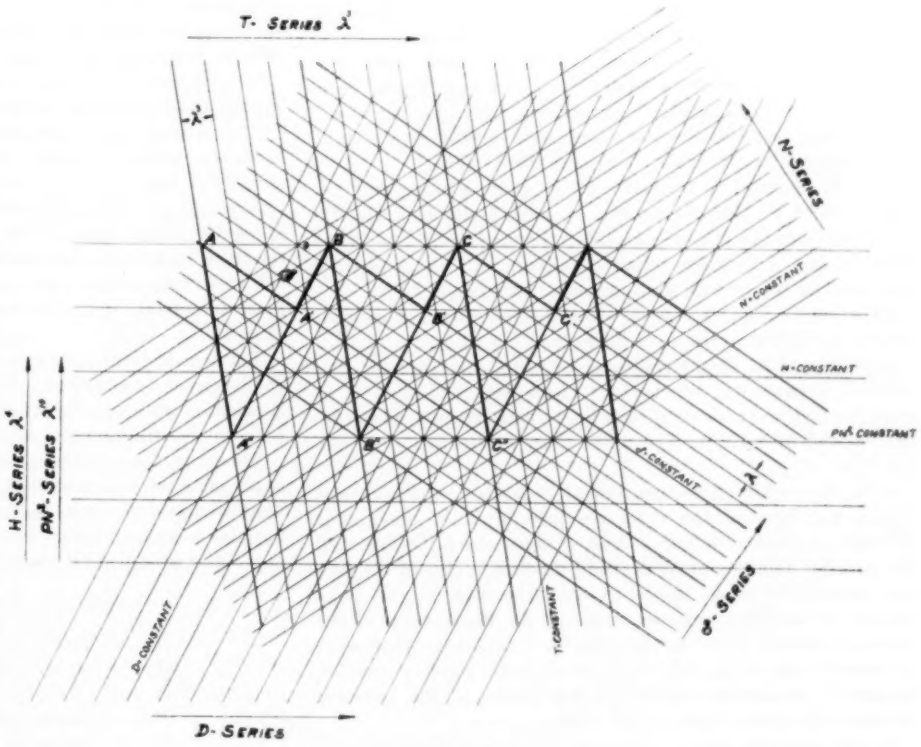


FIG. 10

$$t_l = \frac{T_n}{HD_n^3} = \frac{T_{n-1}}{HD_{n-1}^3}$$

it follows that

$$\frac{T_n}{T_{n-1}} = \frac{D_n^3}{D_{n-1}^3} = \lambda^3$$

The term ratio of the T-series is λ^3, it runs parallel to the D-series, and its general term is γ^{3n}.

The location of the lines of T-const. (Fig. 10) is determined by the relationship: T = const. $\times$ P/N. This relationship shows that the lines of constant torque should pass through each alternate term of N-series (λ^{-2}) and each term of P-series (λ^2).

The lines T-const. are simultaneously the lines of constant torsional stresses at constant shaft diameters.

Constant Wall Thickness or s-Series. In a similar way, using, for example, "the boiler formula" for the stresses in casings due to the internal pressure (σ = const. $\times$ $HD/2s$), we can form a series of constant casing stresses (σ = const.) at constant wall thickness (s = const.).

Proceeding through the terms N_h-const. (H = const.) we pass through the corresponding successive terms of the s-series and D-series. From

$$\sigma = \text{const.} \times \frac{HD_n}{2s_n} = \text{const.} \times \frac{HD_{n-1}}{2s_{n-1}}$$

it follows that

$$\frac{s_n}{s_{n-1}} = \frac{D_n}{D_{n-1}} = \lambda$$

The term ratio of the s-series is the same as that of the D-series, i.e., λ, and its general term is λ^n.

The location of the lines of constant thickness s = const. at constant stresses is determined by the same relationship σ = const. $\times$ $HD/2s$ at s = const. Obviously these lines pass through each term of the H-series (λ^{-4}) and each fourth term of the D-series (λ^4), Fig. 10. Along these lines of s = const., the stresses σ are constant at *constant* wall thicknesses, in spite of the casing size variation, following the D-series.

Now, in addition to lines of *constant stresses* along N^2P = const., where all dimensions *vary* in accordance with the D-series, we have the lines of constant stresses (T-series, s-series) where certain dimensions (shaft diameter, wall thickness) are kept *constant* in spite of the pump-size variation.

Referring to Fig. 10, we can state the following results concerning, for instance, the shaft diameters:

1 If the shaft diameters *vary* following the D-series, the torsional stresses will be constant along each line of N^2P = const., but *different* for each such line; thus, stresses along the line AB will be $1.778 \times 3 = 5.334$ times as great as along the line $A'B'$.

2 If the shaft diameters remain constant, the torsional stresses will be constant along the lines of T = const., but different for each such line; thus, stresses along the line AB' will be 1.995 times as great as along the line BB''. In the first case, using four shaft diameters, corresponding to the same number of impellers, we should expect the fluctuations of the torsional stresses to be in the ratio of 1 to 5.334, whereas in the second case, using *only one* shaft diameter for four impeller diameters, we should expect the fluctuation of the torsional stresses to be in the ratio of 1 to 1.995.

Similar reasoning leads to the same conclusion concerning the stresses in the casing.

The number of impellers (casings) to be included in each group with the same shaft diameter and wall thickness depends on the stress fluctuation which is considered commercially permissible: *The greater the fluctuation in stresses allowed, the smaller the number of different sizes needed.*

This conclusion is similar in all respects to the one derived for the hydraulic standardization. A statement expressing the same relationship for both hydraulic and mechanical standardization might read as follows: *The greater the fluctuation in the stresses as well as in the pump efficiency that is allowed, the smaller the number of different sizes needed to cover a definite field of service.*

At any rate, whatever selection is made following this method of standardization, all the dimensions of the mechanical parts will be included in the original table of preferred numbers if the dimensions of the model pump are preferred numbers.

VI—CONCLUSION

The law of similitude and the relations used in the development of this method are well known to hydraulic engineers. Therefore the author will confine himself to calling attention to certain points only. The method is accurate to the extent to which the law of similitude, as expressed in the equations used, accurately describes the actual performance of centrifugal pumps.

It is known that efficiency does not remain constant at constant n_s, but changes following the variation of sizes and speeds. The capacity is inclined to increase at the same n_s with the increase in size.

The deviations from the law of similitude are the greater the further the dimensioning of the pump, or its operating conditions, is from the normal state of a smooth channel flow and uniform distribution of velocities and pressures (versus shocks, cavitation, etc.).

Whereas these deviations are not of importance as regards the standardization itself, they should nevertheless be taken into consideration if the standardization chart is used as a rating chart for the preliminary selection of the size of pump to meet definite operating conditions.

In such cases correction factors based on the tests should be applied to the readings on the chart.

This method can be used for the standardization of other kinds of turbo-machines, properly adjusted to the individual peculiarities of the case.

The author's strongest conviction is that the method can be applied to the standardization of any industrial product if the relations expressing the law of similitude are established, for the preferred-number idea and the law of similitude are the most completely generalized principles which have ever been known.

The law of similitude was used for standardization of steam engines as early as 1894 by Normand.[4] At the present time, in connection with standardization problems, the subject deserves much more attention and study[5] than ever before.

Therefore the purpose of this paper has been not only the application of this method to the standardization of special kind of machines, as in the present case of centrifugal pumps, but also to arouse interest in its further study and application to any object of industrial production.

SYMBOLS USED IN THE PAPER

H	=	head in ft.
Q	=	capacity in cubic feet per second
N	=	r.p.m.
P	=	horsepower
T	=	torque in foot-pounds
D	=	diameter of impeller in feet
n, q, p, t	=	unit speed, capacity, etc., for the same size
n_1, q_1, p_1, t_1	=	unit speed, capacity, etc., for different sizes
n_{sq}	=	specific speed in terms of capacity
n_{sp}	=	specific speed in terms of power
m_s	=	mechanical invariant
u	=	peripheral velocity of the impeller in feet per second
C	=	flow velocity in feet per second
N_h, N_q, N_p, etc.	=	speed at constant H, Q, P, etc.
γ	=	length scale, term ratio of D-series
σ	=	stress in pounds per square inch
s	=	wall thickness

[4] Jauch et Masinejean. "Cours de machine marines," part 2, Paris-Toulon, 1910, p. 570.

[5] Prof.-Dr. W. Kummer, Ing., Zurich, "Wachstumgesetze und spezifische Drehzahlen der Maschinen," *Schweiz. Bauzeitung*, 1927.

Discussion

J. W. Macmeeken.[6] The exceedingly clear presentation of the subject by the author is refreshing. The mathematical relationships of all the functions of the variables which come under the laws of dynamic similarity are treated throughout from the viewpoint of infinite series, of which any one pump or particular machine is merely one example. The interrelation of these series as functions of the same fixed ratio is expressed with a clearness and consistency that are new in pump literature covering these aspects of centrifugal-pump variations. This method of formulating the laws of variation is applied by the author with great advantage in showing the relationship between "the linear ratio" in its hydraulic aspects and its significance in design for strength of materials.

Diagrams on logarithmic paper such as the author uses have been employed by designers for many years past as a guide for consistent reasoning and comparison of results. This period of their use has seen us through the perfection of the hydraulic turbine and the development of the centrifugal pump to a degree of perfection at last equal to that of the turbine in several important types, but the period of development so far as pumps are concerned is far from completion. Later information acquired in the last few years has afforded the designer unexpected knowledge of limitations of application of the laws of type similarity applied to centrifugal pumps. For instance, a large pump which gives a satisfactory performance with high efficiency has been found to give exceedingly poor results in a geometrical model of considerable size reduction. Conversely, a conveniently dimensioned model when stepped up to a very large pump has been known to fail to perform according to the law of type similarity in some instances, and in others has given a slightly different performance from that expected from calculations. The known difference in dimensions between a satisfactory large pump of economical design and a satisfactory small pump of economical design is so great in the case of a linear multiplying factor of 4 that there must obviously be a continuous variation in the best proportions in all intermediate multipliers in a series of the same type. This is explainable by considering that the laws of type similarity are assertive in maintaining similar ratios in these elements of the machine which involve dynamic effect; for instance, in the proportioning of the maintenance of a forced vortex as a means of generating head. Certain parts of the hydraulic field of fluid flow in a centrifugal pump are not subject to dynamic effort on the part of the rotor; for instance, the approach to the impeller by any of the usual designs of suction casing or the distance through which the fluid travels between the impeller and the exit of the casing are elements that should scarcely be subjected to factor multiplication. In the impeller itself there are flow passages which operate under a different Reynolds number for different pump sizes. Dynamically, an impeller characteristic type is one which is dynamically suitable for a certain maximum total dynamic head with certain concomitant velocities. Therefore the numerical value of $DV\rho/\mu$, known as the Reynolds number, must be different for each pump size for the same fluid. The variation of radii of curvature in different sizes for the same velocity of flow as determined by type characteristics results in great variations in "radiating current" and "induced vortices." For example, an irrigation ditch with water flowing under a velocity of 4 ft. per sec. at a bend will produce elbow flow free from independent vortices. The same channel proportions in a large river with the same velocity of flow have been known to result in a complete circular whirlpool, with severe loss in head.

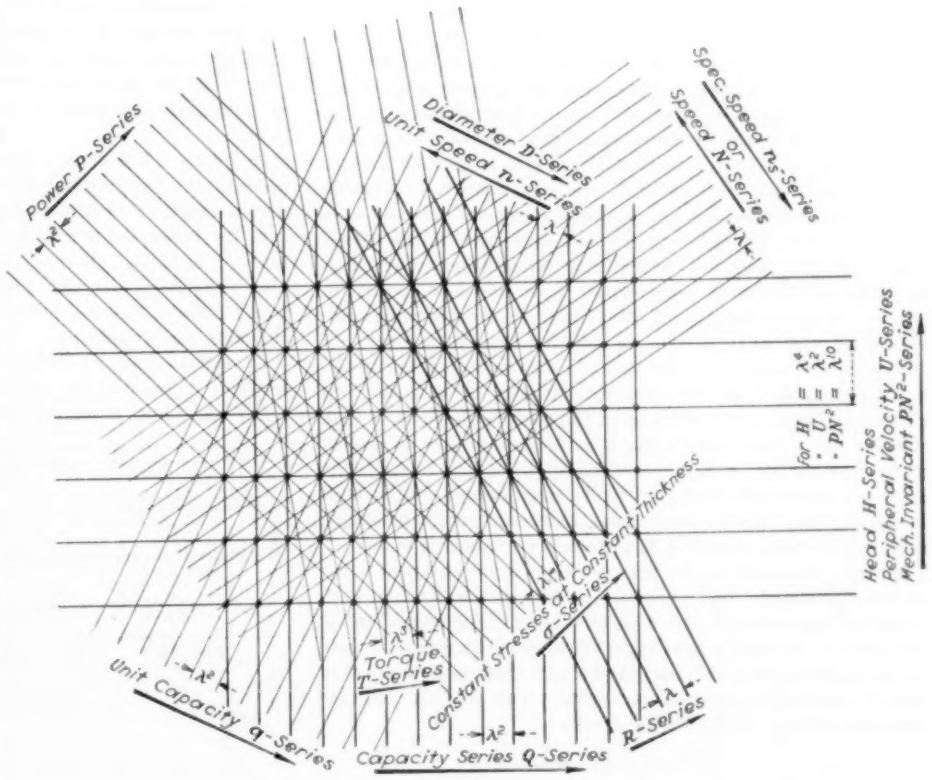


Fig. 11 Reynolds' Number or *R*-Series

The intensity of competition among centrifugal-pump manufacturers today makes the strict use of type multiplication diagrams prohibitive except for the purpose of preliminary comparison through a short range of multiplying factors in the series. A complete standardization based on the law of similarity would result in a series of standard pumps very likely to be satisfactory between certain limits and far short of the best possible above and below these limits. The study of this situation is being gradually advanced by different designers, and valuable knowledge is gained by occasional disappointments and the means that are found effective in their remedy. When this is completed, one important further step in the knowledge of pump standardization will have been reached. The principles outlined by the author remain important as a means for preliminary comparison and first approximation.

[6] Hydraulic Engineer, Byron Jackson Pump Mfg. Co., Berkeley, Calif. Mem. A.S.M.E.

The author's suggestion of the use of preferred numbers in the choice of multiplying factors and ratios is a happy one. Every designer has, however, developed certain preferred numbers of his own, based on actual experiences and arbitrary past selections, which have been fortuitously successful or which have some particular mental association. Many of these are not the result of psychological selection, but are fundamentally dependent on the natural laws of hydraulics, which in their application to power machines have a margin of possible variation. For instance, the ideal multiplying ratio for two consecutive machines of the same type should be such that a small-capacity impeller of the larger machine shall deliver only a slightly greater capacity than a high-capacity impeller in the smaller machine. This ratio has been determined experimentally, but is still possible of further modification.

The series developments used by the author and which plot as straight lines to logarithmic scales in the Cartesian diagrams illustrated by him are not really all straight lines in actual practice. For instance, the brake horsepower used by a power machine varies with size, not in accordance with the laws of similitude assumed by the author. Power losses in different-sized machines of the same type nearly always vary with multiplying factor as a transcendental function of the size of machine. This is true of steam turbines, centrifugal pumps, rotary compressors, etc., and such functions do not plot as straight lines to logarithmic scales. In addition, the combination of varying Reynolds-number indices in centrifugal pumps of different sizes has an effect on the unit capacity and unit speed which causes their deviation from the logarithmic straight-line rule. This variation is very marked in certain types and less marked in others. A knowledge of such effects is of the utmost importance to the designer in proportioning different sizes of machines of the same hydraulic type, not only in order to obtain the anticipated output and the best possible efficiency in each size, but also to enable him to so dimension the various elements of each size that the lowest possible manufacturing cost may be arrived at by economizing in space and material. There exists today fairly conclusive evidence that really large pumps can be built to substantially smaller dimensions than small pumps belonging to the same series, and by so doing their performance is improved. The larger the machine, the better it may be proportioned to hydraulic advantage—a second reason why in standardization work a multiplying factor should be regarded as a "type factor" which requires for maximum advantage important changes in dimensional proportions of the machine for each size, as long as every size of machine is required to operate with uniform physical characteristics of the fluid to be pumped.

Author's Closure

Deviations from the law of similitude were briefly treated in the conclusion of the paper and the attention of readers was called to certain points liable to raise questions. That the discussion of Mr. Macmeeken has been directed to these points should be welcomed, as obviously they need further elucidation.

If the Reynolds number be of such importance, as Mr. Macmeeken seems to believe, its effect could be easily taken into consideration.

Using the diameter D as representative of the size, the peripheral velocity U as representative of the flow velocity, and ν as the kinematic viscosity of the water, the Reynolds number R is proportional to DU/ν, or $R = \text{const.} \times DU/\nu$.

The kinematic viscosity being constant for the case considered, this relationship permits us to form the R-series in the usual way. Proceeding through each term of H_h-series along $H = \text{const.}$ and $U = \text{const.}$, we pass through the corresponding successive terms of the R-series and D-series. We have

$$\frac{R_n}{R_{n-1}} = \frac{D_n U}{D_{n-1} U} = \frac{D_n}{D_{n-1}} = \lambda$$

The term ratio of the R-series is γ and its general term λ^n.

The location of the lines $R = \text{const.}$ is determined by relationship $R = \text{const.} \times DU/\nu$; obviously the lines of constant R pass through each term of the u-series (λ^{-1}) and each alternate term of the D-series (λ^2). The R-series is shown in Fig. 11. It indicates that the Reynolds number increases with the size and capacity of the pump, a fact which to a certain extent explains the better efficiency of the larger-size pumps.

Unfortunately we can not make more use of the Reynolds number in the standardization of centrifugal pumps than to state its relation to the other variables and its effect on the pump efficiency. That the Reynolds number is not the controlling factor as regards pump efficiency follows from the fact that there are large pumps (large Reynolds' numbers) with low efficiency as well as small pumps with high efficiency.

The Reynolds number has not been taken into consideration not only because the similitude, simultaneously geometric and dynamic, cannot exist for motion in a *viscous* liquid, but chiefly because in the application to turbo-machines the friction forces are negligible as compared with the mass-acceleration (inertia) forces.

In addition, the friction losses, along with the other hydraulic losses (shock, conversion, etc.), were found to follow very closely the assumption involved in the relationships used, namely, that the losses are proportional to the square of the flow velocities, or heads. But in the law of similitude as applied to centrifugal pumps, further assumptions are involved, i.e., regular flow, uniform distribution of pressures and velocities, filled channels, no cavitation.

In endeavoring to design pumps of better efficiency, we certainly approach the assumptions required by the law of similitude. Fig. 12 shows results of tests of a pump running at

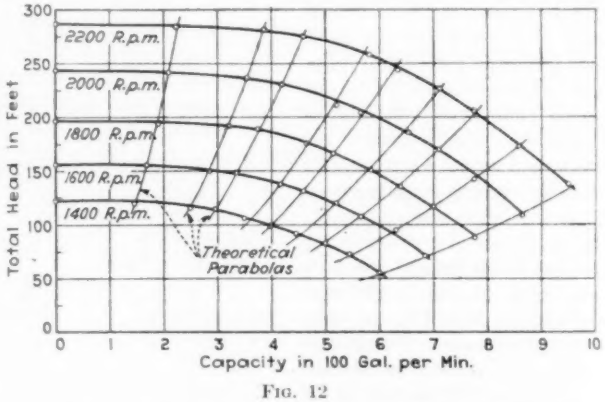


Fig. 12

different speeds and at constant openings of the discharge valve. The high efficiency (best, 86 per cent) of a comparatively small pump and the almost exact location of points of the N_s-series on the mathematically determined parabolas are not a mere coincidence, but seem to indicate a closer relationship between the high efficiency and the fulfilment of the assumptions of the law of similitude.

Using the standardization method presented in the paper, it is possible to handle any information about the pump in question, and to evaluate any new factor which may appear and take it into consideration when selecting the range of the D-, N-, and n_s-series to be used for a given service (see Chap. V on the n_s-series).

The selection of the n_s-series is especially connected with

the matter of efficiency, and here is where the designer can utilize all information like that given in Mr. Macmeeken's discussion. As a means of presenting all the variables with all their interrelations, the standardization method may also be helpful in this stage of the work, but its task is, as a tool, to develop and incorporate in the standardization chart all the series involved, with their multiple interrelations, utilizing any information which is of value for the purpose of standardization. The method can do that, using not only the series already developed, but also adding, if necessary, series of new variables, which it may appear desirable to evaluate, as was done at the beginning of the closure with the Reynolds numbers.

Application of Hydraulic Laboratory Researches

By IREAL A. WINTER,[1] BIRMINGHAM, ALA.

This paper presents results of laboratory investigations made in connection with hydroelectric power-plant designs, the problems encountered leading up to the establishing of a policy of conducting research, the method of testing, and comparisons of laboratory and field results. The object is to illustrate the many advantages of model investigations and to point out the deficiency of scientific knowledge incidental to power-plant design, with the hope that interest will be stimulated in pure-science research. Hydraulic research may be classified under two subdivisions: complex flows created by natural active and reactive forces, such as river channels, harbors, estuaries, tidal effects, etc., and, second, manufactured structures, such as spillways, canals, penstocks, hydraulic machines, etc., where the active and reactive forces should be definitely predicted by the designer. This paper deals with the latter, principally in connection with hydraulic turbines and their associated parts. Results will be shown where slight changes in shapes raise or lower the efficiency of hydraulic machinery as great as 1 per cent. No attempt is made to specifically account for the results obtained, but the apparent inconsistency of performance of water passages designed according to the limited knowledge of positive and negative acceleration will be evident to hydraulic engineers engaged in the utility field.



THE keen appreciation for a need of engineering facts in the preparation of designs for hydraulic structures has given impetus to research in this country to a great extent in the past few years. The self-sufficient attitude of the engineer of the old school has been supplanted by a desire to prepare designs in accordance with results established by laboratory tests on homologous models. This attitude has been brought about mainly by economic necessity, supplemented by engineering difficulties due to a deficiency in materials.

Many engineers are aware of the fact that new and improved methods are being fashioned in the laboratory, challenging the existence of every industry. A casual observation of the rapid change in importance of various industries today is sufficient acknowledgment of these facts. No group of utilities has been more aware of these conditions than the hydroelectric group. This new and energetic industry has grown to great economic importance by untiring efforts in developing its properties and markets through research, its very conception being a laboratory product.

The Mitchell Dam Plant

When designs were being prepared for the Mitchell Dam development of the Alabama Power Company, several engineering problems became evident as the work progressed.

The topography at the dam site limited the space available for spillway capacity when using gates of a depth considered safe for spilling on the river-bed foundations. The rock bluffs forming the abutments presented unsuitable conditions for a hillside spillway. The channel immediately below the site caused a rise in backwater of over 25 ft. during flood periods. As the normal operating head was only 68 ft., the loss in effective pressure at times of freshets, a frequent occurrence on the Coosa River, amounted to an appreciable loss in turbine capacity.

These conditions brought to maturity the conception of the Thurlow backwater suppressor.[2] This type of plant utilizes the hydraulic jump to lower the static pressure over the draft-tube orifice, thereby maintaining full pressure head on the turbine at all times. It also enables the plant to be laid out so that the full width of the river can be used as a spillway. This is accomplished by placing the units upstream of the spillway and discharging the draft tube through the main dam. Fig. 1 illustrates the general arrangement of the plant. Many secondary advantages resulted from the design, and it is considered by the operators as one of the most satisfactory generating stations on the system.

The performance of the backwater suppressor has been as predicted from experimental research. The average increase in kilowatt-hours per year has been substantially 4 per cent. This is the amount of energy actually sold, and is not the available generation had the load been equal to the plant capacity. The possible increase is 7.5 per cent without the influence of the Jordan Dam backing tailwater over the draft tubes. It is expected that a substantial increase over the present generation will be realized when the full efficiency of the spillway water is used in neutralizing the effect of a rising forebay at Jordan Dam.

Of equal importance to increase in yearly generation is the ability of the plant to carry full load at all times.[3] The maximum increase in plant capacity has amounted to 45 per cent, which may not account for a substantial increase in kilowatt-hours, but as a means of reducing capital investment, is of great moment.

---

[1] Hydraulic Engineer, Allied Engineers, Inc. The author was born on July 31, 1894, in Belvidere, Tenn. His engineering activities began in 1912 in the drawing room of the By-Product Coke Oven Plant of the Tennessee Coal, Iron and Railroad Company at Fairfield, Ala. From 1913 to 1914 he was engaged with the Alabama Power Company on the design of hydroelectric stations, resigning in the fall of 1914 to accept a position with Ford, Bacon & Davis at New Orleans, to be engaged in the design of cotton warehouses and grain elevators. In the summer of 1915 he returned to Birmingham with the Alabama Power Company on design of steam and hydroelectric generating plants. During the World War he served with the 37th Engineers, U.S.N.A., with the grade of Master Engineer. From 1919 to date he has at various times been connected with the Dixie Construction Company, Allied Engineers, Inc., and Corps of Engineers, U.S.A., on general design and laboratory investigation of hydraulic machinery and structures.

Contributed by the Hydraulic Division and presented at the Semi-Annual Meeting, Birmingham, Ala., April 20 to 23, 1931, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] Trans. A.S.M.E., vol. 44, p. 361.

[3] N.E.L.A. publication No. 267–10.

All features of this new type of plant were developed by models to a scale of $^1/_{24}$ and $^1/_{10}$ full size. No laboratory facilities were available in the Birmingham district; consequently the work had to be done on a very modest scale. The first working model was demonstrated in a flume costing less than one hundred dollars. The results were so convincing that the management appropriated a substantial sum for carrying the investigation to a conclusion.

Since 1921 all major developments of the Alabama Power Company have been the subject of laboratory research covering structural, electrical, and hydraulic problems. The field of hydraulics has been investigated in connection with spillways, spillway gates, energy dissipators, penstocks, scroll cases, draft tubes, turbines, expansion joints, cofferdams, backwater curves, etc. The results of laboratory tests have been checked in the field where a check has been feasible.

Draft-Tube Tests

When Mitchell Dam was under construction, the new improved concentric type of draft tubes was coming into general use, and it was thought desirable to investigate tubes best suited for the suppressor type of plant.

A series of experiments were conducted at the Alden Hydraulic Laboratory of the Worcester Polytechnical Institute, using a 12-in. homologous runner and various types of draft tubes.[4] The result of these tests was the adaptation of two units using the White hydraucone[5] for good efficiency at all loads and two units of the elbow type modified to use the long downstream leg incidental to the suppressor design. The latter type offered considerable saving in construction cost and equally good peak efficiency, being well adapted to base-load generation.



Fig. 1 View of Mitchell Dam Plant Showing Three Suppressor Gates Open
(Flow is 15,000 cu. ft. per sec.)

A comparison of the laboratory and field results is shown in Fig. 2. No efficiency tests were made on the plant due to the difficulty of measuring the water quantity. The kilowatt output against the gate opening in per cent is used as the criterion for performance. It will be noted that the two machines have identical capacity for the same per cent of gate opening from 16,000 to 18,000 kw. This is in substantial agreement with the model tests indicating that the efficiency of the two units should be the same between 0.20 and 0.235 hp. at 1 ft. head for $D_2 = 12$ in.

The importance of draft-tube design to construction costs and turbine efficiency was well illustrated by the Worcester tests In this same series a model of the old-style elbow-type draft tube used at Lay Dam (lock 12)[6] was built and tested. The interesting results of this work are shown in Fig. 3.

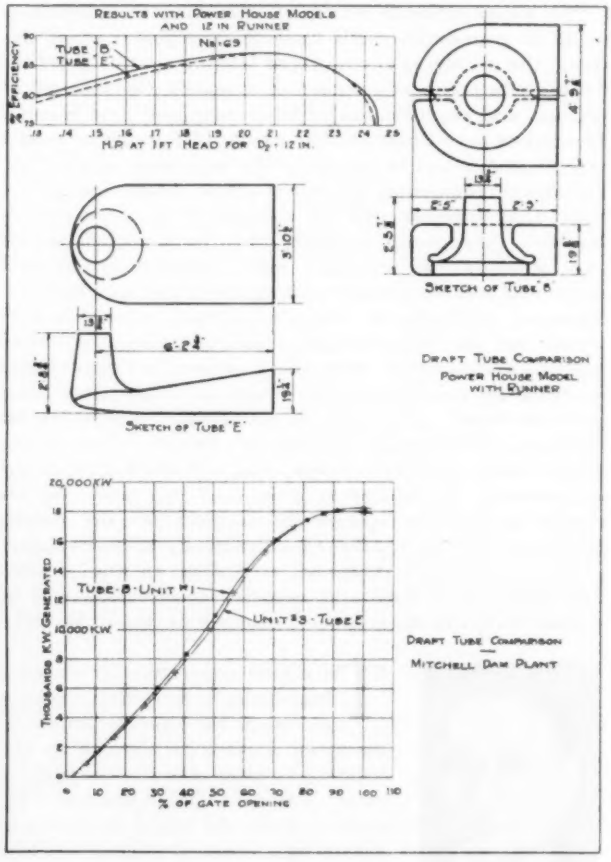


Fig. 2 Comparison Curves for Mitchell Dam Draft Tubes

Comparisons of tests from several angles are given in this figure. Starting with the experiments on a 4-in. model without a runner, the general characteristics for various angles of whirl are shown. Contrary to the general belief that an elbow tube is poorer in efficiency with an increase in angle of whirl, the reverse is true in this case. A draft-tube efficiency of 33 per cent is obtained at 8-deg. angle of whirl and increase to a maximum of 52 per cent at 40 deg. By plotting the area curve of the vertical barrel with sections taken in a horizontal plane, rather than normal to the center line of the tube, the reason for this apparent inconsistency is seen. The rate of expansion is much greater than can be obtained without considerable centrifugal force to effectually fill the tube. Alteration D, Fig. 3, is a semicylindrical fillet blocking off 40 per cent of the outflow area. The efficiency at zero angle of whirl is increased 10 per cent to a value of 46, and to a value of 63 per cent at 38-deg. angle of whirl. A comparison of the rate of diffusion normal to the center line for the two tubes will show that the alteration more nearly approves the possible rate determined experimentally

[4] Trans. A.S.C.E., vol. 87, 1924, p. 893.
[5] Trans. A.S.M.E., vol. 43, 1927, p. 255.
[6] Trans. A.S.C.E., vol. 58, 1915, p. 1505.

for a conical diffuser than the original design which was based on negative acceleration obeying the law of positive acceleration.

The same characteristics for the alteration are shown when tested with a runner, the peak efficiency being raised a full 1 per cent and the maximum power 3.5 per cent.

The performance of the model runner using the Mitchell Dam hydraucone is shown for comparison. It was thought quite significant that a simple alteration in the old-style elbow tube would raise the efficiency of the runner to within 0.5 per cent of the value obtained with the improved hydraucone. Results of research to develop the quarter-turn tube are included in the paper.

The field alteration of the draft tube at Lay Dam apparently

It was assumed that the increase in capacity was due to an increase in negative head within the draft tube. The Holyoke test on the model runner was used as the criterion for change in efficiency, all points being based on a percentage of power in both cases. A new gross head was computed that would produce the equivalent increase assuming the power to vary as $H^{3}/_{2}$. A new quantity of water was computed on the basis of the $\sqrt{H}$. The new quantity was substituted in the efficiency equation, using the actual head and kilowatt output. The resulting curve is shown in the figure. It is believed that this method of determining the effect of alteration in the field is legitimate, although it has been questioned by several engineers. The author has had unfortunate experiences in attempting to check field results

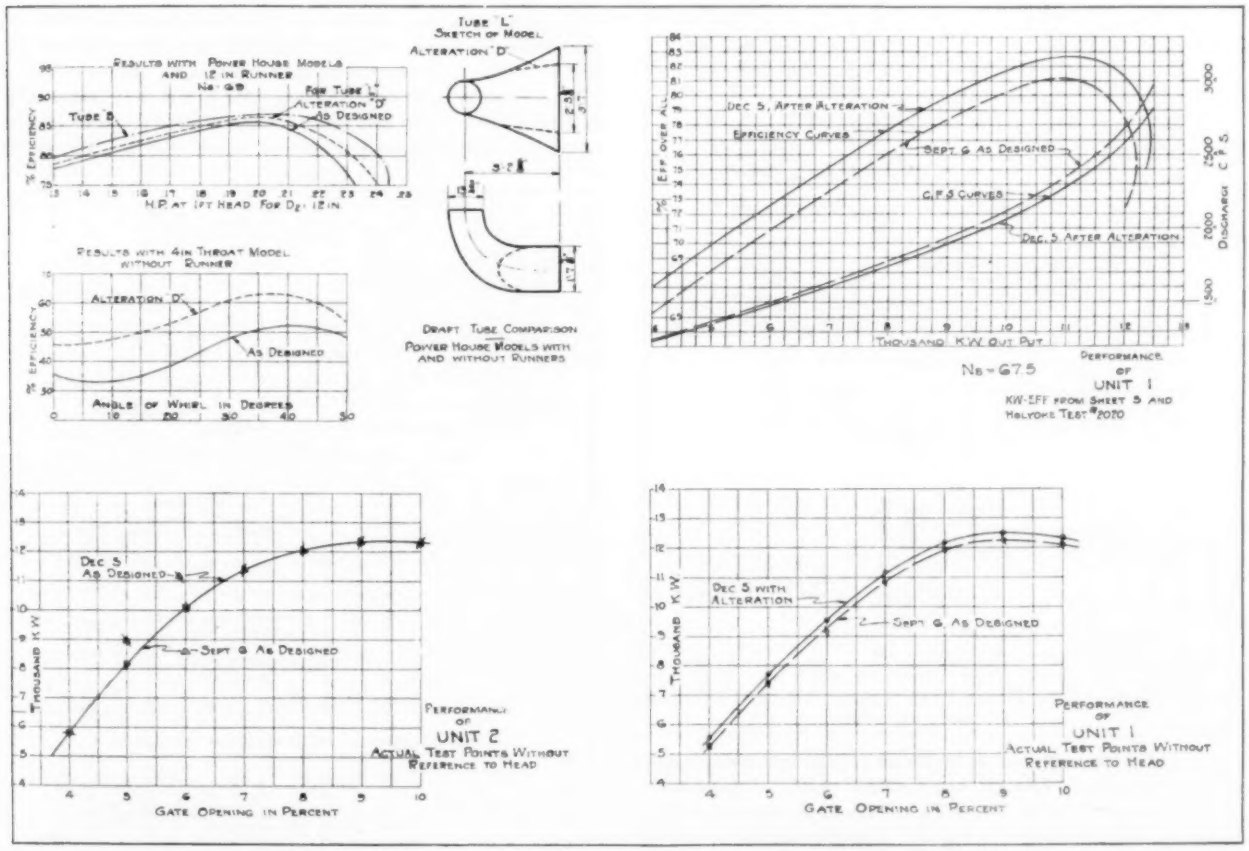


Fig. 3 Comparison Curves for Draft-Tube Alteration at Lay Dam

agrees with the results obtained in the laboratory. The peak efficiency was raised 1.4 per cent and the maximum power 2 per cent. The power output was used to arrive at these figures.

A test of capacity against gate opening was made on units 1 and 2. The head was kept constant for all readings. The switchboard instruments were used to determine the output and were sealed to prevent being altered before a check test could be made on the units after the draft tube change. Unit 2 was used as a pilot unit, the draft-tube alteration being made on unit 1.

The curve sheets show the agreement of the tests before and after alteration. Unit 2 shows consistent agreement of power for gate opening for both tests. These readings were remarkable inasmuch as the actual readings checked as close as 10 kw. in 12,000. Unit 1 showed a constant increase in kilowatts for all gates, the curves being parallel.

with portable instruments. The best criterion for changes in turbine performance in the case just cited is the piston stroke of the guide-vane servomotor in inches and the revolution of the kilowatt-hour meter disk, with due respect for accurate head readings and the relation of piston stroke to actual gate opening in inches.

Martin Dam Tube

The general idea of the modified elbow tube developed for units 3 and 4 at Mitchell Dam was incorporated in designs for the Martin Dam plant. The units at this plant are rated at 45,000 hp. at 145-ft. head. Due to the relatively lower specific speed of the wheels, it was thought that draft-tube efficiency was not as essential as in the first cases cited. The designs were prepared to meet structural conditions to the best advantage.

A series of laboratory tests were made to determine the hy-

draulic performance of tube *N* and *Na* as shown in Fig. 4. Tube *F* is used as a pilot tube for comparison. It will be noted that tube *N* gave a peak efficiency equal to tube *F*, but at other loads the performance was decidedly inferior. From paint experiments made to determine the flow lines within the tube, it was evident that at axial flow an eddy developed in the heel of the tube causing a reversal of flow up into the vertical barrel. A half-cone fillet was placed in the heel which proved effective in removing the reversal flow and improved the performance of the runner to a marked degree as shown by curve *Na*.

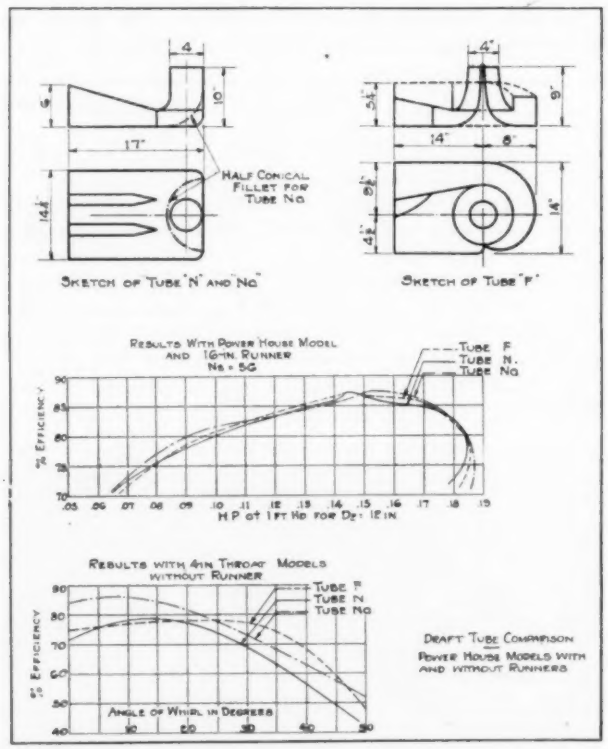


Fig. 4 Martin Dam Draft-Tube Tests; Models With and Without Runner

The importance of draft-tube design for the low-specific-speed runner was clearly brought out in these tests. A comparison of performance of tubes *N* and *Na* at various heads using a 13½-in. model runner is shown in Fig. 5. The effect of the alterations is interesting, as they indicate the necessity of making a laboratory test to determine the effect of any change that might be made to suit field conditions.

Jordan Dam Tubes

The similarity of the cross-section of tube *Na* and the results obtained on the draft-tube alteration at Lay Dam suggested a reversion of type to the old quarter-turn tube. A series of experiments were made to determine the characteristics of the true elbow type. The method adopted was purely experimental. The limiting dimensions were taken from tube *F*, and the outer radius was made to equal the depth of the draft tube. The inner radius and roof line were estimated from designs of earlier types.

The first results showed poor performance as was expected. Paint experiments indicated that the trouble was due to eddies and reversal flows around the inner radius and along the roof of the horizontal diffuser. The regions of ineffective areas were filled in, and the paint test was repeated. This work was carried on in a manner similar to a mechanic spotting a bearing. As the dead area were eliminated, the efficiency improved, until a point was reached where the elbow was filled at all angles of whirl. The final design is shown as tube 1810 in Fig. 6.

The closeness of performance of tubes *F*, *N*, and 1810 showed that the three general types represented in this series were all capable of delivering good efficiency, with the advantages in favor of the elbow tube.

Inconsistent Performance of Tubes

In general, it can be stated that a good draft tube for one wheel will also prove efficient for another, but to risk the design of a major installation without laboratory investigation is not justifiable. The following illustrations are offered:

From tests shown in Fig. 4, it would appear that tube *Na* would be the better tube for comparison with 1810 and *F* in Fig. 6. This test was made, with the results shown in Fig. 7. The increase in draft-tube efficiency due to the installation of the half-cone fillet in the heel netted a loss of 0.75 per cent in peak efficiency, but registered a gain of 8 per cent in horsepower.

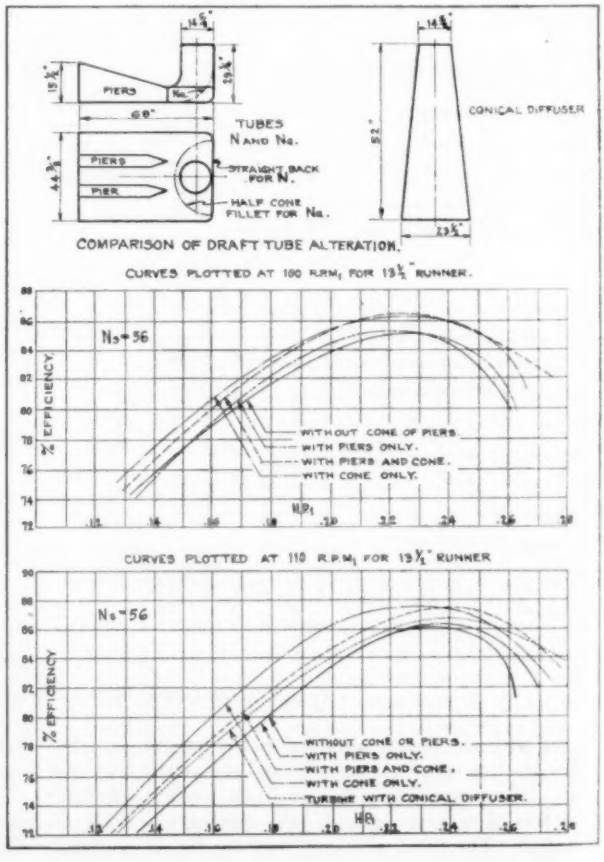


Fig. 5 Experiments With Alteration on Martin Dam Type of Draft Tube

The additional regain in velocity head pulled more water through the runner, but failed to improve the performance at point of best efficiency. The author has no explanation for this fact, as other test data indicate the same inconsistency in a negative way.

Fig. 7 shows the draft-tube efficiency for tube 1810 and a 4-deg. conical diffuser. The conical diffuser has a superior efficiency of from 10 to 20 per cent throughout the range of 45 deg. of

whirl, but no increase in turbine efficiency was found when tested with a 16-in. model runner, although the horsepower did move out a small percentage. The second curve in this figure is a little more encouraging, as the peak efficiencies move up and out as the draft-tube performance improves.

The upper and the second curves in this figure were obtained with runners of different designs, but which were of the same specific speed.

General

With the exception of the cases cited in Figs. 5 and 7, it has been the author's experience that the tube giving the greatest horsepower also gives the best performance curve. This is only stating that the tube with the best regain of velocity head will perform better with a runner.

The illustrations showing the results obtained with 4-in. models without a runner give considerable prestige to this method of testing.

It has been generally understood that the 4-in. models were usable only for rough approximation. This idea is no doubt the result of the method of testing and the failure to appre-

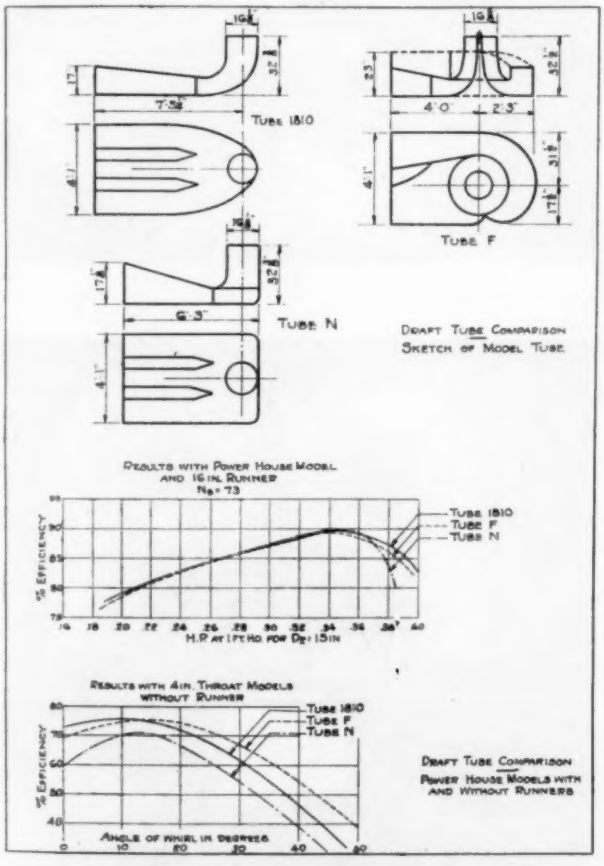


Fig. 6 Jordan Dam Draft-Tube Experiments

ciate the necessity of making a suitable flume where accurate observations could be made.

The power-house type of tubes being designed today leaves very little to be desired in the way of efficiency if we are to be guided by results shown in Figs. 5 and 7. This suggests the idea that the runners are deficient in delivering the water to the diffuser in order that the superior ability of this type of water passage can be utilized.

Scroll-Case Experiments Without Runners

Extensive research in connection with scroll-case design was carried on in conjunction with the draft-tube investigations. The results obtained have indicated that if full benefits are to be realized from tests, complete use of the laboratory facilities should be made. The later plants of the Alabama Power Company have been designed from data obtained by tests on homologous turbines, speed rings, draft tubes, scroll cases, and pen-

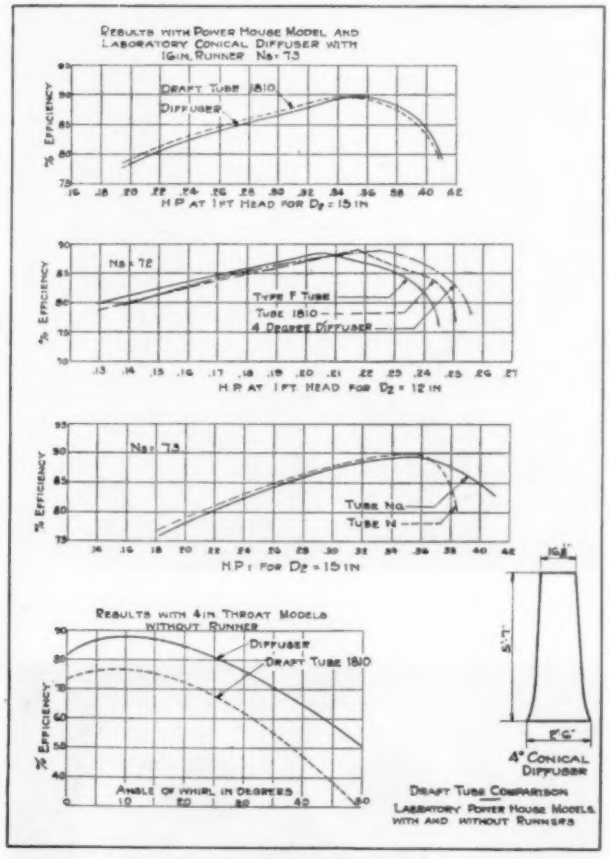


Fig. 7 Comparison Curves for Martin Dam and Jordan Dam Draft Tubes

stocks. The plant has been set up in the laboratory as completely as possible. New runners, speed rings, and guide vanes have been cast to reproduce the actual plant conditions. This is essential if correct values are to be obtained regarding speed, horsepower, and expected efficiency.

The first research in this connection was on the Martin Dam plate-steel scrolls and penstock. This work was done in the Birmingham laboratory on 4-in. models without a runner. The difficulty of obtaining usable data by such a procedure was fully appreciated, but in view of the remarkable performance of the 4-in. draft-tube models, a try seemed justifiable.

Three scroll cases with speed rings to match were built. The turbine proper was complete in detail, except that the buckets on the runner were omitted. The turbine gates were fixed at the point of best efficiency for all tests, the only variables being the scroll case and speed ring. The coefficient of discharge based on the throat discharge area of the runner D_2 was used as the criterion of performance. The flow line within the case and turbine was determined by paint tests.

The installation of the model in the laboratory is shown in

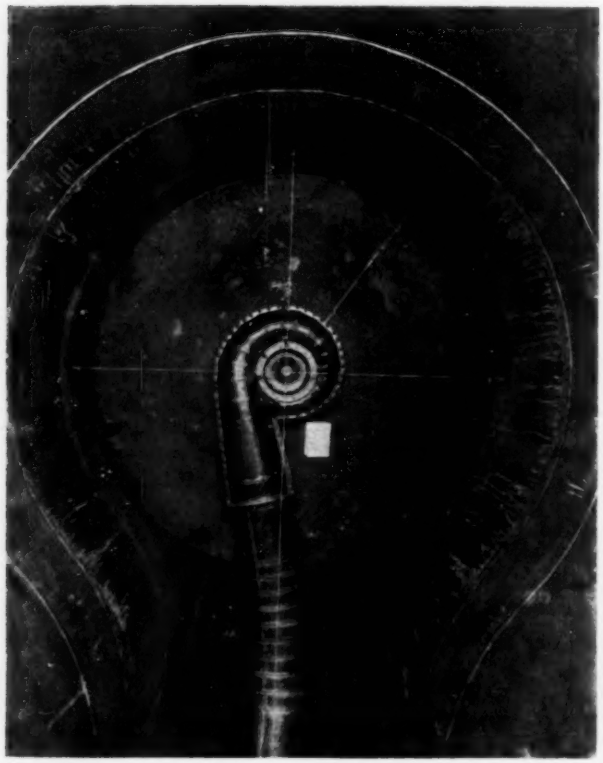


FIG. 8 FLUME ARRANGEMENT FOR EXPERIMENTS ON 4-IN. MODELS OF PLATE-STEEL SCROLL CASES

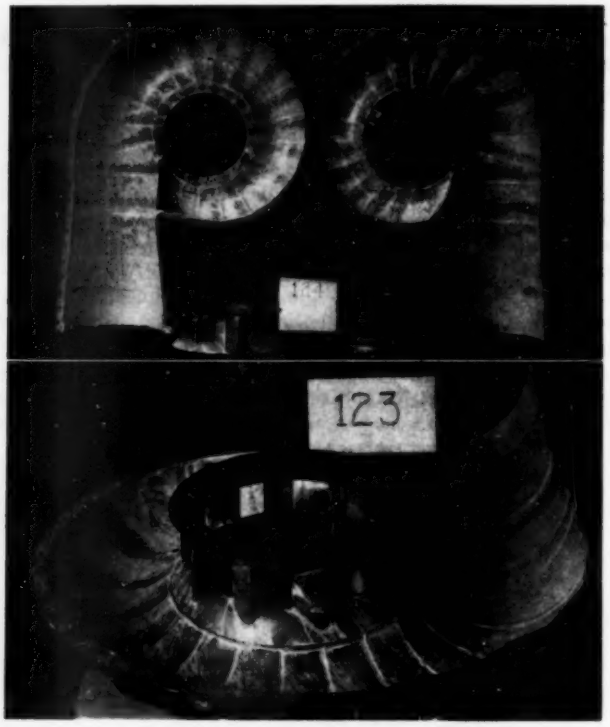


FIG. 10 FLOW LINES WITHIN SECOND 4-IN. MODEL SCROLL CASE

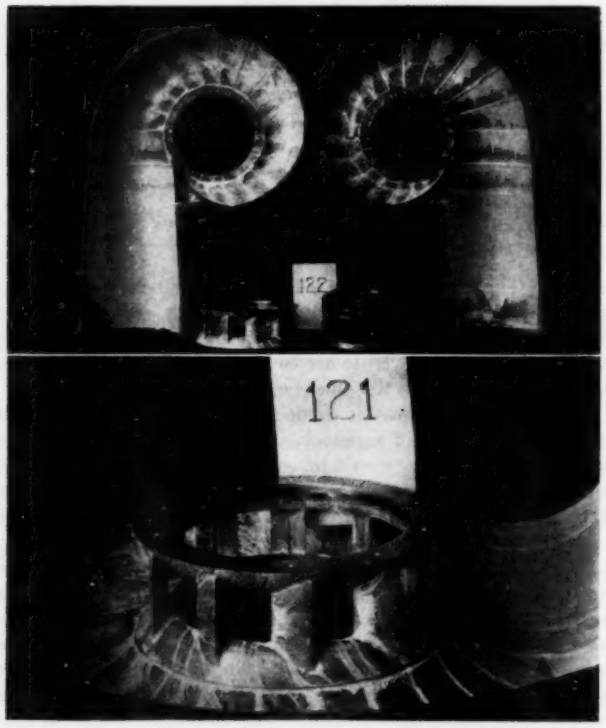


FIG. 9 FLOW LINES WITHIN FIRST 4-IN. MODEL SCROLL CASE

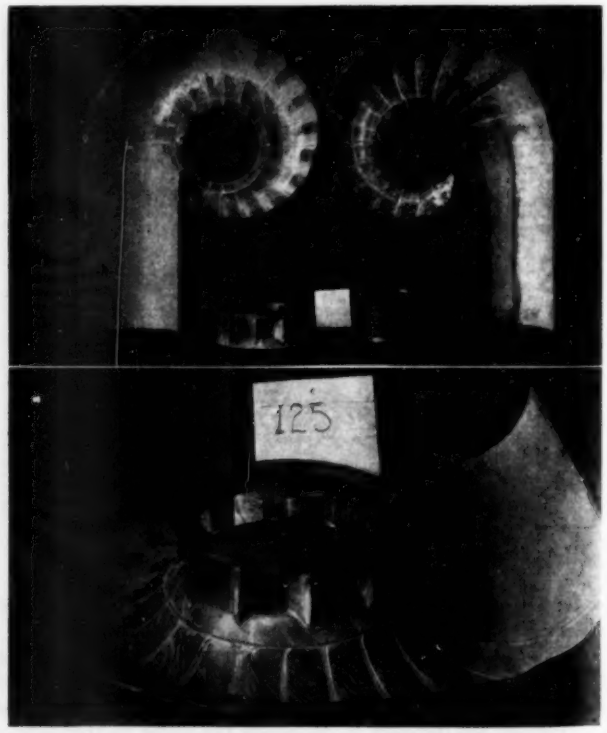


FIG. 11 FLOW LINES WITHIN THIRD 4-IN. MODEL SCROLL CASE

Fig. 8. The flow line for the first scroll designed with increasing velocity around the turbine is shown in Fig. 9. It is interesting to note that the water entering the turbine adjacent to the speed-ring surfaces is flowing radially inward while the water at the center of the distributor is flowing more tangentially and in line with the turbine gates. The failure of the water to follow the speed-ring stay vane is shown by the eddies developing on the turbine edge of the vane.

The second case, shown in Fig. 10, is designed for constant velocities around the turbine, the initial velocity being determined arbitrarily as a percentage of the spouting velocity of the water. The same general characteristics of flow into the turbine are found as in the first case. The speed-ring stay vanes are slightly better in line with the water currents.

The third case was designed along different lines from either of the first two cases. The moment of momentum was deter-

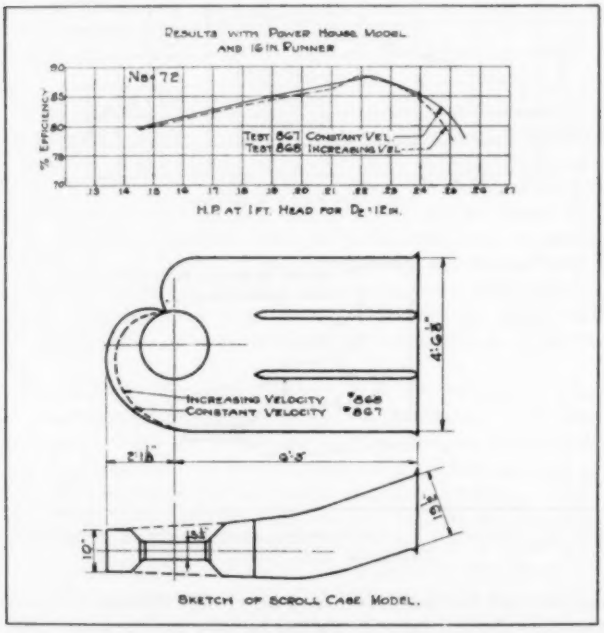


Fig. 12 Scroll-Case Tests Showing Effect of Velocities Within the Scroll

mined for the increment of flow passing the discharge tips of the turbine guide vanes when at the point of best wheel efficiency. All sections of the scroll were then made to conform to the full moment of momentum law. This gave a definite value for all dimensions of the scroll as well as establishing the initial scroll velocity. The speed-ring stay vanes were placed more radially than in the first two cases. These vanes were duplicates of the turbine guide vanes and placed substantially in line with the guide vanes at the point of best runner efficiency. Fig. 11 shows the flow characteristic for this case. The radially inward flow still persists on the upper and lower speed-ring surfaces, but the stay vanes are more in line with the water stream than in either of the first two cases. It will be noted that the eddy has moved to the entering edge of the stay vane, indicating that a slightly less angle would have given better results.

The influence of a 100-ft. radius elbow in a 20-ft. penstock immediately above the scroll intake was included in these experiments. The angle of the elbow was 32 deg. The intake to the penstock was bell-shaped to eliminate the influence of the usual design of penstock entrance. An equivalent length of straight penstock was built as a guide to the losses within the elbow. The tests indicated that the elbow lowered the coefficient of discharge a slight amount. The flow lines in the scroll case did not change, indicating that the elbow did not set up an appreciable roll affecting to any great extent the distribution of the water to the turbine.

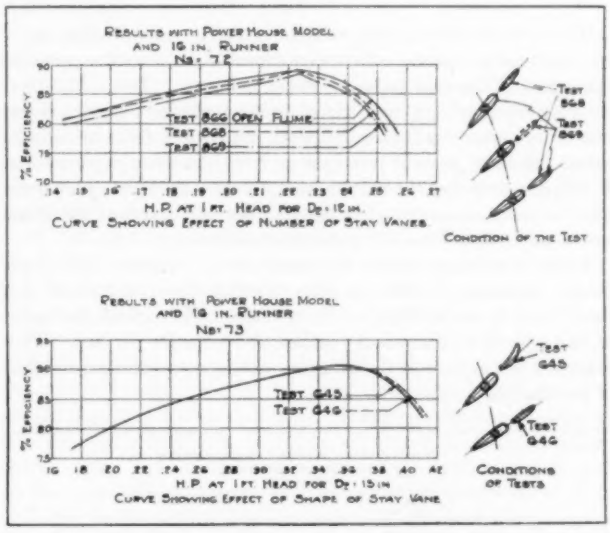


Fig. 13 Comparison Curves Showing Effect of Types of Stay Vanes

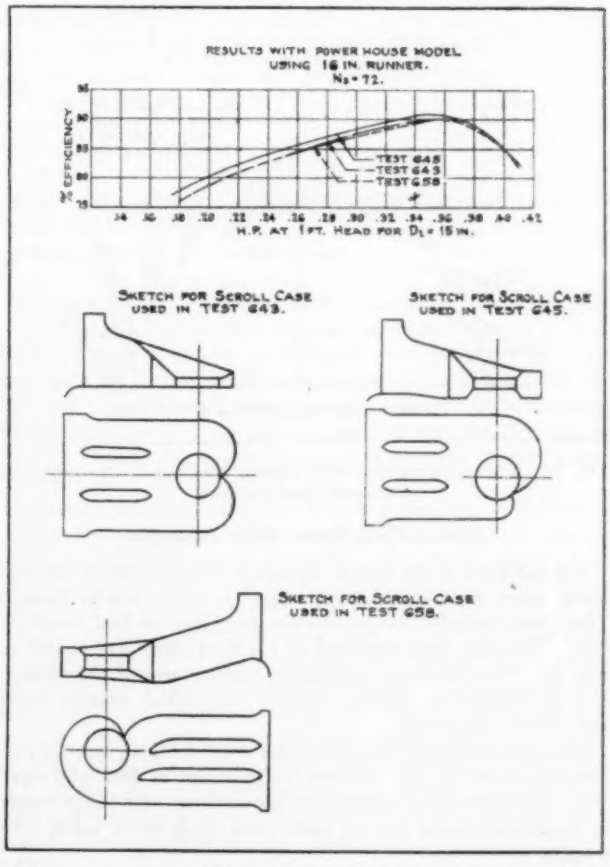


Fig. 14 Comparison Curve Showing Results Obtained With Various Designs of Scrolls

A comparison of the test results for the various conditions given is as follows:

Coefficient of discharge based on runner D_2:
Case 1 = 0.3178 with elbow in penstock
Case 2 = 0.3180 with elbow in penstock
Case 3 = 0.3190 with elbow in penstock
Case 1 = 0.3188 with straight penstock.

These tests were not as conclusive as the experiments on 4-in. draft-tube models. This was expected, due to the opposite functions of the two types of water passages. The author has no definite records of tests on plate-steel scrolls using a runner, but he is under the impression that subsequent tests have indicated the same general relations as found in these experiments. It will be noted that the maximum difference in efficiency between any two cases amounts to less than 0.5 per cent and that the elbow accounts for less than 0.1 per cent in efficiency.

These small models were to a scale of $^1/_{12}$ full size. The indicated discharge based on the coefficients determined by model test is about 3800 cu. ft. per sec The actual discharge of the wheels at the plant reached 3860 cu. ft. per sec. This is a good criterion as to the accuracy of the tests and the precision of the models.

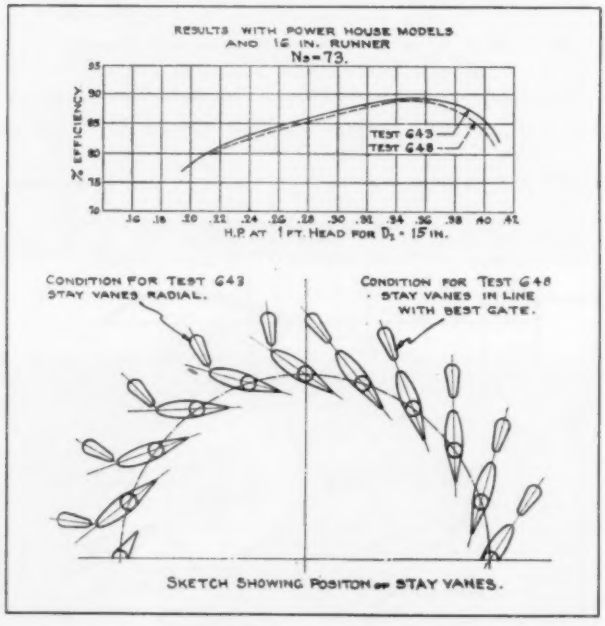


FIG. 15 TEST CURVES SHOWING EFFECT OF STAY-VANE ANGLE ON TURBINE PERFORMANCE

SCROLL-CASE TESTS WITH RUNNERS

The influence of the rate of change of velocity within concrete scroll cases was investigated, using a 16-in. model runner. These tests included the penstock as proposed for field construction. The test data obtained in the first series are shown in Fig. 12. Test 868 is for increasing velocity around the runner, and test 867 is for constant velocity, the initial velocity being the same in both cases.

The peak efficiencies for both tests were identical, but the performance curve for the constant velocity case is decidedly superior. The increase in horsepower is consistent with the increaser in coefficient found on the plate-steel scroll tests, using 4-in. models.

The speed-ring stay vanes used in these experiments were of the guide-vane type. It was thought that the use of a speed-ring stay vane for every guide vane would improve the hydraulic efficiency as well as lighten the upper and lower distributor rings. This did not prove true, as is shown in Fig. 13. The use of 24 vanes instead of 12 lowered the peak efficiency 0.6 per cent, and the entire performance curve proved to be decidedly inferior.

The lower curve in the figure shows the effect of the shape of the speed-ring stay vane on turbine efficiency. The straight-sided vane slightly increased the maximum efficiency and horsepower, and is an economic factor in construction. This is consistent with results found in the paint test made in connection with the 4-in.-plate scroll-case tests.

A comparison of performance of the two general types of concrete scrolls and a third radical design is shown in Fig. 14. Design for test 643 was based on 4-in. models without a runner, assuming that the water should flow radially into the guide vanes. Test 645 is for a scroll with constant velocity, and test 658 is with increasing velocities. Here again the constant-velocity scroll proved the better design, the peak efficiency being raised 0.75 per cent and the general performance curve being improved. The empirical scroll, with a symmetrical discharge and radial speed-ring stay vanes, proved equal to the increasing velocity scroll in peak efficiency, with better performance at part gate.

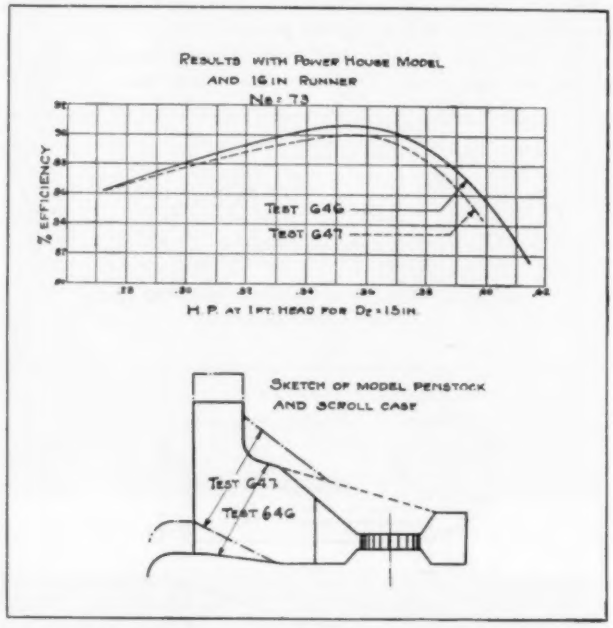


FIG. 16 PENSTOCK EXPERIMENT SHOWING EFFECT OF STEEP SLOPE ENTERING THE SCROLL CASE

An interesting sidelight on the consistency of test data is shown in Fig. 15. This design of scroll was determined in the flume used for testing 4-in. draft-tube models. The only limiting dimension given was the gross width and depth of penstock. A galvanized-iron form was placed around the turbine entrance to conform to the limits of the vertical walls of the penstock and scroll case. The scroll end was semi-circular and without a central baffle. Water was passed through the model to a depth corresponding to the roof of the penstock, and the resulting flow was noted. Two sheets of iron were bent to a shape approximating the outer surface of a scroll and placed in the flume while the test was under way. The two eddies forming about the downstream center line of the turbine entrance were greatly reduced by this procedure. The two sheets were then shaped by hand until all surface indications of ineffective areas were eliminated. The flume was then drained, and all metal parts were painted with white lead and oil. A test with a constant flow of water

was run for two hours. When the flume was drained, the flow lines showed with remarkable contrast, indicating the rate of reduction of area required in the scrolls. The limits of flow were calibrated from the iron sheets and transferred to a working drawing for the 16-in. turbine tests.

It was assumed that radial flow into the speed-ring stay vanes was desirable. Accordingly, a set of radial stay vanes was used in making the preliminary paint test. A set of corresponding vanes was built for the 16-in. runner. To check the effect of stay-vane angle on scroll design, a run was made with the symmetrical case, using the conventional vane angle of about 45 deg. Test curve 643 is for the condition of design, and curve 648 is for the conventional stay-vane angle. A drop of 1 per cent in efficiency resulted from the use of the sloping vane.

The influence of a sharp angle in the penstock immediately above the turbine entrance is shown in Fig. 16. This test was made using a 16-in. model turbine. The steep-shaped penstock for test 647 is not usual in power-plant design even with its penalty of 0.75 per cent on the plant's output.

Conclusions

Data presented in the paper are necessarily limited. The main object has been to point out the many potential mistakes in resorting to individual judgment without thorough laboratory tests. It is of equal importance to know what not to do as well as what to do. Throughout the paper it will be noted that many attempts have been made to use definite laws of motion to effect a proper design. The consistent failure of these attempts to produce the desired results is evident. A closer understanding of the application of fundamental principles to design will be realized when more scientific data are available to point the way.

The necessity of making field tests is of first importance if the full advantages of experimentation are to be realized. It is most gratifying to note the steady increase in field tests being made. Only by experience can we learn the art of measuring flows up to 7000 and 8000 sec-ft. with a fine degree of accuracy. Companies that have withheld their support because of the uncertainty of obtaining reliable results are now making plant tests and contributing their share to the general knowledge of the art. The measurement of the electrical output has not received the sincere attention that it should, no doubt due to the apparent ease with which these readings can be obtained. This is unfortunate, as many cases of conflicting results of overall efficiency measurements have resulted from indifferent output readings.

It is seldom possible to check the effect of various phases of turbine design in the field. The plant is tested, and the results obtained are for one condition only. This fact emphasizes the necessity for obtaining accurate absolute values if progress is to be noted by reference to other plants.

A review of the effect of altering parts of the water passages generally considered non-essential gives encouragement to the idea that substantial increases in efficiency and simplification of structures are to be had by continued effort along the lines of research.

The work outlined in the paper was done under the direction of Mr. O. G. Thurlow, Vice-President, Allied Engineers, Inc., with the author in responsible charge. Acknowledgment is made of the valuable assistance of Messrs. G. J. Hornsby, R. R. Randolph, Jr., Roy E. Davis, and E. B. Coleman, whose efforts were responsible for the precise execution of the work, and to Allis-Chalmers Manufacturing Company, I. P. Morris & De La Vergne, Inc., and S. Morgan Smith Company for their splendid cooperation in contributing their time and information to much of the data presented in the paper.

Discussion

R. V. Terry.[7] Experimental results obtained by the writer on 4-in. draft-tube models are in agreement with the author's in showing that, to a certain extent, the efficiency of most elbow draft tubes increases with whirl.

As a matter of record the writer would suggest that the author include a statement regarding how the efficiency of the 4-in. draft-tube models was determined; i.e., whether it is expressed as a regain in percentage of the throat velocity head or a percentage of the difference in throat and discharge velocity heads; also a statement regarding how the whirl was introduced and where the angle was taken. The angle of whirl of course varies across the throat of the tube.

The writer also has found some inconsistencies such as pointed out by the author. Slight errors are likely in the construction and testing of small models; also our simple theories do not completely cover the complex nature of water flow.

The writer has always used the constant-velocity method in designing turbine spiral casings. The method which increases the velocity toward the small end of the casing is based on the simple theory of free vortex flow. However, the flow conditions in a hydraulic turbine between the casing inlet and the runner are far from free vortex conditions. This space is subdivided by the speed-ring vanes and wicket gates. It is also quite probable that the runner indirectly affects the flow conditions in the casing to some extent. The writer considers that water flow in a spiral casing is more nearly analogous to flow in a pipe bend.

Certain previously published information is misleading in intimating that there is practically no loss of head in a well-designed spiral casing. The writer has found that there is appreciable loss of head in even well-proportioned casings and agrees with the author that casings should be the subject of laboratory investigation.

It is a little startling to find that radial speed-ring vanes give slightly improved results. This could never have been found by theory. The more one studies water flow the more he is convinced that it is far better to consult the water first, as the author has done, and then investigate a theory to correspond.

E. Brown[8] and L. B. Stirling.[8] The paper shows that tests of model turbines and their settings can be relied upon in forecasting the performance of full-sized homologous units. An increasing amount of confidence in such work will lead to the appropriation of funds for test purposes, and the results of studies undertaken over a wide field by various organizations should result in improvements in efficiency and reliability of hydroelectric units.

The Shawinigan Water and Power Company established a turbine testing plant at Shawinigan Falls a few years ago, in which model runners could be tested under heads varying from 55 ft. to 75 ft. or more, the tailrace level being adjustable over a wide range to facilitate studies of the effect of "setting" of the runner. It was thought that tests under such heads would bring to light unfavorable characteristics of runners which would not appear from tests under lower heads. The experience of many operating companies shows that modern high-speed runners, of both Francis and propeller types, exhibit a marked tendency to pitting, both on their under blades and on the draft-tube liners. A considerable amount of testing work has been done in studying the conditions under which such pitting occurs and in efforts to minimize it. These tests include the effect of

[7] Hydraulic Engineer, Newport News Shipbuilding and Dry Dock Company, Newport News, Va.

[8] In charge Experimental Turbine Plant, Shawinigan Water and Power Company, Shawinigan Falls, Quebec, Canada.

form of draft tube, and studies of the power-efficiency characteristics of propeller runners with differently shaped blades, and the influence of the setting of the runners on these characteristics. Many other allied problems have also been investigated.

Like the author, the writers have found the use of paint to be most helpful in certain kinds of work. To study the conditions resulting in pitting in the draft tube, the surface of the draft tube was painted with red or yellow duco and overlaid with a coat of soft white paint. After the paint had dried for about 12 hours, the turbine was operated under some predetermined condition for half an hour. The extent to which the painted surface was affected gave a comparative measure of the tendency of the conditions to cause pitting. In some cases no changes occurred; in others only the paint was removed; while in others both paint and duco were removed. In some cases the pitting was very severe and localized, and in other cases was scattered lightly over large areas. By such tests on a propeller runner it was established that serious pitting occurred over a comparatively small range of gate opening and that a very small change in the shape of the trailing edge of the blades made considerable change in the intensity of the pitting. Some unexpected results have been recorded, as in the author's experiences. Thus, for example, severe pitting occurred at a certain gate opening and speed, under 60-ft. head. The tailrace was lowered 5 ft., and the pitting was almost nil, other conditions being unaltered. A similar experience occurred with two service units on the system. The unit in which the runner appeared to have a safe setting, as judged by general principles of hydraulics, showed considerable pitting. The other runner, which was installed to replace one which had shown mechanical weakness and inefficiency, had a poor setting as judged by the same standards. It shows no pitting after two years of operation. The two runners are not of the same size, but their specific speed is the same, they operate under practically the same head, and they are similar in design.

During the summer of last year an extended series of tests was made in conjunction with the manufacturers who supplied a number of models of propeller runners, so designed that the inclination of the blades could be altered, thus providing for a number of variations at reasonable cost. The settings were altered by variation of tailrace level, and the performance of different models was thus related to the cavitation factor. After selecting the form and inclination of blade which gave the most desirable power-efficiency characteristic, the model was tested for resistance to pitting by using the paint method described. The model runner thus selected was used as the basis of design for the fifth unit of 30,000 hp. recently installed at LaGabelle, where the normal operating head is 60 ft.

Tests made some years ago on the effect of cones of different heights in the draft tube proved that no one cone showed to advantage over the others at all gate openings, when tested over a wide range of speeds. For small gate openings the smallest cone gave best results at all speeds, whereas at the larger gate openings the highest cone had the advantage up to certain speeds, beyond which the advantage lay with a cone of intermediate height. These results are mentioned briefly, to show that small changes have distinctly measurable effects and that it is sometimes dangerous to generalize from the experience of individual tests. The explanation of results is sometimes difficult or impossible, but the facts are of great importance and only appear as the result of tests.

RALPH L. THOMAS.[9] It is particularly notable that the author and his company have had the courage to apply the results of their model tests in actual installations, involving expenditures of large sums of money, sometimes in the face of the skepticism usually met in pioneering work, and that their expectations have been amply fulfilled.

There is one feature of the design of the hydraulic turbine water passages, as shown by the drawings with the paper, concerning which it would be interesting to have additional information. Was the shape of the bell mouth of the turbine intakes as shown in Figs. 14 and 16 the result of model tests? In particular, has the author made an investigation of the proper or most efficient curvature for the top of this bell-mouth entrance?

CARROLL F. MERRIAM.[10] The paper deals with an important phase of a subject which is being given serious consideration by power companies, particularly those constructing important hydroelectric plants. It is a well-established fact that these companies cannot afford to build modern plants without carrying through an extensive testing program comprising experiments such as those the author describes.

There is always present danger of accepting results which are based upon too little testing and of the establishment of engineering dogma which may be blindly followed in the future. For this reason experiments should aim to give clear understanding of the fundamental causes of hydraulic phenomena, rather than to determine simply what designs may under certain circumstances give the best results. These circumstances may be entirely upset by a slight alteration, and conclusions which might be accepted as general may prove on further investigation to be fallacious. The problem therefore demands the finest talent available, as well as a free interchange of information between investigators.

It is interesting to compare our practice with that in Europe, where progressive work in hydraulic-turbine design is being conducted practically entirely by the turbine manufacturers. In this country, although it is true that all of the leading turbine manufacturers have well-equipped laboratories, a great deal of work is being done by the laboratories of educational institutions and by the power companies themselves. The chief engineer for a foreign manufacturer recently remarked that he wished that his clients took the same interest in hydraulic experimentation that we do in America.

It would be very desirable, in order that laboratories may exchange equipment, to standardize upon a particular diameter of runner, and it is therefore urged that 16 in. should be adopted, inasmuch as the following laboratories are equipped to handle this size model: I. P. Morris & De La Vergne, Inc., Philadelphia, Pa.; S. Morgan Smith Company, York, Pa.; Newport News Shipbuilding and Dry Dock Company, Newport News, Va.; Shawinigan Water and Power Company, Shawinigan Falls, Quebec, Canada; Pennsylvania Water and Power Company, Holtwood, Pa.; Alden Hydraulic Laboratory of the Worcester Polytechnic Institute, Worcester, Mass. The economies effected by having it possible to test a runner at any of these laboratories can be appreciated.

In one place the author mentions measuring the electrical output by timing the disk of the integrating watt-hour meter. Excellent results may be secured in this manner, particularly for informal tests. Precautions, however, should be taken to see that these permanent instruments should not be affected in any way by outside causes, such as the change of burden on the transformers caused by the use of other instruments in the same secondary circuits, difference in power factor, etc.

G. W. SPAULDING.[11] The author has clearly shown the need

[9] Assistant General Superintendent, Pennsylvania Water and Power Company, Baltimore, Md. Mem. A.S.M.E.

[10] Pennsylvania Water and Power Company, Baltimore, Md. Mem. A.S.M.E.

[11] Assistant to General Superintendent, Pennsylvania Water and Power Company, Baltimore, Md.

for model testing when dealing with hydraulic design problems and by numerous examples indicates what improvements are possible by this means. It is generally recognized that the field of model testing, with either turbine models or structures, is by no means exhausted, and while each engineering project has its own peculiar local problems, it is plainly evident that a wealth of general knowledge could readily be available if the various agencies would pool their information obtained by model testing. Last year a hydraulic laboratory was built at Holtwood, Pa., to study the problems arising in connection with the Safe Harbor project. Much of the data and information obtained will be of considerable interest, and it is hoped that this information can be made known at a later date to those interested.

The results obtained by the author on his scroll-case tests without runners were exceedingly interesting and emphasized the fact that it is indeed difficult to predict the direction of the actual flow lines. On similar tests which were made in connection with Safe Harbor, it was found that the angle of the speed-ring vane was not critical and that this angle could be varied appreciably without affecting either the power or the efficiency. The angle actually used on the Safe Harbor turbines is approximately 25 deg. with the radius. The vanes are symmetrical about a center line, and it was decided to use one stay vane for each guide vane in order to reduce the cost of the speed rings.

A number of different tests were made with the final draft tube chosen for the Safe Harbor plant, to determine the effect of sloping the horizontal leg of the tube upward, to determine the most efficient rate of diffusion in the horizontal leg of the tube, and to determine the most economical horizontal length of the tube.

It would be interesting to obtain additional confirming data for a wide range of conditions including various types of draft tubes and with runners of various types and specific speeds, before agreeing with the author's statement that "in general, it can be stated that a good draft tube for one wheel will also prove efficient with another wheel." This is a matter which has been the subject of many discussions. It is hoped to make certain comparative tests along this line at the Holtwood Laboratory in the near future.

The author points out that the necessity of making field tests is of first importance, if the full advantages of experimentation are to be realized. It is planned at the Safe Harbor plant to use two methods of index testing, each of which will first be used on the homologous setting at the Holtwood Laboratory, and it is hoped that the coefficients obtained on the model can be used for the actual unit. One of these index methods is the Winter-Kennedy principle, for which the author was responsible. In addition to the index testing, actual efficiency tests will be made, using a large bank of current meters. The engineers of this company have spent considerable time and money in studying the question of field tests, and we are going to be very critical of the results obtained with these current meters. It is recognized that these field tests will be of primary value in analyzing the conclusions reached from the model tests.

LEROY M. DAVIS.[12] Not only is this paper valuable for its positive information, but it clearly shows the danger of designing an important hydraulic structure from the known laws of hydraulics without substantiating this design with suitable model tests. The importance of model testing, as a cheap and accurate means of determining the action of an unprecedented hydraulic structure, is just beginning to be appreciated by the American engineer. It seems probable that in the future the laboratory will be considered a necessary medium for arriving at an efficient design.

[12] Pennsylvania Water and Power Company, Baltimore, Md.

Model testing in the hydraulic field is not limited to draft tubes and intakes. For years model tests have been used extensively on turbine design. Recently tests have been made on spillways and on piers, to determine the breakwater effect of dams and cofferdams, and also the effect of excavation in rivers.

It would be interesting to know how the models which the author used were constructed. The Alden Hydraulic Laboratory has developed a method of building draft tubes which is especially well suited for models. A skeleton of wood is made up composed of rings and longitudinal braces. The inside is then lined with sheet copper which is hammered into shape and soldered together. For convenience the tube is made in two sections, which are fastened with bolts. The copper lining can be accurately formed, the wooden frame giving the necessary strength. The tube is light and inexpensive, and experience has shown that there is very little distortion when the wood becomes soaked. This same construction could be adopted for intakes and scroll cases. Such construction is limited to models for comparatively low head testing with low draft heads.

J. B. CUTLER.[13] The author states that "a review of the effect of altering parts of the water passages generally considered non-essential gives encouragement to the idea that substantial increases in efficiency and simplification of structures are to be had by continued effort along the lines of research." The writer assumes that by non-essential parts of water passages the author refers more particularly to speed-ring vanes, scroll-casing layout, and details of draft-tube design. It is undoubtedly true that higher efficiencies than at present obtainable will be developed by improvement of these details. The margin for improvement, however, above the present high-efficiency values of approximately 94 per cent is not large, and it does not seem probable that an increase of more than 1 or 2 per cent in actual calculated efficiency will be obtained.

The author's reference to the Thurlow backwater suppressor, with which it is possible to maintain rated output from a unit under greatly reduced heads caused by excess stream-flow conditions, gives rise to the thought that possibly greater importance should be attached to investigations conducted along this general line. The Moody ejector turbine, like the Thurlow suppressor, and other features developed for the attainment of the high capacities under reduced heads have been used with very satisfactory results. Considering the excess water available to the power company as an additional source of energy apart from the machine itself and without cost, the efficiencies developed from the standpoint of net return to the company might be considered as being in effect appreciably in excess of 100 per cent.

In view of this possible greater net revenue to the power producer from such devices as these, it would seem that further research along this line should perhaps be of greater ultimate value than along the line of improving features of design within the established water passages of the turbine itself. This conclusion is in contradiction to the generally accepted understanding that development in turbine design is definitely limited by the small margin in actual net efficiency, yet remaining under 100 per cent. In the writer's opinion the proved performance of the Thurlow suppressor should suggest fields of research the limits of which are as yet undetermined.

H. A. HAGEMAN.[14] For a number of years it has been the practice of this corporation to do model testing in connection with the design of important structures of hydroelectric develop-

[13] Safe Harbor Water Power Corporation, Baltimore, Md. Mem. A.S.M.E.

[14] Chief Hydraulic Engineer, Stone & Webster Engineering Corporation, Boston, Mass.

ments. Recent work along this line includes a variety of model tests of the spillway dam for the Osage development in Missouri now nearing completion, and the Rock Island development on the Columbia River, near Wenatchee, Wash., at present under construction.

In connection with the latter development, a model was constructed on a scale of 1 to 100 which reproduces the entire power site over a distance of 6200 ft. An extensive series of tests have been made on this model which have proved of great value in determining the effect of the river flows in an unusually rugged section of river channel. The largest flow on record at this power site is about 740,000 c.f.s., and flows corresponding to even greater than this amount have been experimented with on the model. As a result of these model tests, it has been possible to measure headwater and tailwater levels for a wide range of flow conditions, to determine cofferdam heights for protection of the work under construction, and to study the effect of changes in location and alignment of the various structures. Certain of these model tests showed that, by directing the river discharge from a battery of large flood gates through a deep channel below the power house, a reduction in tailwater level during high flows could be obtained very similar to that obtained by the Thurlow suppressor. No special construction was used to obtain the ejector action, which was accomplished by an arrangement of structures in relation to the natural topography. It is expected that for flows of about 200,000 c.f.s., the minimum head on the plant can be increased by this means at least 3 ft., which for an installed capacity of 150,000 kw. will mean a very considerable increase in available power.

The writer is thoroughly in accord with the practice of conducting model tests and believes that much valuable information can be obtained relative to the flow of water through hydraulic structures at small expense.

S. O. Schamberger.[15] The author's description of laboratory work done and field results accomplished in connection with the various hydroelectric developments of the Alabama Power Company indicates that considerable distance has already been covered in determining the value of experiment whenever there is a reasonable doubt in the mind of the designer as to the adequacy of a structure or the value of a change in design or operation.

Recent years have seen real interest taken in the subject of model testing, and one need only refer to the report of the Committee on Survey of Hydraulic Research of the Hydraulic Division of the A.S.M.E. to be convinced that there must be some justification for the work. Publicity is lacking, however. There may be some justification for the withholding of test data by equipment manufacturers on the grounds of economic pressure, but certainly this is true to only a small extent in the consulting field and not at all with operating companies. The author is to be commended on the experimental data he has made available to all.

Losses of any kind in an operating structure cost money. The elimination of losses is therefore warranted if the cost of elimination is not too great. Model testing to determine the ways and means may be the first step in the elimination of these losses if one can depend upon the accuracy of the results obtained. This paper, if it has done nothing else, has pointed out to an extent (not, however, to the extent that it might be hoped for) that results obtained by model testing can be checked in the field. It is to be hoped that more data can be made available in the future which will give the designer and the operator confidence in his ability to obtain field results as obtained by model testing.

[15] Hydraulic Engineer, New York Power and Light Corporation, Albany, N. Y.

Experiments described in this paper are practically all of the type where a change in form or line results in improvement in operation without appreciable increase in cost. In this type of experiment, the exact relation between the model and the prototype is not absolutely essential, as a dollars and cents value need not be placed upon the gain obtained by the model. It is important, however, to know that the field installation will react to the change in the same way, if not exactly in the same amount, as in the model. The paper has shown conclusively that this is true.

There is another type of testing which has its value to the designer in that losses experienced in waterways can be determined and evaluated before the construction, and in this way

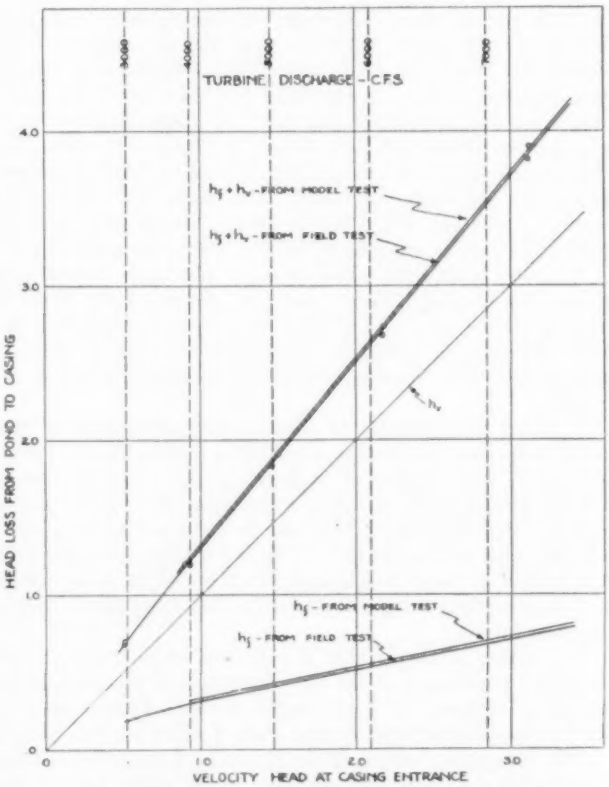


Fig. 17 Comparison of Lost Head as Determined From Model Test and From Field Test

(Conducted at New York Power and Light Corporation's Spier Falls hydro station.)

the most economic design be determined. With this type of testing, it is important to know the relation between the model and the prototype, and more data should be obtained from the field before confidence can be placed wholeheartedly in the model builder.

During the past two years, the writer has had opportunity to use a model on the scale of 1 to 48 to determine forebay and canal losses in a heavy rock cut, and also to determine the economic limit of increasing the velocities through an intake structure. These experiments were conducted at the Alden Laboratory at Worcester Polytechnic Institute in connection with the addition of a 37,500-kw. unit at the Spier Falls Station of the New York Power and Light Corporation. The test of these models has been briefly described in an article by C. L. Avery, published in the December, 1930, issue of the N.E.L.A. Bulletin.

The Spier Falls plant was built in 1902 and designed for an ultimate capacity of 9000 c.f.s. The characteristics and the

advantageous location of the plant with respect to the interconnected system made it desirable to increase its capacity for peak operation. The quantity of water through the forebay was to be increased from 9000 c.f.s. to more than 17,000 c.f.s. The enlarged forebay was in heavy rock excavation, so that dollars spent in taking out the rock should earn dollars in increased capacity.

Prior to the construction of the model, an actual test was conducted in the field to determine losses in the existing installation, and then that installation was set up in model form to determine the accuracy which might be expected from the model. The results were sufficiently convincing so that testing was continued to cover the entire construction project. After the model experiments for the determination of the final alignment of the forebay and the dimensions of the intake, construction was finished and a field check was made.

Because of the lack of time in testing the model before construction was actually under way, there were minor changes made in the design which resulted in a structure and water passages slightly different from those tested in the model. It is felt that the forebay model is as close a representation of the actual forebay as it is practical to make. There are, however, certain differences in the intake and penstock model, as follows:

1 The angle of the floor of the intake was 9 deg. steeper in the model.

2 The elevation of the stop-log sill was 2 ft. higher in the model.

3 The area of the head-gate opening was 3 per cent smaller in the model.

4 The percentage of the total acceleration produced in the transition section was 15 per cent less in the model.

5 The percentage of the total acceleration produced in the penstock bend was 13 per cent more in the model.

6 The model was tested without racks.

Of these variations, the first three tend to decrease the loss in the field, while the last three tend to decrease the loss in the model.

The curve shown in Fig. 17 indicates the agreement between the field test and the model test. For best gate capacity of 6200 c.f.s., the model indicated 0.56 ft. loss from pond to waterwheel casing entrance, while the actual field test showed 0.54 ft. The model test in this particular case indicated changes in design which resulted in a saving of some $80,000 in construction cost without appreciable increase in operating loss and unquestionably warranted the cost of the experimental work, which was in the neighborhood of $8000.

It is hoped that this comparison between the model test and the actual field check will help confirm the conclusion which the author has drawn—that model testing is a very great aid to the designer and may result in savings in construction and operation far in excess of the cost of experiment.

Author's Closure

Since presentation of the paper, field tests have been completed on the Lower Tallassee Plant, where identical turbines with different speed rings were installed. Fig. 18 shows results obtained on unit 1 with a 24-stay-vane speed ring and unit 2 with a 12-stay-vane speed ring. These results are in agreement with model tests as shown in Fig. 13.

The field tests define the shape of the performance curve more completely than is possible in the laboratory. It will be noted that the efficiency in model test 869, Fig. 13, shows a tendency to drop sharply after reaching the point of maximum efficiency. This characteristic is positively brought out in the field tests. The absolute difference in peak efficiency was found to be 1 per cent in the plant as compared with 0.6 per cent on the model. This difference is not due necessarily to inaccuracy in testing, but is inherent in the present laboratory methods. By selecting six or seven gate openings and covering a wide range of speeds, the performance curves are determined by intersecting the resulting horsepower-speed–efficiency-speed curves. The peak efficiency is established from the envelope. This method leaves considerable latitude in plotting the resulting performance curve. The need of laboratory apparatus whereby a constant speed on the turbine under test can be maintained is evident. This is only

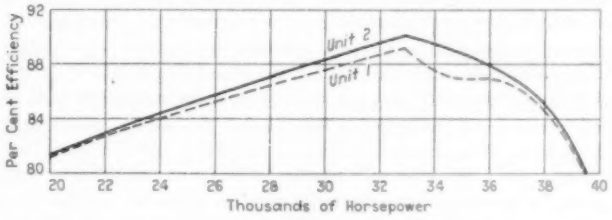


Fig. 18 Lower Tallassee Field Test, Over-All Plant Horsepower

(Comparison of 12 and 24 speed-ring stay vanes. Unit 2 has 12 stay vanes, and compares with test 868 of Fig. 13; unit 1 has 24 stay vanes, and compares with test 869 of Fig. 13.)

emphasizing the necessity of keeping all factors in model and prototype as similar as is practical.

Mr. R. V. Terry's comments regarding scroll cases is most interesting. The author is inclined to believe that the scroll case functions as a combination of vortex and elbow. By referring to test 121, Fig. 9, it will be noted that there are three separate streams of water entering the speed ring. The upper and lower streams tend to flow radially, following the elbow characteristics. The central stream tends to flow more tangentially and in agreement with the vortex law. Mr. William White, of Allis-Chalmers, emphasized this fact during an informal discussion of the paper.

In making the 4-in.-model draft-tube experiments, a turbine speed ring with guide vanes was used. The guide vanes were at various angles adjusted to produce whirl in the draft tube corresponding to 0, 15, 30, and 45 deg. with the vertical axis of the turbine. The angle was measured at the outer walls of the draft tube. The efficiency was completed by the formula $E = (C2/C1)^2 - 1/C2^2$ in which $C1$ = coefficient of discharge (based on runner D_2) without draft tube, and $C2$ = coefficient of discharge with draft tube attached. This formula is a function of external and internal head and has no factor for the inflow and outflow velocity heads.

The effect of cones in the draft tube, as noted by Messrs. E. Brown and L. B. Stirling, is of interest, as it was found in the Worcester tests that a large cone extending to the runner shaft increased the efficiency of the turbine. The function of the cone in this case was to fill a void under the runner rather than to correct areas within the draft tube.

The bell-mouth penstock entrance used in the plate-steel scroll case experiment was not arrived at experimentally. The object of the experiment was to determine the effect of the penstock elbow at the scroll entrance. The following table of plant tests, based on weighted values for all sections of water passage, will indicate to Mr. R. L. Thomas the combined head losses from forebay to turbine entrance to be expected in actual plant operation:

	Ratio L/R	Head loss, % H_v
Martin Dam	40.0	36
Upper Tallassee	21.8	65
Jordan Dam	7.6	33
Lower Tallassee	47.5	83

in which L = length from penstock entrance to net head piezometers in feet and R = hydraulic radius in feet. Martin Dam

has a plate-steel penstock in a single pipe. Upper Tallassee, Jordan Dam, and Lower Tallassee are three-pipe concrete penstocks. The Upper Tallassee water passages are comparable to the Safe Harbor design.

Mr. Carroll F. Merriam states the fact that conclusions should not be deduced from even a great number of repeated experiments unless the phenomenon is clearly understood. Hydraulic research as conducted by various operating companies in this country is of a defensive nature. Only limited funds are available for offensive research to point out the phenomenon, enabling the engineer to make a direct approach to the solution of his problems.

The position taken by Mr. G. W. Spaulding relative to the performance of draft tubes with various specific speed wheels is well founded. The author wishes to qualify his statement that "in general, it can be stated that a good draft tube for one wheel will also prove efficient with another wheel," to the Francis type runner, as he has had no experience with the propeller turbine.

The construction of draft-tube models in sheet copper, as described by Mr. LeRoy M. Davis, represents an advancement over the method used by the author. The models referred to in the paper were constructed of white pine and followed pattern-work practice. An average 16-in. model draft tube costs $350, while a scroll case may reach $500. Deterioration is rapid due to the wood swelling and joints opening, making constant inspection and repairs necessary.

As pointed out by Mr. Davis and Mr. Hageman, there is much to be gained by building a model of the complete project. In the case of Mitchell Dam, the plant arrangement using the Thurlow suppressor is equivalent to increasing the over-all turbine efficiency by 4 per cent.

A valuable contribution to the paper is offered by Mr. S. O. Schamberger in presenting the results obtained at the Spier Falls Plant. The agreement of head losses as determined on the model and at the plant is remarkable. An interesting comparison of intakes and entrance losses could be made by reducing the Spier Falls values to those shown in the table for various plants of the Alabama Power Company.

The author wishes to thank the contributors to the paper for their assistance in materially adding to the usefulness of the data presented.

MAN-53-1

American Management in Europe

Industrialists in England and on the Continent Exhibit a Growing Interest in Methods Installed in the United States—The Purchase of More Efficient Machines and Equipment Has Not Brought the Expected Prosperity, and It Is Now Found That Management Is as Important as Processes

By WALLACE CLARK,[1] NEW YORK, N. Y.



THE scope of management engineering is no longer confined to the installation of so-called mechanisms of management or the technique of carrying on the routine of a business. It now covers all the psychological, physical, and economic conditions which affect a certain enterprise. It applies method to every phase of industry and nothing is left to the old haphazard way of getting into an unexpected situation and then moving heaven and earth to get out of it. The executive in these days is expected to foresee practically every change in conditions both inside his company and those outside which will affect it. Where possible he is expected to prevent an undesirable event from happening, or when he cannot do that, he must already have planned a course of action which will make the best of the changed conditions. It is obvious that the man who first sees his way out of a difficulty has a decided advantage over his competitor.

The management engineer has for a long time provided the executive with the technique for the routine management of production and distribution, and more recently he has provided budgets for the financial direction of the business. He cooperates with the statisticians in applying forecasts of general business conditions to a specific industry. He works with the economists in their attempt to secure a greater general prosperity and with the workers in their desire for greater security, better living conditions, and the chance to use and develop their creative energies.

The Application of Method

The application of method to his daily duties does not deprive either the worker or the executive of initiative. On the contrary, it frees more of his time for those parts of the work which require the use of his creative faculties rather than his memory.

If a workman has to do a certain task for the first time, there is no good reason why he should be allowed to flounder around and discover his own method when the same problem has been met and solved a thousand times by other men. It is obvious that the existing knowledge should be made available to him in the form of a method, and that he should use his mind in improving that method. If the situation is complicated, he must use his judgment in deciding which of the several methods will accomplish the best results under those particular circumstances. The application of method means the selection of the best from the many current ways of doing a thing, improving that, if possible, and then teaching it to all who have to do that thing. In this way the individual starts from the best-known method and employs his creative ability in developing a better way. He does not waste his time in discovering things which were discovered long ago.

The same is true of the executive, for the application of method to his work frees his mind for the use of keen analysis and good judgment in determining a course of action and of will power and resourcefulness in following that course.

One of the greatest services the management engineer can render is to make existing knowledge and method available, so that constructive minds will not need to go over ground which has already been covered; to simplify the routine so that it will require a minimum of time, and in other ways to free the creative energies of executives and workers.

The pioneering work of applying science to management has been done and the mechanisms or technique of management have been developed. Much will be done in future to improve this technique and extend its application to new fields, but the task of the management engineer now is to place this science of management at the disposal of every part of industry and then to concen-

[1] Wallace Clark & Company, Consulting Management Engineers. Mem. A.S.M.E. Mr. Clark was graduated from the University of Cincinnati in 1902, and then entered the employ of the Lodge & Shipley Machine Tool Co. In 1910 he was engaged by the Remington Typewriter Co. in New York City. When H. L. Gantt was retained by that company to install his methods of management, Mr. Clark was selected as an understudy and later as an executive to carry out the new system.

In 1917 he resigned from the Remington Company to join Gantt's staff and went to Washington, where he had charge of the latter's work of planning the utilization of ships controlled by the U. S. Shipping Board. In 1919 he started his own consulting practice, which he has since maintained.

During the next six years he investigated the operation of the U. S. Patent Office for Herbert Hoover, then Secretary of Commerce, and was a member of the commission which drew up the plan for the organization of the proposed Federal Department of Public Works. In 1926, as the engineer member of the Kemmerer Finance Commission which stabilized the currency of Poland, he recommended methods for the reorganization of the Polish salt and tobacco monopolies. In 1927, at the request of the Minister of Finance and a group of manufacturers in Poland, he reorganized some of that country's leading industries and since then has extended his services to Germany, France, Switzerland, Czechoslovakia, Rumania, England, and Denmark. He is the author of a number of books, among them being "The Gantt Chart," which has been translated into seven languages.

Contributed by the Management Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

trate his energies on developing the individuals and stimulating their creative brains.

Making These Methods Available Abroad

Should an American engineer make the methods developed here available to industry in foreign countries?

That is a question which is sometimes asked and more often suggested indirectly. Should America keep her prosperity to herself and not share it with the rest of the world? Should we too develop the intense and selfish nationalism which brought on the World War and is still a grave danger in Europe?

To the engineer or the scientist the answer seems obvious, for to him right methods of work and good conditions for workers have no nationality. Politics is outside his domain. If improved processes and methods, through some later political upheaval, are turned to the uses of war, it is because civilized brains and intelligent forecasting are not concentrated on removing the underlying causes or conditions which lead to the political upheavals or catastrophes. If the economics of nations were planned according to engineering principles, there would be friendly cooperation in place of fear and hatred.

America, as a matter of fact, has gone far in making her methods available to foreign countries. We began years ago to ship our machine tools and other labor-saving devices to Europe, and they have had a marked effect on increasing the productivity and wealth of those countries. We have exchanged our inventions for those originating abroad.

It seems to be generally agreed in Europe that our methods of management have been largely responsible for our recent period of prosperity, and our books and papers on management have been eagerly translated and published. Countless delegations and individuals have been sent to study our plants, and American executives have freely opened their doors and have given the visitors all the information they could take away with them.

Industrialists and engineers have apparently learned that no country can live entirely to itself, and that our prosperity is closely tied up with that of the rest of the world.

Maintaining Our Standard of Living

The present business depression has lasted so long that some in Europe are asking whether we can maintain our high standard of living, and even more frequently in the American press some one says that the standard of living of our workmen must be reduced.

To the engineer who has taken part in the growth of management, any loss of what men have gained through the development of industry is unthinkable.

If the workman is to go back to an outgrown stage in his progress, there will be no need for the inventor to bring forward his ideas for improvements aimed to make life easier and more satisfying. The scientist will have no incentive to carry on his experiments for the benefit of mankind, and the engineer will be out of a job.

It is not the American way to be satisfied with things as they are. It is not intelligent, with the advance we have made, to continue to suffer these periodic depressions. The task before America is to regain our prosperity and then to stabilize it.

We have an abundance of raw materials and foodstuffs, plenty of machines standing idle, and millions of men and women wanting work. There are many more millions who need the good food, better clothing, housing, and other things produced by those machines.

The money to buy the products of our machines must be placed in the hands of those who need to buy them. Some way must be found to direct a flow of money to consumers which will more nearly balance our ability to produce goods. When this problem is solved, we shall be able not only to maintain our standard of living, but to continually raise it.

Progress in Europe

The leaders in European industry generally recognize that a higher standard of living for their own people can be attained only through a better development of their own resources, increased productivity of labor, and better methods of management to eliminate waste and to coordinate the various parts of each business. Much progress has already been made in the application of American methods, as will be shown in the summaries from different countries. Some of the characteristics of European industry are these:

There is less mass production there than in America because of tariff walls, different habits of peoples, and lower purchasing power.

Turnover of investment is slower, due to less effective management.

There are not enough practical men among the executives, due to class distinctions.

The introduction of results of research into practice is slower.

More labor is wasted, because wages are low.

There is more respect for knowledge and authority than in America.

Workmen have greater security.

These characteristics of industrial conditions in Europe apply only partly to Great Britain and not at all to Russia, where the fundamental structure produces different shop and business conditions. I should also except the three Scandinavian countries, where the wages and standard of living compare fairly well with those in our country.

In the installation of management methods in Europe among the obstacles to be overcome are: the necessity of working through interpreters in several languages and differences of mentality and psychology, but these can be overcome by carefully planning the installation.

In the application of American methods the following points are noticed:

In Europe more time is required to secure delivery of purchased materials.

Rearrangement of old buildings is more frequently necessary.

Reserved-time scheduling is more needed than chain production.

Market research and sales organization have not made much progress.

Budgeting is well understood.

Methods of executive direction are as much in demand as in America.

There is now in Europe a more general understanding of the fact that conditions are changing and that methods must be flexible.

Management Helps the Worker

The introduction of the new management into European plants has almost invariably brought about better working conditions, higher wages, and steadier employment, just as it has in America. It is fortunate that there is on both continents a growing recognition of the worker. It is particularly easy to see this in some of the older countries where there are so many reminders of the early stages of the industrial era, when a worker's time, health, and even life itself were held so cheap. The long hours he worked, the miserable wages he received, the inhuman punishments he suffered for any infringement of the rules, and the conditions under which women and children were employed seem incredible today.

Characteristics of European Industry and Growing Interest in American Methods

THE growing interest of European industrialists in American methods of management is well known. For a long time they believed that the purchase of the most efficient machines and equipment would bring them American prosperity, but they have learned, as we did over here, that management is quite as important as machines and processes.

As a body of technicians The American Society of Mechanical Engineers may be interested in some of the characteristics of European industry noted in the author's daily work of installing management methods and how those methods are adapted to foreign conditions. During the last three and a half years in the plants and mines of nine European countries, his organization's clients have been engaged in the following industries: mines of salt, iron ore and coal, blast furnaces, foundries, forges, rolling mills for steel, brass and copper sheets and tubes, locomotives, railway cars, automobiles, airplane motors, agricultural and textile machinery, cellulose and mechanical pulp, paper, lumber, printing, lithographing, clocks, screw-machine products, chocolate, cotton, wool and linen textiles, municipal tramways, tobacco monopolies, and state industries.

Mass Production

The characteristic of European industry most frequently discussed is that there is less mass production than in America. One reason is that the countries are smaller and the tariff walls prevent the distribution of manufactured articles to a body of consumers large enough to justify mass production. Consumption is also limited by the marked differences in habits of the people in the various countries. A third reason is that wages are low, and therefore the purchasing power of the average family is not sufficient to enable it to buy many of the manufactured articles which in America are purchased in mass. Consumers, particularly in France and England, also demand greater individuality of design than we do.

Nevertheless there is a good deal of mass production in Europe accompanied by strict standardization of design and the use of automatic machines and of moving belts for assembly. Where production of this kind has been successful it has so simplified the management that other manufacturers have looked on it with envy and have frequently installed the equipment for chain production when the volume of sales did not justify it, and as a result the fixed investment became so great that interest and depreciation swallowed up all the profits. Manufacturers in this situation have often taken the course of dumping their products on the market without profit in order to keep their machines running.

However, in a majority of instances the adaptation of mass or chain production has been extremely profitable. Most of our clients, by standardizing design, reducing the number of styles, and rearranging production to match the tempo of moving belts, have increased output by 25 per cent or more, and have so greatly improved the quality of product and lowered its cost that they have built up domestic sales, and some have opened up foreign markets which were not known to exist.

Turnover of Investment

A slower turnover of investment is another characteristic of European plants. Raw materials are kept in stores much longer than necessary and work moves deliberately through the process of manufacture, which in turn makes it necessary to keep larger quantities of finished goods in stock. This slower time of production requires more investment in proportion to the volume of sales than in the United States.

This difference in tempo between the industries of the two continents has of course a decided influence on resulting profits. European executives are learning that it is more economical to turn over their capital quickly than to secure additional capital on which dividends or interest must be paid.

The American technic of production invariably brings about a swifter flow of material and ideas, and therefore a better turnover of investment. In a Polish plant, for example, where about 3000 workmen are employed, the new methods of management increased production 55 per cent with the same number of workers, very small investments in equipment, and with no changes in executives.

Practice and Theory

Another characteristic of Continental industry is the tendency of the managing staff toward abstract theory, and therefore one of the problems which faces the consultant in organizing a European plant is that of securing a good balance between practical and theoretical men. The executives from the superintendent up almost invariably have engineering degrees with good theoretical training behind them, but seldom any actual shop experience in operating machines or processes.

On the other hand, the good shop men are very seldom promoted to positions of responsibility in the management. A workman expects to live and die in the class in which he was born. There is almost no training of good workmen to become foremen, and no classes to fit foremen for higher responsibilities. Evening technical schools are rare and the same is true of correspondence schools. The workman does not expect promotion from the ranks, and does little to fit himself for it.

In organizing American shops we do not have much difficulty in securing an even number of technically trained men and those who have been brought up in the shop. By placing them side by side we develop a management which is more sure of itself and quicker to make decisions and take action. In Europe this is more difficult to accomplish because of class distinctions, but fortunately it is becoming easier each year as class lines are relaxed.

In a Polish spinning mill we found a need for more supervision over some of the difficult operations such as doubling and twisting. The superintendent said there was no one who could be assigned as supervisor. On being asked who knew most about this particular operation, he replied without any hesitation by naming one of the operators, but said that she would not do because she was just one of the peasant girls and no one would respect her and follow her orders. However, he consented to give this girl a chance, and a few weeks later told us that the other operators were all willingly obeying her instructions, and that the production of that battery of machines had so far improved in quality and quantity that they were no longer holding back other operations. He then asked us to do the same thing for succeeding operations.

Such promotions from the ranks are not as frequent as they should be, and the reason is not so much that executives oppose it as that it does not occur to any one that it can be done.

Practical Application of Knowledge

Due to the theoretical bias of the average European executive, it is easier to get a matter studied or investigated than it is to get the resulting knowledge introduced into daily practice. In the United States we are more apt at making practical use of what knowledge we acquire, and therefore the results from research are proportionately greater than in Europe, although they have carried on their investigations and experiments for many more years. This fact is realized by the forward-looking managers in Europe, and they are devoting more attention to seeing that the

results of research are "introduced into the life," as they express it.

In a French cotton mill we found girls in the packing department doing all the heavy lifting, while one of the plant engineers for some months had been designing equipment to avoid this fatigue. But the girls went on lifting, while the engineer, enjoying his problem, went on refining point after point of his design. When the engineer was instructed to install the equipment without the final touches, the production of that department increased 45 per cent. Then the executive began to set dates for the completion of all work done on his instructions, and incidentally the girls enjoyed higher wages with a great reduction of fatigue.

Waste of Labor

On the Continent there is extreme economy in the use of materials, which are expensive, and a prodigal waste of the time of workers because wages are low. This is the opposite of the practice in America and must be kept in mind by the consultant.

In a cellulose-pulp mill in Germany twelve men used to push railway cars from one point to another in the factory yard. Those cars are about one-third the size of American freight cars, but when filled with pulp wood they are quite heavy. One man and an electric truck, which was already in the plant but seldom used, were substituted, and eleven men were transferred to other work which was more important and required more skill. Similar examples could be cited without end.

Due to the lower level of wages it is obvious that many labor-saving devices which are economical in the United States will not pay for themselves in Europe within a reasonable time. This must be kept in mind in appraising the efficiency of a plant, and it is one of the reasons why the production per man is not so great as in America, where a workman has at his command more power-driven tools and transportation equipment.

However, the level of wages in Europe is rising, and we have in almost every case found managers willing to increase wages and to share with the workers the savings resulting from better management.

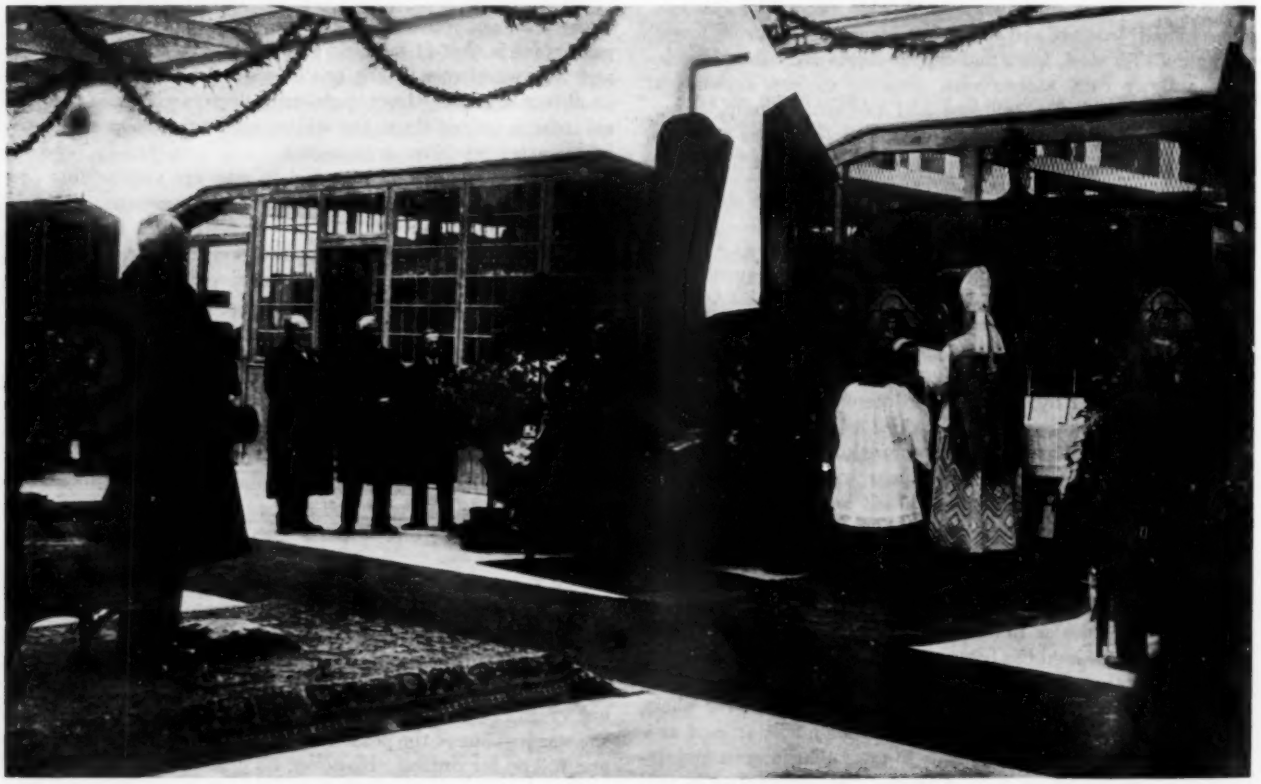
Respect for Knowledge

The respect for knowledge in Europe is probably greater than in America, where money and influence seem equally important. For this reason the work of the consultant is somewhat easier because he can usually be sure of a sympathetic consideration of his proposals.

Security for Workers

Europe provides for workmen greater security and continuity of employment than we do in America. A worker cannot be dismissed without notice ranging from two to four weeks, and consequently foremen do not depend on threats of discharge to maintain discipline.

In most of the countries workmen have old-age, sickness, and unemployment pensions administered by the governments. These things help to free the workman from worry about the maintenance and happiness of his family, and enable him to become more proficient in his trade. However, the dole both in Germany and Great Britain has serious disadvantages, and the difficulty of taking it out of politics seems almost insurmountable.



Blessing the New Machinery in a Polish Factory

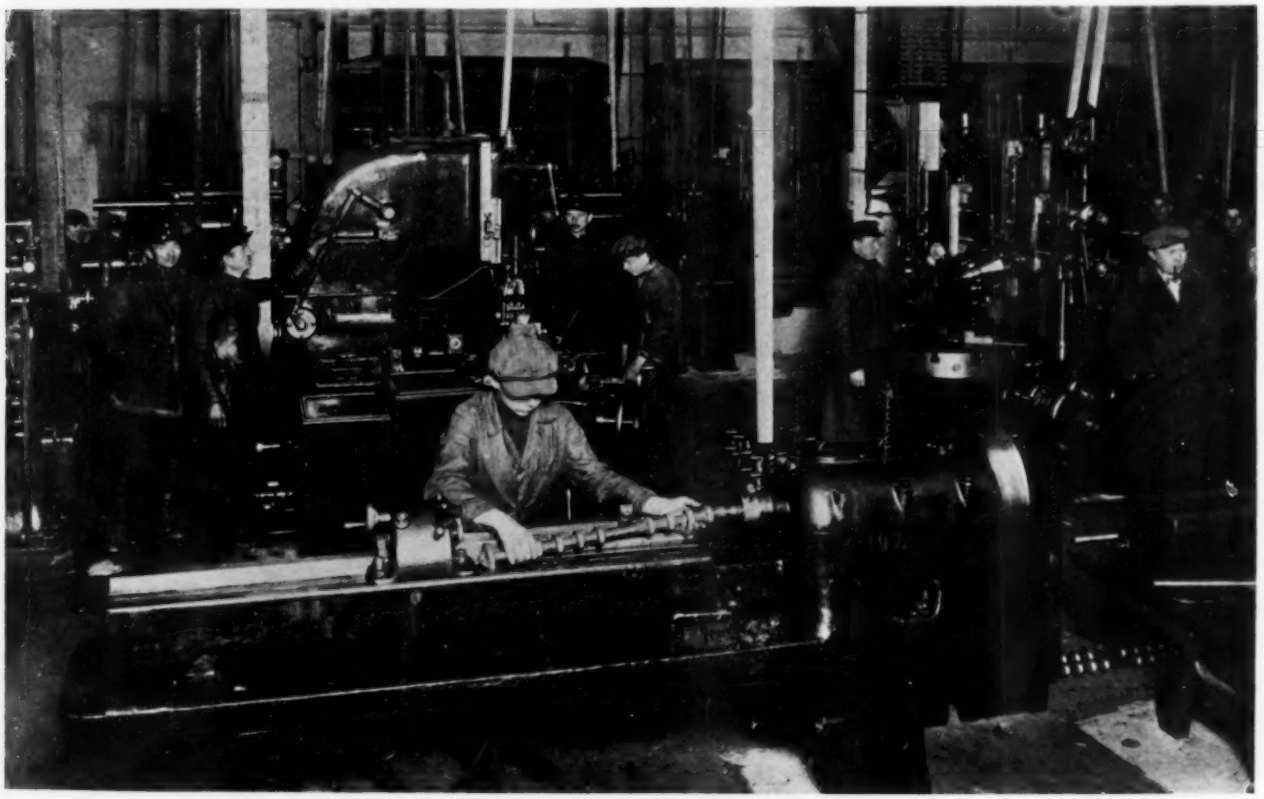
Respect for Authority

European workers have more respect for authority than American, probably because of class lines. A worker will seldom disagree with his superior, and this often makes it difficult to get at the essential facts in regard to many conditions.

In parts of Germany and in countries east of it, whenever a

workman sees an important executive or visitor he takes off his hat, and the older men also bow from the waist. Etiquette requires the executive not merely to touch the brim of his hat, but to lift it.

In France and England this is not the custom, because workmen there are more independent and are not afraid to disagree with their superiors. In England the workmen are backed up by strong organizations, while in France, where workers' organizations are not so strong, the managers have found that their people do not work well in an atmosphere of repression. There is, therefore, more appearance of freedom, and in place of hat lifting and bowing one is more likely to hear a polite "good morning." A French workman may even disagree with his foreman, but he will express himself with the greatest politeness, and also with what we would consider a great many unnecessary words.



Modern Machinery in a Polish Plant

In all European countries there are, of course, plants in which the relations between the owners and the workers are extremely cordial, due to long years of wise and sympathetic consideration of the needs of the workers. As a rule, however, the bow of the workman and the hat lifting of the manager are not accompanied by any relaxation of the facial muscles. One longs for the smile and quick nod which flash through an American shop.

Great Britain and Russia

These characteristics of industrial conditions in Europe apply only partly to Great Britain and not at all to Russia. British shops are in many ways more similar to American than to Continental shops, as England was the cradle of modern industry and we have learned more from her than from other countries.

Russia is, of course, unlike other countries in its fundamental structure, and therefore shop conditions are quite different. For instance, class lines do not separate the workmen from the executives, because there is only one class. Very few managers and engineers have been left over from the old régime, so their places must be filled from the ranks of the workers. There is therefore a determined effort to train exceptional workers for the responsibilities of management, and to provide them with the best available methods. In no European country is there so much eagerness to learn American management methods as in Russia.

The development of managers and of technical engineers appears to be the key to the success of the much-discussed Five-Year Plan. It is a tremendous task to transform an agricultural into an industrial country within such a short time, but the difficulty of changing peasants into machine operators and skilled workmen is nothing compared to that of developing within such a few years a group of men capable of managing and providing the technical knowledge necessary for industry on so large a scale.

The developing of Russian managers is more difficult than usual due to the absence of rewards as a stimulus to improvement. A manager gets a salary which is frequently not more than 50 per cent in excess of the wages of a skilled workman. This does not enable him to provide for his family a home which will free his mind and energies for his responsible work.

In the early part of this year the heads of the Soviet Government realized that the workers' organizations were interfering too much with the authority of the managers, and they therefore passed new laws giving the managers more authority and at the same time providing prison terms or exile to Siberia for those managers who do not carry out their programs successfully. To an American mind the substitution of the fear motive for the desire for reward and self-improvement cannot develop the best managers. The most satisfactory results from scientific manage-

ment have been secured where the direction of industry is the most democratic.

Technic of Installation

There are certain obstacles to be overcome in the installation of American methods of management in foreign countries. One of the first to be met by one who does not speak the continental languages is the necessity of working through interpreters. Obviously it requires a little more time to repeat every speech, but, on the other hand, translating has a tendency to make a conversation more precise and to the point. While a paragraph is being interpreted one has a chance to think through the next point to be made.

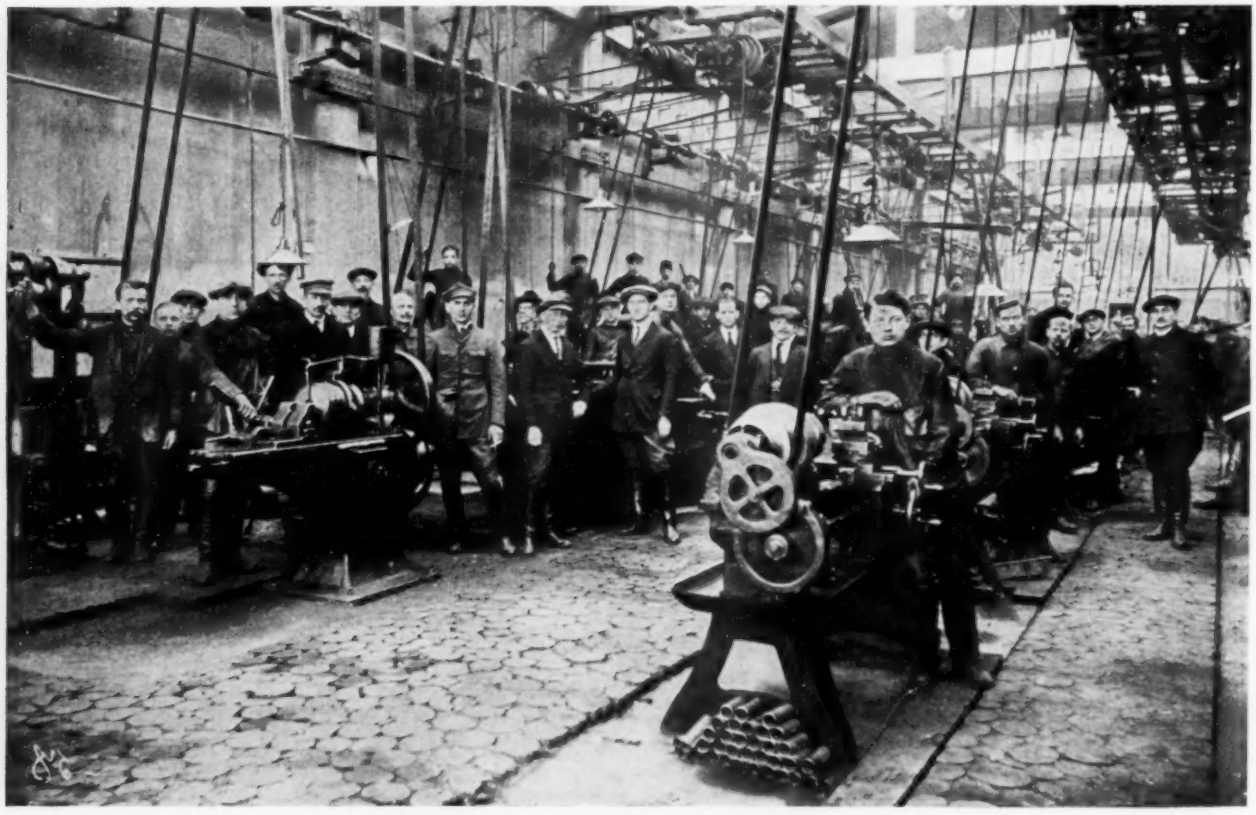
It is of course necessary in our work for an interpreter to understand thoroughly the various phases of management and the special terminology in both languages. Frequently he has to stop and give a long explanation of a word. For instance, there does not seem to be in French a single word which translates "planning," and he must say: "the preparation, distribution, and routing of work."

In each country it is necessary to study the mentality or psychology of the workmen and executives in order to understand them better, and to know in what particular way any subject should be presented in order to create interest.

We have found that the best results are accomplished when the installation is made by a combination of native and American engineers, one being most proficient in the technique of management and the other knowing the national psychology and habits, as well as the developments in the past which have led up to present conditions. In each of the countries in which we operate, the staff is therefore made up in this way, and the native engineer usually spends twice as many weeks in a plant as the American.

Our consulting work consists mainly of the installation of management methods, that is, adapting methods to a particular situation and staying with it long enough to accomplish practical results. In some cases we analyze the condition of an existing company and recommend a plan for the future conduct of the business. In other cases, after our installation is finished, we supervise the maintenance of the methods and the progress made in the execution of the more important business policies which we have recommended. This service is applied to groups of plants as well as to single companies.

It may be sufficient to outline our technic of installation rather than that of analysis.[2] We handle several clients at a time, spending one week with A, a second with B, a third with C, then returning to A. We have found that an installation is more thorough and therefore followed by better results when we do not split a week between two clients, but devote the entire five days to a single plant.



Group of Workmen in a Polish Factory

We work on the assumption that the increase in production, improvement in quality or service, and reductions in cost which are realized during the term of our engagement are less important than those which should follow after we leave. It is therefore necessary for the managing staff to understand these new methods and their principles, and how to get them accepted and used by the entire personnel. One of the most important duties of the consultant is to train some of the members of the clients' staff so that they can continue progress after his engagement is ended.

[2] A more complete description of methods is given in the author's paper entitled "The Technic of Installation of Scientific Management," presented before the International Management Congress in Rome, 1927.

When we begin our visits a client assigns to us for training one or more men from his staff, relieving them of all other duties so that they can spend their entire time on the new work. These understudies watch every step and soon are able to carry forward many things themselves. We ourselves do not do any repetitive or detail work which can be done by the employees of our clients.

During a typical week's visit to a plant we first go over with our understudy the things they have been doing since our last visit, explaining any points which have not been clear and improving the operation of the method. We then develop the next phase of the installation and begin its actual use. Several things are kept going at the same time, each one being looked after by a different understudy—for example, the installation of planning, storeskeeping, operation studies, costkeeping, and the physical arrangement of the shop. On Thursday a report is dictated outlining as clearly as possible what has been done during the week, attaching copies of all instructions, designs of forms, prints of floor plans, etc., and concluding with the statement of what we recommend to be done before our next visit. This report is presented in two languages, and on Friday afternoon is discussed in detail at a conference of the chief executive of the plant and those members of his staff who are concerned with the execution of the recommendations.

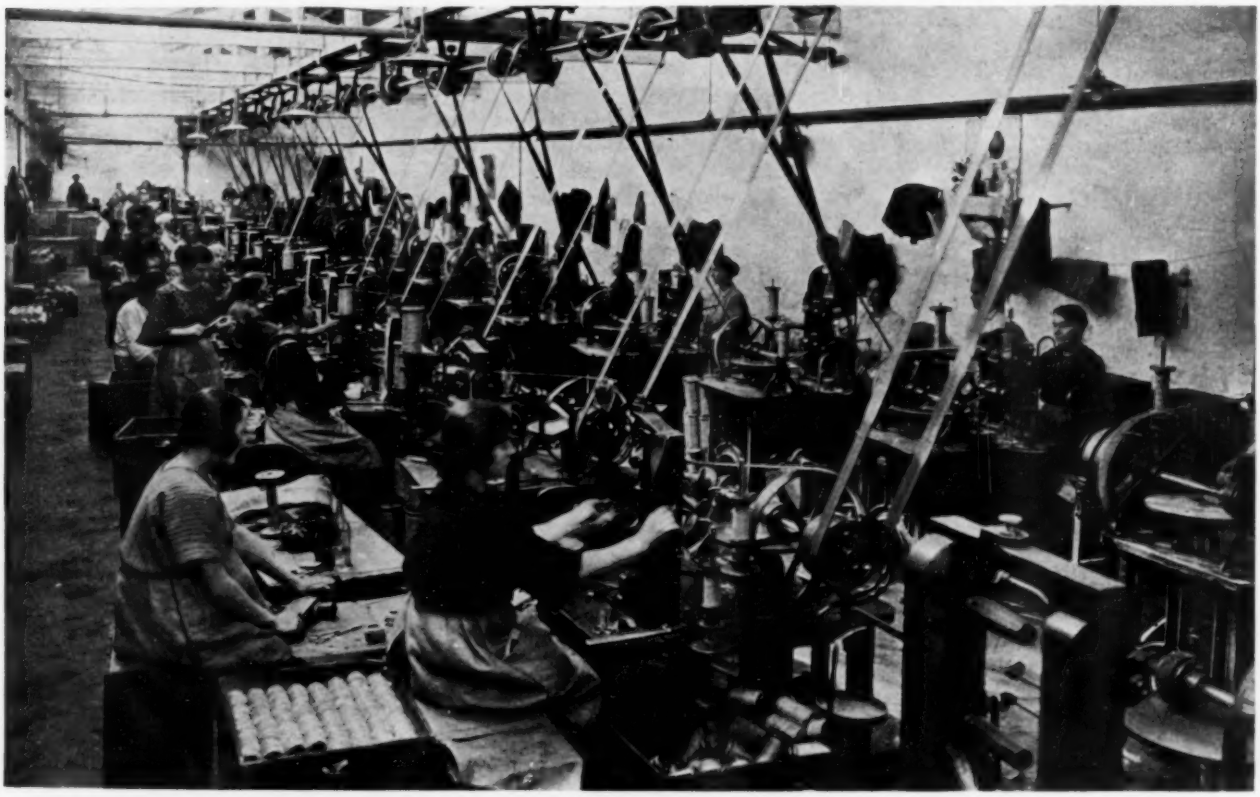
These reports therefore become records of progress or lack of progress, and are extremely useful in securing cooperation and the quick introduction of methods.

After we have spent a short time at a plant and understand the conditions and the personnel, we schedule our own work of installation on a Gantt chart, sometimes a year in advance, showing the various phases of the installation and the weeks we shall be at the plant. For each department in the plant there is a schedule on which a date is set for the beginning and completion of each phase of the installation. As the weeks pass, heavy lines are drawn on the chart to indicate the progress compared with the plan.

Each of the client's understudies has a chart showing the details for which he is responsible. Throughout the installation the time element is emphasized, because one of our major aims is to accelerate the tempo of the entire business, and the promptness with which the executives and staff make decisions and execute instructions determines the tempo of the plant.

Differences in Application of Method

In Europe, as in America, every plant is different, and in installing the so-called mechanisms of management it is quite impossible to impose any rigid set of methods; it is obvious that they must be carefully adapted to the individual business and also to the mentality of the workers and the executive staff.



Women Workers in Poland

We have found, however, that in general the mechanisms, such as planning, storeskeeping, salesmen's quotas, and so on, which were first developed in America, apply equally well to conditions on the Continent. Their application is apparently universal, and the changes which need to be made to fit local conditions are not fundamental.

Storeskeeping

In installing Gantt storeskeeping methods, for example, we find that the use of standard bins and boxes, the physical marking of order points, and the routine of ordering, receiving, and issuing materials is practically the same as in the States. There is, to be sure, a greater tendency to require signed receipts for items of small value, but this is not difficult to overcome. In Europe the tendency to save material is sometimes carried too far, for there is too much delay in selling scrap and in disposing of obsolete items which have little chance of ever being used.

The difference is most noticeable in determining "order points" and "quantities to order," which of course depend upon the time required to secure a new supply. Some countries are not self-contained industrially and therefore many items must be imported from other countries, and the delays in going through the customs considerably lengthen the average time of delivery. Even in the more highly industrialized countries the service from suppliers is not nearly so prompt as in the United States. Filling and shipping an order the day it is received is almost unheard of. In those countries in which cartels have eliminated competition, the service is noticeably poorer.

All these things cause the storeskeeping and purchasing departments to allow long periods for deliveries of materials—usually far longer than is necessary even under such conditions. It is necessary to get a purchasing department to adopt a more exacting and aggressive attitude in dealing with their suppliers, but even when the storeskeeping and purchasing departments have done all they can, it is not possible in the average European manufacturing plant to do as large a volume of business on as low an inventory of materials and supplies as it is in America, due to the longer time of delivery.

Plant Layout

There is little difference in the application of the principles of plant layout in Europe, for managers are just as anxious for straight-line production as in the States. Because there are so many old industries, the problem of rearranging the equipment and machines in old buildings is more frequently met than at home. In one plant we had to change the layout of machines in a building which was about 130 years old, and a direct flow of material was secured without breaking down any of the walls.

Planning and Scheduling

Since there is less mass production in Europe, there are fewer shops in which the rate of movement is set by an assembly belt. The need in most cases is for reserved-time scheduling, that is, the type of planning which reserves the time on all machines so that when material is issued from the storesroom it will move at a steady rate through the shop. Wherever installed this type of planning has brought large increases in the volume of production, ranging from 12 to 40 per cent.

Some of the delays in planning are surprising. When a new cupola was to be installed in a foundry we had to reschedule the work to allow a day for the building of an improvised altar on the molding floor, and another day for the priest to bless the new equipment so that it would be safe for use.

Costkeeping

In costkeeping no important alterations in methods are necessary to meet European conditions. We have found that the modern cost methods are more readily accepted than in America because executives are more willing to adopt a method as a whole without wishing to alter details. One client, the director of a municipal tramway company, said that he expected us to provide methods adapted to the peculiar conditions of his plant, and that the members of his staff would give us all the information which we needed, but would not attempt to mix their old methods with the new.

Methods of costing provide the most effective means of improving conditions. In a Polish pulp mill it had been the custom for years to send six men down into a digester to tramp the chips before cooking. At the end of their tramping these men would come up out of the digester on a rope ladder, stripped to the waist, dripping with perspiration, covered with sawdust, and breathing through handkerchiefs which were almost useless in protecting them from the dust in which they worked.

Without any reference to the welfare of the workers, a brief comparison of the cost of time lost in this way with the small gains of tramping quickly brought about the installation of a revolving spreader for the chips which made it unnecessary for any men to go down into the digester.

In the same mill, after the cellulose had been discharged from the digesters, workmen had to go down into the pits and shovel the mass into a chute. Their wooden shoes did not prevent them from getting wet to the knees and the fumes of SO_2 were so strong that they had to come up for air about every ten minutes.

The traditional attitude in such a case is that men are cheap and there are plenty of them. But in order to reduce costs a suction device was installed, and the men no longer had to work under those unhealthy conditions.

Many examples could be cited to show how American methods break through the European attitude toward cheap labor and improve conditions for the workers.

Improvement of Operations

In the study and improvement of processes and operations there is a great deal of inertia to be overcome. Usually the first reaction to a proposed change is that it is impossible, and the second that it is not worth while, because the time saved would be wasted some other way. However, as soon as the interest of the managers is aroused they become eager to make these studies with their own men, and after some training in the technique these men succeed in bringing about marked improvements.

The men chosen for this work sometimes have technical training, and in other cases are workmen who have developed exceptional skill. They are so close to the workers that they are able to stimulate creative ideas in the workers' minds, and as a result, the latter take much more interest and pride in undertaking and mastering new things.

Incentive Payments

The idea of incentive payments is not new in Europe, and in the majority of plants in the author's experience, piece rates are in common practice. The rates have not usually been set with sufficient care, nor has there been any attempt to determine the best method of doing the operation; therefore the rates are not successful in securing uniformly high production.

In applying a bonus system there are no unusual difficulties, except that simplicity is even more imperative than in the States because workmen are not so well educated. In Russia, where the equality of workmen is emphasized, piece rates in several shops were said to be a recent development.

Sales and Distribution

In spite of the wide acceptance of American methods of production, Europe has not yet awakened to the importance of distribution. The need for market analyses is even greater than in America, because there are more pronounced differences in the customs and buying habits of the people in the various sections of European countries.

In Russia there is no problem of selling because there is almost no private trading, but distribution is extremely difficult through such an immense territory.

Budgeting—Executive Control

Within the last few years Europe has taken a keen interest in the matter of financial budgeting. This was shown by the remarkably successful Budgetary Conference held in Geneva in July of this year under the auspices of the International Management Institute.

Even in plants where expenditures are closely watched, as they usually are on the Continent, it has been surprising to see how ex-

penses have been reduced by careful planning and the fixing of responsibility which the budgets have brought about. Accountants have welcomed them because they do so much to clarify the relations between them and the executives.

Recent developments in methods of executive direction have attracted the attention of forward-looking executives in Europe, who are eager to learn the most dependable technique of securing cooperation and of directing instead of following the progress of a business.

It is obvious that one of the principal duties of a chief executive is to prepare his company to meet whatever happens in the future, and this directs his attention to forecasting conditions in general business as well as in the specific business in which he is engaged. But European statistics needed to form a basis for forecasts are far from complete. The various governments keep records of imports, exports, bank clearings, and other matters which pass through their hands, and they publish figures which are usually prompt and accurate. Some Chambers of Commerce also provide valuable data, but the information about specific branches of industry, which in this country is secured through trade associations or the Department of Commerce, is rather meager. For this reason planning for the future of a specific business is more difficult in Europe.

One client owning plants in three countries and distributing his products in six countries, prepared forecasts for all those countries, and in spite of the incompleteness of the data he was able to secure fairly reliable forecasts of changes in business conditions.

Results

In order to illustrate the results obtained, other than in increased production or in savings, by a typical installation, part of an address delivered before the Engineers' Association of Warsaw by Mr. J. Zaporksi, Technical Director of Lilpop, Rau & Loewenstein, a company with 3000 workmen which builds railway passenger and freight cars, is given below. Mr. Zaporski says:[3]

One of the features of the new method as a whole and in detail is a considerable clearness and a simplification of work in all its phases. The installation was begun from the last phase of manufacture, the erecting shop, and this has brought about its penetration to all our needs which actually derive from that shop and are dictated by it.

The benefits obtained are general. The production has been entirely mastered, while the normal management duties connected with this production have been mechanized, that is, brought to quite determined forms and phases which put clearly defined responsibilities on the shoulders of each of the executives, and, in case of divergences from the normal flow, allow the fixing of responsibility for each of them.

By this mechanization of every-day duties, the initiative of managers and subordinates has in no way been limited; on the contrary, executives have today more time to study the improvement of production, since most of their every-day troubles have been taken off their shoulders and placed on those of less important men by means of planning.

A proper and accurately defined division of responsibilities between members of the management and employees and a strict control of the timely execution of the duties of each of them obliges them to bring to attention all difficulties which they meet in their work, which causes an automatic improvement of the whole organization.

In closing I wish to express some of the general impressions on the work done.

The entire personnel of our plant has become deeply interested in the new organization. It has penetrated into every part of the factory life, leaving everywhere its mark. Naturally there were some difficulties which, however, were easily overcome and we can say that the better a man understood the spirit of the methods and the more deeply he penetrated into those ideas, the more eager he became to follow them and the better results he obtained.

The organization breaks no one's individuality and harms no one; on the contrary, it develops personal efficiency, representing his progress in units which are easy to measure. Clear-cut duties and responsibilities allow each member of the organization, irrespective of the work he performs, to show his own accomplishment.

All unnecessary ballast, which appeared to be useful but in reality obscured the clearness of things, was removed. A sound foundation of management was created, that is, a plan thoroughly thought out and made in advance for all work. The principles of fair play are apparent everywhere; work is made easier, conditions of work are improved, and earnings are increased.

Progress of Management in Europe

In every large European country there is a national organization for the advancement of the science of management. These groups have received financial support from their respective governments, which have been anxious to obtain the advantages of good management in their struggle for export markets. The work of collecting information and broadcasting it to technicians, managers, and the public at large, as well as the translation of books and papers on management subjects, has been done by these national groups.

The International Management Institute in Geneva has shouldered the difficult task of coordinating the work of these national organizations and stimulating the free interchange of information among them and the American groups.

Four international management congresses have been held in Europe, presided over by such statesmen as Mazaryk, Mussolini, and Tardieu, and attended by thousands of delegates who have gone back to their homes in every industrial country with an added realization of the importance of management.[4]

At the present time there is in Europe a more general understanding of the fact that conditions are changing. Industrialists are realizing that the acceleration of change which has taken place since the Great War cannot be expected to slow down. They are therefore reorganizing their businesses, not only to fit changed conditions but to anticipate further changes. They are seeking more flexible methods, more effective planning for the future, running comparisons of what actually happens with those plans, and are training executives to see the tendencies thus revealed and take the action necessary to get the best results from those changes.

Technological Unemployment

In Europe more consideration is being given to technological unemployment than here at home. Labor is less mobile there because it is more attached to the land, and also because jobs are more difficult to find and therefore losing one's position is much more serious than here.

In many cases the head of a business gives his personal attention to providing positions for the workers who are displaced by labor-saving machines or processes. He does this in a variety of ways: by passing a part of the savings due to the new machine on to consumers in the form of lower prices in order to increase the volume of sales; by improving quality; by extending the service to consumers; by more intensive advertising; by taking on additional lines of manufacture, and sometimes by building entirely new plants to absorb the workers released by the introduction of new equipment or methods.

In those countries where there is a good deal of unemployment due to economic conditions, the introduction of better methods is particularly difficult, but it must go on because the high costs of production and ineffective sales methods have been largely responsible for the gradual decrease of business and the consequent increase of unemployment.

Management Helps the Worker

The introduction of the new management into European plants has almost invariably brought about better working conditions,

[3] See Appendix No. 4.

[4] In order to give a more complete picture of the progress of management in Europe, brief reports from leading industrialists in nine countries, summarizing what has taken place there and pointing out present tendencies, follow as appendixes to the paper.

higher wages, and steadier employment, just as it has in America. It is fortunate that there is on both continents a growing recognition of the worker. It is particularly easy to see this in some of the older countries where there are so many reminders of the early stages of the industrial era, when a worker's time, health, and even life itself were held so cheap. The long hours he worked, his miserable wages, the inhuman punishments he suffered for any infringement of the rules, and the conditions under which women and children were employed, seem incredible today.

The general welfare and prosperity of a nation appear to go hand in hand with the better appreciation of the worker. A realization of his importance as a consumer has done much to bring him better wages and a higher standard of living, which in turn has helped general prosperity. His influence as a shareholder is also beginning to be felt, and in many plants on both continents he has a good deal to say about management policies.

In Russia the satisfaction of the worker is regarded as one of the prime objects of industry. This year, when asked by the Soviet Institute of Management Technic to develop and install management methods in an entire branch of industry, the author was requested to keep the following objects in mind:

To provide safe and healthy conditions for the worker;

To remove from his path the obstacle which prevented him from doing good work;

To enable the worker to understand his part in industry and to have maximum freedom for self-expression.

This is of course a statement of aims and these conditions have not yet been introduced into industry; however, when her plans have been carried further, it seems quite possible that Russia may make a major contribution to the welfare of mankind by developing the worker and improving his position in industry, and, even though we may not agree with her political dogmas, the Western world should not hesitate to learn as much as it can from this gigantic experiment.

One of the greatest services the management engineer can render is to make existing knowledge and method available so that constructive minds will not need to go over ground which has already been covered: to simplify the routine so that it will require a minimum of time, and in other ways to free the creative energies of executives and workers.

The pioneering work of applying science to management has been done, and the mechanisms or technic of management have been developed. Much will be done in the future to improve this technique and extend its application to new fields, but the task of the management engineer now is to place this science of management at the disposal of every part of industry and then to concentrate his energies on developing the individuals and stimulating their creative brains.

Conclusion

When we read the reports of disarmament conferences, hear the threats and counter threats of government spokesmen, and watch the frantic building of tariff walls, the picture is discouraging.

The workman on the frame of a fifty-story skyscraper knows that his safety depends on his ability to focus his eyes and concentrate his mind on the steel beam immediately before him. It is the same with the engineer—he does not allow himself to be distracted by other things or to worry about the complexity of the maze about him. He takes one thing at a time and organizes that to the best of his ability, and then goes on to the next, building carefully, even if the extent of his work seems small.

The engineer is busy with his plans and charts, his records and forecasts, but nevertheless he is sensitive to what is going on around him and the changes which management is bringing about. He sees it provide machines to take the burdens from man's back, allowing him to use his mind rather than his animal strength, and to free himself from poverty. He sees industry teaching men to understand more clearly their aims and to cooperate in reaching those objectives; fair play becoming a daily habit rather than a text for a sermon; the authority of leadership going to those who are capable of leading without regard to class lines.

The engineer sees executives imparting to others the knowledge which they formerly kept entirely to themselves, and whole groups of industries overcoming their suspicions and exchanging information for mutual benefit. He sees commercial agreements being made which disregard boundaries and forget the fears and hatreds of the past. He hears the shouting of warriors and idealists exhorting people to follow them to freedom and happiness, but as he goes on with his work, he watches the machine age and the new management breaking down barriers, healing old wounds, wiping out poverty, and giving men more freedom to develop themselves and to prepare the next generation to lift higher the torch of progress.

Appendix No. 1—Management in the U.S.S.R.

By WALTER N. POLAKOV[6]

TODAY scores of leading American engineering firms, from General Electric to MacDonald Engineering Co, from Du Pont de Nemours to General Motors, are doing construction and equipment-installation work in the Union of Soviet Socialist Republics in the usual manner. The installation of management-engineering methods presents, however, many unusual peculiarities due to local specific conditions, the most important of which are:

1 The entire national industry is run strictly according to the plan prepared by the State Planning Commission for a 5-year period; everything from the production of fuel and steel ingots to fishing and fur trapping is so preplanned.

2 Industry throughout being run not for profit but for utility, its aim is to make goods in the quantities required and of qualities and at costs prescribed by the plan.

3 Industry is owned and managed by the workers themselves, and very inadequately directed by managerial staffs of limited experience.

4 Every department of every plant or factory is operated under a strict budgetary plan, as if an independent enterprise, with a foreman as the sole responsible head.

For these four main reasons, the former enthusiasm for "Taylorism" in planning and functional foremanship brought no results, because while satisfying condition 1, it was appallingly inoperative under condition 3, and not well adapted to condition 4.

For years, as well as at present, every industrial establishment was required to have an efficiency department (Rationalization Bureau) whose duty was and is to devise means for increasing production and reducing costs. Since mechanical ability is more readily found than managerial talent, most of the work was limited to the development of jigs, simplified operations, and the like in mechanical devices, all record keeping and organization problems being neglected.

[6] Consulting Management Engineer, Walter N. Polakov & Co. Inc., Moscow, U.S.S.R. Mem. A.S.M.E.

The recent popular fervor to accomplish the 5-year plan in four years disclosed most convincingly the managerial shortcomings, proving the old truth that good organization avails far more than perfect equipment, and that sound management yields more than large plants.

Under these conditions, especially in view of the dire shortage of managerial ability, it was inevitable that the principles of management as laid out by Gantt, including his charting mechanism, should become indispensable for the proper running of that great national industrial machine of the U.S.S.R.

Layout charts meet the need of condition 1, idleness charts help materially to satisfy condition 2, and progress charts and man-record charts assist greatly to live up to the task imposed by condition 3.

Budgetary departmental and plant control is adequately guided by the Clark's executive control charts, which offer even to inexperienced foremen and other executives a ready guide by means of which they may cope with budget and production programs.

Inasmuch as standards once adopted in the U.S.S.R. become industrial laws enforced by the state, the managerial methods and mechanisms which produce results also become laws.

Appendix No. 2—Scientific Management in Holland

By C. H. VAN DER LEEUW[6]

NEW methods in management and all questions touching the subject of efficiency are on the whole received with an open-minded interest in Holland. Several firms of consulting engineers are putting their services at the disposal not only of manufacturers, but also of organizations of public utilities and the like.

The Netherlands Institute for Efficiency, together with the Netherlands Institute for "Documentatie en Registratur," are holding yearly congresses at which different ways of improving efficiency, possibilities of rationalization, methods of planning, of budgetary control, etc., are treated in several short papers and in the discussions which ensue.

Contrary to practice in the United States, manufacturers are apparently reluctant to publish anything regarding steps taken in the new direction, so that many most efficient systems are being put into effect which never come to the attention of the public at large.

Another thing found frequently in Holland is the tendency of manufacturers to solve their own problems without calling on outside help. In fact, many would rather attain 80 or 90 per cent efficiency with their own staff than 100 per cent efficiency with outside specialists. A certain amount of secretiveness also keeps many people from calling on outside help.

Some of these tendencies are sound enough, provided they do not result in necessary improvements being left undone.

There is a trait in their character which makes the Dutch want things to be shipshape and run in the proper way without much talking about it; while this certainly opens the door to modern methods, it nevertheless renders communications as to results obtained rather scarce.

The relation between employer and employee is on the whole one of mutual understanding, making possible the cooperation of all in introducing new methods—a cooperation essential to attain ultimate success. The difficulty experienced here as elsewhere is in the following-up procedure after the system has been installed.

[6] Director "De Erven de Wed. Ja van Nelle," Rotterdam, Holland.

Appendix No. 3—Swiss Societies Devoted to Scientific Management

By ALOYS HENTSCH[7]

THE Commission de Normalisation de la Société Suisse des Constructeurs de Machines was created in 1918, and in the following year became the Association Suisse de Normalisation. The Bureau de Normalisation du V.S.M., which acts as a central office of this association, has achieved notable results in different fields, both from a national point of view and by contact with foreign organizations which deal with standardization, and it has published, among others, standardization tables which are current in Swiss industries.

The SFUSA (Schweizer Freunde des U. S. A.) organized in November, 1925, and February, 1926, a series of lectures on rationalization. Following these lectures, a commission was set up, the Kommission für Rationelles Wirtschaften (K.R.W.), designed to further measures helpful to rationalization in Switzerland. This commission organized a series of national and international lectures, whose success was noteworthy. It also installed a library and a lecture hall, and a number of pamphlets relative to rationalization were published under its auspices. Its efforts to create an "Association Suisse de Rationalisation" resulted in setting up a group of this kind, whose future is still uncertain.

The Union Centrale des Associations Patronales Suisses, at Zurich, after a conference in the Federal Polytechnic School with Mr. H. S. Dennison in 1927, inaugurated groups for research and information, now three in number. Group 1 is composed of seven large non-competitive businesses employing at the beginning 35,000 workmen and clerks. Group 2 is made up of twelve medium-sized metallurgical and engineering businesses, and Group 3 comprises nine medium-sized textile enterprises, the two together employing 9000 workmen and clerks.

The Société Suisse des Constructeurs de Machines established two groups for research and exchange of information which are at present attached to the Institute for the Rational Organization of Industrial Exploitation, established at the Federal Polytechnic School and dealing, under the first group, with seven manufacturing businesses of various kinds employing 100 to 150 workmen each, and under the second group with six metallurgical and engineering businesses, each employing from 500 to 1200 clerks and workmen.

In January, 1928, the Commission Romande de Rationalisation was founded in Latin Switzerland. Its principal aims are:

1 Coordination of all efforts relative to scientific management in Latin Switzerland.

2 Contact with similar institutions in national or international fields with a view to exchanging information and to concentration of research.

3 Study and dissemination of methods giving the maximum of efficiency in production and distribution.

4 Anti-waste campaigns, whether of raw material, machinery, or man power.

5 Study of "human factor" problems concerning work, hygiene, safety, insurances, industrial relations, etc.

6 Propaganda for rationalization: the principal aims being the lowering of costs and sales prices, and the increase of output, production, consumption, and profits.

7 Office for documentation and information relative to scientific management.

8 Study of specialized technical problems directly concerned with scientific management.

[7] Director of the Commission Romande de Rationalisation, Geneva, Switzerland.

Divers sections were planned (industry, agriculture, arts and crafts, commerce, private administrations, banks, and public establishments). One of these, the Industry Section, got to work at once and instituted a group for research and exchange of information, which now comprises twenty businesses employing a total of more than 9000 workmen and clerks.

The principal questions thus far studied are:

Wage payment	Rationalization of buying
Lighting	Heating
Guides for foremen	Insurance
Industrial accounting	Filing
Preparation of work	

Other organizations are collaborating in the spread of rationalization in Switzerland. First of all, the Institut pour l'Organisation Rationnelle des Exploitations Industrielles (Institute for Rational Organization of Industrial Exploitation), founded at the Polytechnic School, got in close touch with industry, commerce, and government, and devoted itself more particularly to the collecting of Swiss and foreign papers relative to the science of organization; collaborating in the exchange of various kinds of information between different bodies; undertaking certain researches corresponding to the needs of the economic life in Switzerland, instituting lectures and conferences on the science of organization; and issuing various publications.

Finally, certain organizations of a commercial character, as well as a number of independent experts, have also participated effectively in the organization and reorganization of businesses.

The J. J. Rousseau Institute at Geneva has given part of its time to the study of the psycho-physiology of work; similarly; l'ASORT has devoted itself to vocational guidance. The Fondation Suisse pour la Psychotechnique, established in Zurich, (Swiss Institute for Psychotechnic) has opened a number of institutions of psychotechnic to which employers (in industry or public service) frequently apply. In this field, other organizations, among them the Association Suisse pour l'Orientation Professionnelle et la Protection des Apprentis (Swiss Association for Professional Guidance and Protection of Apprentices), have attained noteworthy results by different methods.

As regards teaching, lectures devoted to the methods of scientific management and allied problems are given in the technical schools and certain universities.

In business, the organization of groups similar to the retail research associations is being considered; centralized purchasing already exists, as does also chain-stores distribution. The professional associations could serve in certain cases as groups for the exchange of information.

The Swiss Office for Commercial Development, with branches in Zurich and Lausanne, affords to business men, and more particularly to importers and exporters, much valuable help.

Finally, the public utilities, the CFF (Federal Railways), PTT (Post, Telegraph, and Telephones), etc., are applying more and more scientific management in their respective fields of action.

Studies have also been made by different bodies on questions of national welfare, especially in the field of apprenticeship, insurance, crafts accounting, building, commerce, and banking. Professional associations as well as employers' associations or syndicates are also studying questions relative to scientific management.

Exhibitions, some held occasionally such as the Schweizerische Austellung für Frauenarbeit (Swiss Women's Exhibition), others periodically such as the Foire d'Echantillons de Bale ou le Comptoir Suisse de Lausanne (Sample Fair at Bale or the Swiss Counter at Lausanne) make some contribution to the study of certain problems. Finally, efforts are being made to insure a certain amount of coordination in the clock industry, and in banking. In industry, individual endeavors proceed as circumstances offer, and within the limits possible under the particular economics suitable to Switzerland, which is a small country without raw materials, but with skilled labor, which has to devote its whole attention to exportation. Banks have also perfected their individual methods of work.

In Switzerland, rationalization has advanced more on the technical side than on the side of sales. Industrial agreements have as yet no very large place in the scheme of things (cf. Federal clock industry), but the research groups would seem to have a bright future before them.

Appendix No. 4—Results of Reorganization of Plants in Warsaw, Poland

By J. ZAPORSKI[8]

PRODUCTION in the plants of Lilpop, Rau & Loewenstein has, during recent years, been limited almost exclusively to the erection of various types of railway cars. The entire production has been concentrated on Bema Street, in the suburbs of Warsaw.

In May, 1920, a contract was made with the Ministry of Railways for the supply of rolling stock to the Polish State Railways, and the gradual rebuilding of the factory for that purpose was begun. In 1922 the first post-war freight cars were made, and in 1923 the first new passenger cars. Primarily the real work of the factory was the repairing of cars—in fact, their complete reconstruction. However, all endeavors of the management of the plant were now directed toward the building of new cars, and finally in 1928 the factory discontinued accepting cars for repair, as it was becoming more and more difficult to combine repairs with the constantly increasing number of new cars erected.

In the meantime a number of subsidiary shops were installed. It would be difficult to find more troublesome work than that of the erection of railway passenger cars. There are many different craftsmen needed in building a passenger car—blacksmiths, tinsmiths, boilermakers, hydraulic workers, glassmen, electricians, carpenters, furniture makers, upholsterers, painters, varnishers—while the number of operations in the erection of a car amounts to over 700, and there is an immense number of different parts. It is obvious that the organization of such a complicated task is not easy.

In 1928, in order to bring this production under the control of scientific management, which is the only rational method nowadays, the company sought the assistance of Mr. Wallace Clark, an American engineer well known for the exceedingly good results he had obtained in such work. Mr. Clark, together with Mr. A. Kucharzewski, started work in June of the same year.

Planning Work Ahead

The initial attack was made on the final phase of manufacture in the erecting shop. The first step was to show each foreman in this department the necessity for planning work ahead for the following day and for checking later to learn whether the work thus planned was actually completed. From that moment the foremen began to acquaint themselves with the graphical planning of work on Gantt charts. It was naturally difficult for a foreman, having his men scattered in small groups in several cars, to keep well informed as to every detail of the realization of the plan, especially since his main duty had always been to see that the work was well and promptly done. Therefore the preparation of the daily graphic schedules was delegated to planning clerks who were chosen from the shop inspectors. Of course the planned

[8] Technical Director, Lilpop, Rau & Loewenstein, Warsaw.

progress of work depended first of all on whether materials and tools were available for the operations in sequence, which the foremen were obliged to investigate beforehand.

The technique of making these plans was mastered fairly easily, and the foremen with the planning clerks began to teach the men to follow the plans, seeing at the same time that materials and tools were available when needed.

At the same time it became evident that much time was being lost by the workmen in getting the necessary materials and tools from the storesroom, and as a consequence this task was immediately delegated to unskilled workmen, while the duty of making out stores issues and other clerical work was delegated to planning clerks, so that the foremen could give all their time to their proper duties, such as supervision of the quality and prompt execution of work by the workmen, spending only an hour in the afternoon with the planning clerks on the preparation of the plan for the following day.

Due to the fact that the work in car erection had been divided into large units of time, covering groups of cars and ranging from 100 to 900 hours in duration, the preparation of daily plans at first met with difficulties, as these long units were not suitable for planning work day by day. The erection was therefore subdivided into smaller elements not exceeding 8 hours each. The succession of separate operations was revised and the operations were arranged so that they should not interfere with one another through an excessive number of men working on one car at the same time.

A study of the daily plans of the various foremen revealed differences in the sequence of separate erecting operations, and therefore a standard schedule was developed for the entire erection with the object of harmonizing the daily plans of separate erecting groups, such as carpenters, blacksmiths, painters, upholsterers, etc., so that one group should not interfere with the others.

The aim of this central planning in the erecting shop was to collect accurate data on the time needed for the separate elements of the erection and to prepare a general schedule of erection. The order of work made on that basis gave automatically the date of completion of the erection of each car. This further made it possible to simplify the making up of daily orders of work. The general plan of work gave also a true picture of how much time and how many men are needed for a complete erection of a car.

With such a schedule, it is possible for the head of a department to follow the progress of work with the greatest accuracy, and to learn the causes of delays and to eliminate or greatly reduce them by making rational alterations in the daily orders of work. The Central Planning Office, coordinating the progress of the general plan with the daily orders, could also determine proper dates of delivery of parts from other departments and the storesroom, which would result in a more even loading of other departments and a considerable decrease of stocks as they ceased to be overloaded with products which would remain for weeks on shelves without being used.

Endeavors to adhere more and more strictly to erection schedules showed that a large number of operations could be simplified. An Improvements Section was therefore created for studying all erecting operations with the view of simplifying them. Its procedure consists in accurately describing the existing method of work, and later in making recommendations as to simplification of separate processes, showing the savings in time which can be secured.

Improvements of Operations

It would seem that the erection of cars exclusively by hand by skilled workmen would be the best way to do it, and that a non-specialist would not be likely to improve upon that method. Yet, in reality, quite the opposite is true, and each study gives positive results, the whole giving very large savings. In order to illustrate the value of studies and improvements the writer will cite an example. The erection of the floor of a car took 188 working hours as per the schedule of the Technical Bureau. After a study was made and the existing method described, the latter was changed, and as a consequence the floor is now being erected in the erecting shops in 18 hours, while the increase in work in other departments due to the change amounts to only 51 hours, which means a total of 69 hours as compared with the previous 188 hours.

The complete erecting time of the Westinghouse brake has been decreased from 148 working hours to 71, and as a result a second-class passenger car is now being erected in less than 90 days as compared with about 180 days formerly, while a third-class car is erected in 60 days instead of about 130. It should be noted that the premiums earned by workmen have considerably increased in spite of the fact that labor costs per car have greatly decreased, this being due to eliminating the time wasted on unnecessary or insufficiently studied operations.

Working Conditions

Apart from the improvement of operations, general conditions in which the work is carried on such as floors, methods of handling materials, light, heating, ventilation, and sanitary equipment have a great influence on the decrease of cost of production.

It is a mistake to think that it is possible to produce cheaply under unsatisfactory conditions. Good working conditions are much more important for a man than for a machine. Our advisers have given much care and attention to the improvement of working conditions in the shops and of interdepartment and shop transport.

A layout or floor plan of each shop was made providing good, smooth roads for transport, places for storing materials before and after machining, lockers for workmen's clothes, toilets, etc. As a consequence a larger output per workman has been secured due to better conditions and easier supervision and a large reduction in the cost of transport, while the number of workmen engaged in transport has decreased from over 80 to about 30, and the investments made in materials-handling equipment have already largely paid for themselves.

Costkeeping

One of the most efficient and accurate tools for controlling the activities of superintendents of the various producing departments is a rational method of costkeeping. The method generally used does not provide sufficiently accurate figures for a comparison of estimated costs with the actual costs, due to the fact that it does not take into account the degree of utilization of the capacity of the shop.

The modern method applied in our plants introduces a so-called "idleness" account to which is charged a certain part of shop expenses which should not be charged to production account, due to non-usage of the full producing capacity of the plant. When this account is introduced, the actual costs correspond to the estimated or budgeted costs provided the budget was prepared correctly and the management directed the execution of a given order in a sufficiently efficient way. In case of disagreement, an investigation is made to determine which of the two is at fault and who is responsible for the state of things. One of the advantages of this method of costkeeping is that it affords the possibility of having a statement of costs completed three to four days after the finishing of an order. This enables the management to intervene promptly in case an increase of costs is noticed, and is exceedingly valuable for the management.

The above method requires considerable preparatory work, and therefore our advisers have from the very beginning of their work divided all costs chargeable to ownership and maintenance of grounds and buildings among the separate departments. Each department has then been charged with costs of supervision, lighting, heating, etc., on the basis of the square meters of floor space occupied. The cost of each square meter of floor space has been further charged to the area occupied by a car or by a machine or working place, according to the character of the shop. By adding to these costs the expenses of owning and maintaining a machine or a group of machines, we obtain the shop expenses to be charged to separate machines or working places during the year. This amount divided into the number of working hours in the year gives the cost of a machine-hour or the rate for a working place. The product is then charged with these costs multiplied by the number of hours of work of a machine or working place in the erecting shop. In the case of our plant this is for one car. The difference between the former method and the present one is that the product was previously charged with shop expenses in proportion to labor costs, which is the usual method in most plants. If we analyze the former method carefully we see that it is far from being right, as shop expenses do not depend on labor costs but only on the above-mentioned facts. The evident results of the new method of cost keeping are:

Results of New Method of Costkeeping

1 They make precalculation easy and accurate, as the accountant uses figures of materials, labor, and machines and work places already decided upon.

2 When an order is completed, comparison of actual figures with those of the budget is easy, no unforeseen increases of shop expenses being possible if the work is done in the time and under the conditions stipulated.

3 It is now possible for the management of the plant to secure the true cost of each order irrespective of the load on the shop, which is extremely important from the point of view of control of preliminary calculation, and from the point of view of the balance sheet of each separate order and the activity of the shop superintendents.

4 In quoting prices to buyers the method is exceedingly valuable with the present keen competition, as it shows to what extent prices can be decreased in an effort to diminish the losses due to idleness of the plant.

The technique of keeping the new costs up to date requires that the costs incurred each day by a given order be communicated to the Cost Department as soon as possible, that is, within 48 hours. In connection with this the workmen's earnings are calculated each day on production cards and sent the following day to the Payroll Department, whence they go one day later to the Cost Department. Production cards provide a very strict control of production and of the presence of workmen in the shop.

Storeskeeping

The reorganization of stores consisted first of all in the physical arrangement of materials, that is, in the application of a method of storage which gives a most economical use of the available space and easy service to the stores and shops.

With that object in view a rack consisting of 6 or 8 divisions was designed, each of which was provided according to the size of material with 2, 4, 8, or 16 exchangeable bins, this insuring a very easy handling of these bins. At the same time the bins used by the shops were standardized to make it possible to exchange an empty stores bin for a full bin brought from the shop. Due to the installation of this system the erection of a new stores building was found unnecessary.

Much attention has been paid to the decrease of the investment in stock, at the same time keeping in stock all materials needed to insure a normal flow of work through the plant. Strict adherence to the principle of maximum and minimum resulted in a decrease of the value of stocks by 5 per cent and an increase of production of 27 per cent. A continuous method of control of stocks has also been applied which enables all errors to be discovered every two or three months and not after a year has elapsed, as is usually the case.

The handling of the paper control of the stores has also been considerably improved by eliminating all those operations which have only had the appearance of a control of stocks, and in fact have only made the matter more complicated; a clear and accurate control has thus been obtained with a decrease in the personnel.

Conclusion

In the foregoing the writer has endeavored to present the salient features of the newly installed method of management without going into such details as forms, blanks, charts, etc., which would take too much time and, for that matter, are peculiar to our special work. However, it should be stated that one of the features of the new method as a whole and in detail is its clearness and the simplification of work in all its phases which it effects. As mentioned at the beginning, the installation was begun at the final phase of manufacture, the erecting shop, and from there it has penetrated into all our activities which actually derive from that shop and are dictated by it. The benefits obtained are general. Production has been entirely mastered, and the normal management duties connected therewith have been mechanized, that is, brought to forms and phases which put clearly defined responsibilities on the shoulders of each of the executives, and, in case of divergences from the normal flow, allow the fixing of responsibility for each of them.

The initiative of managers or subordinates has in no way been limited by this mechanization of every-day duties; on the contrary, executives have today more time to study the improvement of production, since most of their every-day troubles have been taken off their shoulders and placed on those of subordinates by means of planning.

A proper and accurately defined division of responsibilities between the members of the management and employees and a strict control of the timely execution of the duties of each of them serves to bring to light all difficulties which they meet in their work, the result being an automatic improvement of the whole organization.

The entire personnel of our plant has become deeply interested in the new organization. It has penetrated into every part of the factory life, leaving everywhere its mark. Naturally there were some difficulties at first, but these were easily overcome and it can be said that the better a man understood the spirit of the methods the more eager he became to follow them and the better results he obtained.

The organization destroys no one's individuality and harms no one; on the contrary, it develops personal efficiency, recording its progress in units which are easy to measure. Clear-cut duties and responsibilities allow each member of the organization, irrespective of the work he performs, to manifest his accomplishments.

All unnecessary procedure, which appeared to be useful but which in reality obscured the clearness of things, was removed. A sound foundation of management was created, that is, a plan which was thoroughly thought out and made in advance for all work. The principles of fair play are apparent everywhere; work is made easier, conditions of work are improved, and earnings increased.

Appendix No. 5—Application of Modern Management Methods in France

By SERGE HERANGER[9]

FRANCE has not been behindhand among European countries, especially during the last decade, as regards the application of modern methods of industrial organization, although the progress that is being made is perhaps not as rapid as might have been expected, considering that France was one of the foremost countries in Europe to interest itself keenly in scientific management at the time when Taylor and Gantt were proceeding with their pioneer work in the United States.

This is due, therefore, not to a lack of interest in modern methods of management, but rather is a consequence of the mentality of the French. France is an old country with old traditions, and the French executive is apt to be overprudent and cautious with regard to new ideas and new methods, and hesitates to invest his money in reorganizing his plant. This is a psychological problem that the American engineer must carefully bear in mind when he is consulted by French executives. Another important factor is that the current opinion in France couples American industrial organization exclusively with mass production, and a question put very often is whether modern American methods can be applied to the small-series or irregular production work more frequently done in France.

In spite of this, there has undoubtedly been a steady progress in the application of new management methods to French industries. The phase of management which has been more particularly developed is that relating to plant production, and there are now many plants that have a more or less elaborately organized planning and scheduling department, modern storekeeping and cost systems, etc.

Office organization has also been well developed. On the other hand, practically nothing has been accomplished with regard to the application of sales organization, distributing, marketing, mergers, etc., although these questions have raised a great deal of interest and there is every probability that in the next few years and independently of the much greater development of plant organization these problems will also be taken up progressively in view of their application to French industry. The great incentive which will insure this and which is always present in the minds of French executives is the unequaled industrial development of the United States.

Appendix No. 6—Scientific Management in Great Britain

By B. SEEBOHM ROWNTREE[10]

THE war and its post-war consequences at one and the same time increased the need for more scientific management in Great Britain and made the attainment of this object more difficult. Britain is more dependent than any other nation upon foreign trade. During the war Britain was compelled to give up a large part of its overseas trade, partly because of the necessity for concentrating upon munition production for itself and its allies, and partly because of the submarine menace. Other nations which had continued to buy from Britain largely as a matter of routine either found other sources of supply or learned to produce for themselves, and at the end of the war they proceeded to protect their new industries by tariffs. For these and other reasons it became imperative that Britain should reduce its costs of production, and in particular, its labor costs.

At the same time, circumstances made this more than usually difficult. In the first place, there had been under the voluntary system the loss of the best part of a generation which should have been contributing enterprise and new ideas to industry. Then industry had to bear the heavy financial burdens left by the war. Moreover, these burdens were seriously increased by the pursuit of a social policy designed to improve the living conditions of the general mass of the people. Thus, either in direct contributions or in increased taxation industry had to contribute toward the cost of unemployment insurance, pensions, improved housing, and so on. At the same time capital was dear and enterprise was compelled to offer attractive terms in order to induce investors to put capital into industry rather than into gilt-edged stocks. On top of all this British producers had not only to face the handicap of falling world prices, but the special handicaps arising from the return to the gold standard in Britain itself.

Meantime wage rates had risen very greatly, and for some years have stood at from 70 to 75 per cent above prewar rates, while in the two years succeeding the war average weekly hours of work in British industry fell by some 10 per cent. While therefore there was obviously the most insistent need for more scientific management in every direction, and particularly in the reduction of labor costs, conditions made it difficult to carry through the necessary changes to the same extent and with the same rapidity as in certain other countries.

It is impossible to state the achievements in any statistical form because the necessary statistics do not exist. There has undoubtedly been a notable increase in the use of payment by results in the engineering and kindred trades. The official view expressed by the Balfour Committee some two years ago was that in the exporting group of industries, those paid by results were not far short in number of those paid by time. These, however, are the industries in which trade unionism is more highly developed, and the probability is that payment by results is much more extensive in the less highly organized industries, and it would not be surprising to find that in British industry as a whole there are now more workers paid by results in one form or another than those paid purely by time.

In certain directions considerable progress has been made in the application of psychology to industrial problems, and in the study of the physiological effects upon workers of methods of work, useful work having been done in these two directions by the National Institute of Industrial Psychology and the Industrial Fatigue Research Board. A good deal of this work is still at a pioneer stage, and many employers have yet to be convinced of its value, but there are prospects of its yielding results of outstanding importance in the sphere of industrial efficiency if it is judiciously pursued, and in particular, if it carries along with it the consent and cooperation of the worker.

In this connection it may be said that great efforts have been made in the post-war period to secure a greater measure of real cooperation between employers and employed. Probably more progress has been made in this direction in Great Britain than in any other. Not only have individual employers gone a long way in the direction of taking their employees into their confidence and in giving them a share in the prosperity of the business in one form or another, but the great organized bodies of employers and employed have been meeting in council on this subject over a considerable period and have achieved some considerable measure of agreement. This is typified by the work of the Melchett-Turner Committee, a joint committee of the National Confederation of Employers Associations and the Trade Union Congress, which incidentally has unanimously recommended the setting up of a National Industrial Council.

[9] Hotchkiss & Company, Paris.
[10] Chairman, The Rowntree Cocoa Works, York, England.

As regards industrial processes, there has been some development along the lines of mass production typified by the motor-car industry, and some considerable progress has been made in standardization and in simplification, particularly under the inspiration of the British Engineering Standards Association, but progress along these lines has probably been less than in some other countries, and certainly less than in the United States. This is no doubt due to some extent to conservatism and to Britain's post-war difficulties, but it is also justified to some considerable extent by the fact that Britain manufactures for such widely diverse markets that standardization can only be carried up to a certain point. The situation is quite different from that of the United States with its great homogeneous home market. As regards equipment, there has of course been a great development in the direction of the use of automatic machinery, but here again this has been limited by the facts just mentioned. In particular the use of automatic machinery is limited by the extent to which the product can be standardized. As regards power, there has been a very substantial increase in horsepower, the figures for 1924 showing an increase of 58.9 per cent over those for the last Census of Production in 1912.

With regard to the application of scientific management in individual factories, the situation is very "spotty." A number of plants can be found in Great Britain which have made an intensive study and an energetic application of scientific methods over a long period. These are comparable with undertakings of equal size in any part of the world, and are well distributed among different industries. On the other hand, there are a large number of small, backward factories which from the management standpoint are still medieval. While there has been notable progress in the last decade, the country has lacked the leadership provided in the U. S. A. by the report on "Waste in Industry" and subsequent work by the Department of Commerce and by private institutions. Consciousness of management as a separate issue, distinct from the technical evidence of this, is to be found in the inadequate volume and character of the management literature. The rapid growth of management research groups, modeled on the Manufacturers Research Association, of Boston, indicates, however, a growing appreciation of the need for scientific study in this field.

Appendix No. 7—Scientific Management in Italy

By DR. FRANCESCO MAURO[11]

THE difficulties special to Italy, among them lack of the most important raw materials, high taxation, and the restricted purchasing power of the internal market, have combined to compel Italian industry to push to the furthest the spirit of enterprise and initiative.

Without considerable ingenuity and keenness to work it would have been impossible to live, even less to develop.

For this all-important reason Italian industry has quite naturally been only too ready to set about studying new methods of scientific management and business administration.

Thanks to the drive and propaganda work of the E.N.I.O.S., the national institute of which the writer has had the honor to be president since its foundation, an even greater development is taking place in the attitude of mind which must exist before scientific management can be applied with good results.

It is necessary to keep in mind several factors which indicate that the problems to be solved in Italy are quite special and sometimes of great interest owing to their original characteristics. Among these factors the writer will limit himself to mentioning:

1 The large population, and therefore the plentiful supply of labor, which means that much account has to be taken of the unemployment problem, technological and otherwise.

2 The relative scarcity of capital, which necessitates making the utmost possible use of existing equipment and the careful study of any change to avoid unproductive expenses.

3 The comparatively small size of many businesses and the fact that they are distributed throughout the country makes it advisable to proceed with caution in order to avoid the social and economic difficulties which would follow a too extensive and hasty relocation of industries.

Changes are now in process and, in spite of inevitable obstacles, are giving satisfactory results which encourage considerable hope that further and deeper changes will be forthcoming.

[11] President of the International Committee of Scientific Management, Rome, Italy.

Appendix No. 8—Scientific Management in Czechoslovakian Industry

By ING. DR. EMANUEL SLECHTA[12]

THE principles of modern works management were known in Czechoslovakian industry in prewar times. However, industry was only induced to study it carefully and to use its methods after the war, because of many reasons which influenced the growth of manufacturing costs. In the first place, Czechoslovakian industry lost a great part of its original home market in former Austria, and had to start exporting more than half of its production (in some cases even not less than 75 per cent). This brought it into competition with foreign countries, for instance with German industry. Furthermore, the effects of tax burdens and debts to the banks dating from the times of inflation should be noted, and finally also post-war social legislation, which because of the dues which must be paid for accident, social, and pension insurances requires up to 12 per cent of the amount of wages paid. Sales conditions have made it impossible for our industry in general to think of mass production. There are of course some exceptions, such as the shoe factory of Messrs. T. & A. Bata, at Zlin, where 120,000 pairs of shoes are now manufactured a day. Management has made an effort to decrease manufacturing cost in all its items—material, wages, and overhead charges—this, of course, without large investments, since the greater part of Czechoslovakian industry is composed of small and medium-sized concerns. (According to the statistics of 1926, the average industrial concern employed 16.5 workmen; there are, however, several works in Czechoslovakia each employing over 20,000 workmen.)

Since in Czechoslovakia, as also elsewhere in Europe, materials used in manufacturing are considered of greater importance than are wages, works managements have given especial attention to economy in the use of matter and of energy, and particularly to heat economy—witness the excellent results obtained by Skoda Works and the Vitkovice Iron Works. Materials are also economically utilized in all Czechoslovakian works, not only as regards raw materials, but also waste products. It may be said that, especially in the regions far distant from the coal-mining districts, heat economy will be striven for still more, since coal represents a large item in manufacturing costs.

As regards the part which wages play in manufacturing costs, it may be said that the modernization of our works was first manifested in the installation of new automatic (labor-saving) machines, used either directly for manufacturing purposes or for transporting and auxiliary work. The war prevented many works

[12] Consulting Engineer, Prague, Czechoslovakia.

from renewing their machinery plant, but after that old machines were replaced by new ones, this being made possible by special exemption. Coal mines, machine shops, chemical works, etc., could be mentioned here in which the percentage of machine work is constantly increasing as compared to hand work. At present this is true of the larger concerns only, but it will soon extend also to small- and medium-sized works, as may be seen from the interest which they are taking in small materials-handling equipment.

The laws in force limit to some extent the use of certain systems of wage payment, so that thus far there has been little or no application of time and motion study in most of the Czechoslovakian factories. However, some of the large concerns have of late given these matters attention, and as a result are getting ready to introduce in their works either the Halsey, Emerson, or Bedaux system of wage payment. This, however, is exceptional, and in the majority of cases day wages based on time and piecework fixed by experience are being paid.

Preparation and planning of work according to the principles of modern engineering practice has been brought to a high degree of excellence only in several of the leading concerns, especially in the engineering line (precision and mass production) and in the chemical and textile industries, which latter up to the present have employed the principles of modern management to the greatest extent. Future development will be in the direction of establishing planning departments, as may be seen, for instance, from the interest—especially of the medium-sized concerns—taken in papers presented on the subject.

Control of the works on the part of the management is effected in the majority of cases by periodical statistical reports. Only a few leading factories have introduced the modern graphical control system, and these only because of the pressure of the economic crisis.

It is evident from what has been said that the lack of preparatory and planning work is responsible for a lack of documentary data for use in calculations. It is apparent that the calculation of costs and prices is another question which is receiving much attention.

As for the Czechoslovakian factories, it is necessary to bear in mind that while there are of course very modern concerns complying with all requirements of work, in the majority of cases old works are involved, and therefore also old buildings, which only after the war were slowly adapted to the changed conditions. For this reason it is difficult in the old works to adjust the working conditions to the modern requirements of manufacture, i.e., arrangement of the machines and plant in the workshops, departments, etc., so that this circumstance should be taken into consideration.

A greater advance may be seen in the organization and management of office work and offices.

On the whole, one cannot speak of an intensive use of modern principles of management in the Czechoslovakian industries, the few large and medium-sized concerns in which they are employed being exceptions. However, the era of more scientific management will start with the arrival of the younger generation of engineers in the field of works management.

Appendix No. 9—Scientific Management in Poland

By CZESLAW KLARNER[13]

WHAT methods have you used to increase the productivity of those enterprises which you are leading?

One of my first aims has been to introduce methods of scientific management. In this I have had as a goal the bringing of waste under control and the putting of the particular business dealt with on a basis of right management. In this respect I am seconding the work of American engineers in Poland, helping our industries to make use of their knowledge and experience.

Are there any conditions which you find prerequisite for the introduction of up-to-date methods of management?

For the work of such experts it is absolutely essential that the men at the wheel of these enterprises understand the principles of the new management and are willing to introduce them into daily practice.

What is being done to help bring about conditions favorable for the introduction of better management?

The Institute of Scientific Management in Warsaw, headed by Mr. P. Drzewiecki and Prof. K. Adamiecki, has done a great deal in this direction, and as a result the necessity for scientific management is now more and more penetrating into the minds of leaders of our commercial and industrial enterprises.

Could you quote a specific instance where scientific management has been successfully applied in Poland?

In a company of which I am president and in another of which I am vice-president, together employing about 7000 workers, the introduction of scientific management has effected great savings and at the same time has given higher wages to the workmen. Organization of this kind requires a larger number of brain workers than formerly in order to deliver the task to the workman in a prepared and well-digested form, so that he will not lose time in overcoming difficulties which prevent his progressing with the job.

Do you find any differentiation necessary in Poland in applying these methods?

We must remember that under Polish conditions it is not possible to copy blindly American examples, because the costs of money and of labor in Poland differ entirely from those costs in America. We Poles must use good judgment in applying those methods, especially in costly labor-saving investments.

What do you think of the possibility of improving the management of state-owned enterprises and increasing in this way the budgetary income of the Polish Republic?

When private enterprise has accepted on a large scale the right kind of organization, the state will also become interested, and will apply the same methods to its enterprises. The principles of scientific management are well adapted to certain kinds of enterprises conducted by the state, such as cigarette factories, coal and salt mines, etc.

Appendix No. 10—New Management Methods in Germany

By DR.-ING. OTTO KIENZLE[14]

BOTH in the metal industry and in other industries in the last 10 years have scientific management methods been extensively introduced.

Much attention is being given to the new accounting systems which provide means for figuring gain and loss by setting up monthly balances. In these the overhead expenses are apportioned systematically and methods are presented for figuring costs with respect to the special conditions obtaining in a given factory. Much attention is also being given to the percentage of full running time of the different shops, and this has led to a differentiation between fixed costs—similar, though not equal, to

[13] President of the Association of Chambers of Commerce and Industry of Poland. Interviewed by Stanislas Lubienski.

[14] Koch & Kienzle, Berlin.

idle costs—and costs varying with the output. In quite a number of places where these methods have been installed the second step in budgeting has been taken.

In connection with this, methods of calculation and time studies should be mentioned. A committee (Reichsausschuss für Arbeitszeitermittlung, abbreviation "Refa") has collected and published in extensive handbooks rules and data for operating times in the metal and wood industries, foundries, etc. Accordingly, we use the so-called "Refa Methods" and have Refa engineers using these methods.

Technically in machine shops the method consists in providing for each part carefully prepared drawings with tolerances, operating plans, and control plans. As to the shops, the machine tools of recent design show great progress; in particular, cutting speeds have been greatly increased. Whenever it has been possible to adopt mass production, a great deal of special machinery has been designed and installed; e.g., the multi-spindle drill press is as well known in Germany as in America. One of the most important differences between a shop of today and one of 10 years ago has been brought about by the introduction of line work and means of transportation. Many important simplifications have been accomplished, and costs have been greatly lowered by the standardization of materials, tools, machines, and machinery equipment which has found its way down to the smallest shops and has consequently promoted specialization of production to a great extent. Internationally, in Europe a number of practical results have already been attained by means of standardization.

Continuously increasing attention is now being paid to the human element. Theoretical methods have been replaced by practical ones. Improvements in the equipment of shops, and especially of working places, have done much to facilitate work and to alleviate the working conditions, and the writings of our leading specialists in management are dealing more and more with these problems.

Discussion

G. S. Griffith.[15] During a three-month inspection of European factories made in the summer of 1929, the writer noted the following:

1 There is a general tendency toward the use of American methods and machinery in their factories. Some of the factories have about 40 per cent of our machinery, but they have found that this alone does not bring our prosperity. The gradual adoption of the American methods of management is bringing the desired results.

2 Some of the plants are using our progressive assembly methods with very good results. This method was employed in the Renault factory in France. However, the difficulty in employing this method of manufacture in Europe is due to the smaller markets for their products caused by the number of countries and the various tariff walls.

3 The worker's position in Europe in general is more secure than in the United States. The foreman is compelled to give from two to four weeks' notice to the employee before the latter can be laid off. He does not have to keep threatening the worker with the idea of being laid off as is done in the United States in order to hold the employee's respect for the employer. The workman has more respect for his employer principally because of the class distinctions which are so prevalent in Europe.

4 The layouts of the plants are being changed to permit the better arrangement of the machinery in order to increase production. The machines are placed in groups or run by individual motors, which does away with the overhead drives. Lighting conditions are being improved, and more thought is given to sanitation and better working conditions for the employees.

5 Storerooms are being centrally located and American methods of stockkeeping are being adopted, which has greatly helped in the matters of inventory and stockkeeping. It is necessary for the European concerns to keep larger inventories because many of the countries are not self-contained, and in order to get parts, some of which come from other countries, a longer time is required than in this country.

6 The wages of European workers are small compared to those of the American worker, but the former do not seem to require more money due to the standards of living not being the same. What the American calls an absolute necessity the average European worker would call a luxury. Do not think, however, that the writer means if their wages were higher that their standard of living would not be higher. The industries over there are placing many of their workers on the incentive-payment plan, which has increased the production.

7 The apprentice system is still in vogue in Europe, although it is dying out in our country. There are three classes of apprentices: the "regular," who applies to learn a trade but does not pay for the privilege; the "premium," whose parents pay to have him learn a trade; and the "pupil," whose parents pay in order to have him trained to become a foreman or an executive. The apprenticeship usually requires from 3 to 5 years.

8 Some of the employers are helping their employees to gain a better education by educational programs. Some of them have established training classes and schools for the workers, but not to the extent that our industries have done in this field.

9 The working hours are being lessened as the employers realize that long hours harm the employees. The average working day is from 8 to 10 hours. More and more the idea of giving vacations to the employees is gaining ground. Some of these vacations are with and some without pay, and vary from 1 to 3 weeks in duration, depending upon length of service and the position held.

10 Measures are being taken to prevent accidents. Guards and safety devices are being applied to the machines, and campaigns against accidents are being instituted. In Germany some of the industries are effectually working along this line and use many posters to illustrate their points.

11 Some of the industries have systems to help the employees save a percentage of their wages, as well as accident and unemployment insurance and old-age-pension plans. In a number of the European countries the governments now take care of men out of work and provide for old-age pensions.

In conclusion, the writer would say that American management policies are taking root in European industries and that the latter are showing a gain as the result of adopting these methods.

Lillian M. Gilbreth.[16] This Society and all engineers in the management field are fortunate to be represented abroad by the author, who interprets Europe to us and us to Europe.

The paper furnishes, tactfully but explicitly, many lessons to engineers. Some of these come through advice on how to handle European problems. The author says that the former emphasis on machines rather than on management has shifted. This means that we must be prepared to send management *with* machinery or perhaps even management *before* machines. And the same thing holds true of mechanism. He says that while mass production is increasing, many industries still handle small diversified lots. This means that our management, mechanisms, and machines must be adapted to the small lot.

The author notes a difference in tempo in Europe from what we have here—a slower speed. This affects both machine speed and man speed—and we should have the matter in mind as we

[15] Roselle, N. J.

[16] President, Gilbreth, Inc., Montclair, N. J. Mem. A.S.M.E.

send over machines and equipment, as well as the men who will sell, install, demonstrate, and perhaps maintain them.

The author notes the distinction between the theoretical and the practical man—which implies the need of our men knowing both theory and practice and being able to present either to any group—if they are to retain leadership. Our engineers must not only be prepared to demonstrate what they say, but to persist even though they find such demonstrations greeted with astonishment.

The greater lag between being convinced by the theory and initiating the practice which the author finds abroad, implies that our engineers must be ready and able to push things across against both inertia and custom.

And so on through the entire paper. The differences which the author has observed, reported, and explained are regarded by him not as hindrances but as challenges—and challenges which he shows us how to accept and meet successfully.

Much of the author's material can be used to cope not only with situations abroad, but here. This he makes clear, for he recognizes not only the differences but the likenesses. Many of our workers, plants, and industries are more European than American, even yet, and his techniques of recognizing and handling these are most serviceable. Others which might be rated by some standards as typically American in origin, organization, and efficiency, are European in psychology. And the writer means no criticism of such plants—as Mr. Clark means no criticism when he says "European . . ." Through them all exist identical problems, i.e., of measurement, standardization, and teaching—which make the author's varied and extended experience invaluable to every one ready to use it.

Recent observations in Europe, Asia, and this country by other engineers who specialize in management all confirm the author's findings—and emphasize the value of his paper.

FRANK M. RYAN.[17] The author has given us a very clear picture of how European executives react to American management methods, so it may be of interest to examine the reaction among the workers.

Class consciousness manifests itself in many ways and sometimes acts as a serious handicap to the increase of individual production, thus delaying the accomplishment of the management engineer's task.

In practically all European countries the Socialist party is a strong factor in politics. As a result, social-insurance laws are common and the worker is accustomed to seeking financial aid from the government for childbirth, sickness, old age, unemployment, etc.

This influence tends to dampen individual initiative and the desire of the worker to attain the maximum production of which he is capable. It leads him to consult the government agencies on many problems which the American worker handles individually or upon advice from the employee relations department of the plant in which he works.

The governmental contact, through state-owned railroads, post offices, telegraph and telephone systems, and compulsory military training, in addition to all the agencies of state aid, is a factor in the worker's life which the average American does not appreciate.

This background makes it more difficult to stimulate the cooperation of the workers in better methods of production, but once this has been accomplished, given the same working conditions and a proportionate wage incentive, a good European operator will turn out 60 to 80 per cent of the production of a good American operator.

[17] Works Manager, Norton Grinding Wheel Co., Ltd., England. Member Ingénieurs Civils de France; Former Secretary of the Association of American Engineers in France. Assoc-Mem. A.S.M.E.

It has been reported that the man-hour production of the Irish workers in the Cork Fordson plant has exceeded that of the American tractor builders.

As a rule, European workers react most favorably to American methods of management and, once the introductory period is passed, sincerely appreciate the greater interest which American-trained executives take in bettering working conditions and eliminating fatigue and working hazards, and all those factors concerning the worker which American management has come to realize are so important.

CHAS. M. MILLS.[18] Any one who has been in Europe in the past five years and has been in touch with industrial organization and management there, knows the reputation which the author and his associates bear among the leading industrialists in Germany, France, Switzerland, Poland, and England. His methods and personality have won him clients and friends in many corners of Europe. His desire at all times to work with people, to outline projects and not to superimpose them, have built his character and reputation upon firm foundations. He has become a worthy international ambassador representing the best in American traditions of thoroughness, accuracy, and cooperation.

It is therefore with considerable trepidation that a casual observer, making only two extended trips to Europe in the past five years, attempts to comment upon the author's excellent summary of European conditions in management.

The writer is inclined to ask the author what effect international agreements, syndicates, cartels, or pools have had in the regulation and promotion of mass production, not only on domestic consumption, but with regard to exports. Has there not been a development from an economic point of view toward the furthering of mass production by international agreement in certain basic industries? What attempts have been made toward an economic United States of Europe or common consuming and export market? The writer has reference, for example, to syndicates in steel and iron, coal, and petroleum.

The author mentions the tariff, habits, wages, and desire for individuality in design as restrictions in mass production. Is there not another important factor in the decrees or national acts that restrict importation—acts in addition to regular tariffs? In Hungary, for example, only three hundred Ford cars can be imported per year. The reason given is to afford protection to a state automobile factory—to a home industry supposedly created for mass production.

Slow turnover of investment is of course recognized as an outstanding European characteristic. In this connection, the writer would inquire how far European industry has been hindered by foreign capital, chiefly American. The word "hindered" is used, for many experts believe that the availability of American capital and the eagerness with which American investment banks have sought to offer money has caused some European industrialists to expand in projects of rather risky and unstable nature. The availability of such capital would naturally lessen the chances of economies through a more rapid turnover of investment.

The author cites the waste of labor in employment in moving freight cars by hand and in treading shavings into pulp in a paper mill. Is it not true that the average European industrialist, largely because of class distinctions, has not recognized the dollar-and-cents savings in the development of human relations in the plant? Has not the enactment of compulsory social legislation—forced upon the employer—blinded his eyes to the potential

[18] Formerly Assistant to President, Standard Oil Company of Ohio, Cleveland, Ohio.

saving in his own plant under the provisions of such laws?

Social legislation has provided greater security and continuity of employment in Europe than in our own country, according to the author. How far has such legislation restricted the employer in management control, say, for example, in Germany? How far have these laws affected the work of the consulting engineer?

In storeskeeping, in plant layout, in planning and scheduling, and costkeeping the methods of application are more or less the same as in the United States. In sales distribution a great field seems to be open to American marketing specialists.

The author mentions the International Management Institute in Geneva as the one group coordinating national management associations. As one who visited this organization last summer and who is somewhat conversant with its scope and purpose, the writer would say that there is a fine opportunity for American engineers to support this institute. We have some things to learn from Europe as well as to give to Europe.

It has been said that American engineers should not be so willing to promote better management methods in Europe which may result in increased competition with foreign nations. The author has most ably answered these critics in his conclusion. International peace can be promoted better by ambassadors of knowledge and scientific approach like the author, and by an international institute of scientific management, than by naval treaties and international politics.

ALLAN H. MOGENSEN.[19] I should like to stress the value of travel and observation. I think we in America do far too little of this. As for our going abroad to visit plants and study processes, we seldom visit plants even in our own vicinity. Before we feel secure in our superiority, we should at least see the others to be sure that we are correct. One of the reasons for our present economic dilemma is our belief that we can proceed by ourselves without regard for what the other fellow is doing, or getting all the facts before going ahead.

Mention has been made of the rapidity with which the European plants take hold, and it is stated that the foreign manufacturer can undersell us because of his low labor costs. These are of course due to the lower standards of living in these countries. These standards will eventually rise. They have always risen in the past and we should expect this trend to continue. We should expect our own standards of living here to continue, despite the pronouncements of even some of our financial leaders to the contrary. When European living standards approach or equal ours, there will be no problem. But what are we to do in the meantime? How are we to meet this competition?

I think that we can find the answer in the proper selection of machine equipment. While it is true that in many fields we are overequipped, it may be that we have the wrong equipment. The textile industry offers a case in point here. We need more careful selection of machinery and equipment, purchased on a scientific basis, and the problem of depreciation handled properly. No machine or tool should be purchased unless it can be economically justified. But on the other hand, if there is a machine available that will enable us to produce more effectively and economically than we are doing at present, we cannot expect to meet competition unless we secure it.

I should like to ask the author how he feels about the question of control in industry. Is not the sort of control exercised by the government in Poland over the salt mines the answer to our problem?

We should not attempt to limit production, but do we not need some sort of control to prevent excessive expansion of plant capacity? We know that the coal industry is in a serious situation because of excess capacity. If the bituminous mines can produce 750,000,000 tons a year, and we use only 500,000,000 tons as in 1929, we have excess capacity, and some one must suffer. Each mine must operate inefficiently, as long as no one is willing or can be forced to shut down. Exactly the same situation exists in the shoe industry. We have capacity enough to produce almost twice as many pairs of shoes as we do produce. Therefore, each pair of shoes we buy carries so much direct labor, so much material, and overhead for two pairs. The author himself has often told us of plants operating at only 40 per cent of capacity.

Is it not this lack of proper planning and exercise of control that puts us in our present situation? Procter & Gamble are buying new equipment and building new plants, but are doing it as justified. They have a definite long-time plan, and follow this closely. When they build a new plant it is not because some one in the organization is overenthusiastic because business is good, orders are coming in, and there is money in the bank, but because the facts show that it will be needed at a certain definite date.

I want to emphasize the difference between failing to modernize at the right time, and expanding unwisely and blindly at the wrong time. Can we do this ourselves, or shall we require some super-controller to tell us when and how much?

AUTHOR'S CLOSURE

Mr. Mills inquires as to the effect of international cartels and syndicates on mass production. These agreements have undoubtedly brought about some standardization of product which has favored mass production and also they have been successful in minimizing dumping on export markets, but in the various domestic markets it seems that they have not been good for industry. The rigid control of prices at rather high levels has prevented any marked increase in the use of products of cartelized industries, and the apportioning of orders to plants without regard to efficiency or costs stifles initiative on the part of executives and results in a lower grade of service to customers.

The tendency during recent years has been to raise the tariffs of the various European countries as in the United States, and in addition there have been the national acts restricting importation mentioned by Mr. Mills. These tariffs and restrictions have been designed primarily to foster domestic production, but they result in limiting markets to a single country and restricting the possibilities of mass production and the consequent cheapening of the daily necessities of life. Tariffs encourage wasteful production, and the consumer has to pay for it.

It is difficult to say how far European industry has been "hindered" by American capital. It is the author's opinion that on the whole both have benefited by the collaboration, although it is true that money from this country has sometimes been provided without any attempt to study the markets, the plants, or the ability of the management staffs. The resulting losses have been painful to the Americans and discouraging to the borrowers. In the majority of cases, however, the lender retains experienced counsel to study the situation, agree with the borrower as to how the money is to be used, and then to see that the plan is carried through wisely. An outstanding example was the recent three years' service of Charles S. Dewey, who, as financial adviser to Poland, not only assured the security of a loan of many millions, but by his guidance of that government through a difficult financial period made a notable contribution to the welfare of the country.

It is undoubtedly true, as Mr. Mills suggests, that the social legislation in Europe causes many employers to go no further than the law requires in developing their relations with workers, whereas it would be to their financial advantage to do so. For example, when accident insurance is paid into a government

[19] Assistant Editor, *Factory and Industrial Management*, New York, N. Y., Jun. A.S.M.E.

fund at a certain per cent of wages without regard to the accident rate in that particular plant, the employer often feels that there is no incentive to carry on a safety campaign, whereas if he did show an interest in accident prevention, the increased running time of his machines and the improved morale of his workers would bring substantial savings.

Within the author's experience social legislation has not directly restricted management control to an appreciable extent, although it has strengthened workers' unions which in turn have restricted employers, sometimes wisely and sometimes unfairly.

It does not seem that social legislation affects in any way the consulting engineer. On the whole it is good for industry, and the consultant merely arranges the details of his client's business accordingly. For instance, in several cases the so-called social allowances, comprising sickness and accident insurance, vacations on pay, pensions, unemployment insurance, etc., average 12 per cent of the payroll. Therefore, in all calculations of costs and in payroll budgets this 12 per cent is added to the wages. Even with this addition the wages are low, and one can hardly say that such legislation is a burden on industry.

Mr. Mogensen brings up the question of control in industry and asks if government monopoly is not what we need in America. There is no doubt that we need a more wise and far-seeing direction of industry, but this can be built up better by industry itself than by federal or state governments, because politics so often places control in the hands of men who do not know how.

A "super-controller to tell us when and how much" is suggested by Mr. Mogensen, and perhaps in some future age such an all-wise being or group will appear, but in the meantime it is possible to introduce more complete planning in the various sections of industry, particularly in regard to the distribution of money and purchasing power.

Modern management has provided a technic which simplifies the most complex situations and plans their control. This can be applied to the direction of the changing forces of industry, and benefit not only one class or one country, but mankind as a whole.

MAN-53-2a

Apprentice Training in Virginia

By C. F. BAILEY,[1] NEWPORT NEWS, VA.

Virginia, along with the rest of the South, is becoming industrially minded. This stresses the need for education for the industries and the arts. The transportation companies, iron works, and shipbuilding concerns require skilled mechanics, trained through apprenticeship. The primary principles in apprentice training are, first, careful selection of candidates; second, importance of the supervisor; and, third, the sympathy and aid of the management. The advantages of apprentice training include recruiting skilled mechanics, with a spirit of loyalty and service, some of whom will develop to foremen and higher officials. To promote this, training schools, on company's time, are advisable. The apprentice group should be self-governing. The conclusions are that the wages should be liberal, with opportunity for increased earnings through premium or bonus work. The prospects for apprentice-trained men were never so bright as now. Virginia manufacturers are taking increased interest, and they recognize the necessity and value of apprentice training. The business of building involves both materiel and personnel—the building of men as well as of machines and ships.



THE South, including Virginia, is rapidly becoming industrially minded. With this come the problems arising from mass production and the need of education for the industries and the arts. Business prospects which were quenched and blighted by the Civil War are now far in advance of any previous period. Development of the large natural resources of the South, such as timber, iron, coal, and water power, is demanding skilled mechanics and engineering talent. In the Virginia Section such talent is principally required by the transportation companies, bridge and iron works, and in shipbuilding, wherein 20 or more independent trades are practiced in the building of a modern high-class ship. In other industries where mass production is the order comparatively few mechanics skilled in the trades are employed.

We see, then, in the Virginia Section a few towns widely separated in which are industries requiring skilled mechanics, and therefore presenting opportunities for apprentice training, which involve somewhat different problems from those found in densely populated sections wherein a single city may constitute an entire section of the Society. This does not excuse the widely separated and less populated sections for neglecting the issue, although the adoption of a general rule is probably not workable for all.

The purpose of the present paper is to create an interest in this subject in the Virginia Section and elsewhere. This has been attempted through correspondence with the principal industrialists in the State, through a meeting of the united engineering societies, through interchange of visits of officials of several industries, and through the Department for Vocational Education, wherein the acceptable principles of apprentice training have been presented, discussed, and illustrated. Considerable interest has thus been created which, with the proper nutrition and fostering, will doubtless be increased to a marked degree.

There are only a few towns in the section where apprentice training is established to any considerable extent. However, there are a number of smaller organizations and contractors who take a few apprentices in certain of the basic trades.

The State Department for Vocational Education is largely directing this matter, and through the influence of its agents great benefit has resulted. There are no very complete data compiled as to the number of apprentices in the State. The report of the Department of Labor and Industry, September, 1929, gives some information, but it is difficult to classify accurately. There would appear to be about 600 apprentices in the different trades that are practiced in the State.

The Principles Involved

In the discussion of this subject by the author at a joint meeting of the engineering societies at the Virginia Polytechnic Institute on April 26, 1930, certain important principles and qualifying factors were brought out which control to such an extent as to constitute the laws governing successful apprentice training.

First, the careful selection of the apprentice group with particular reference to initiative, self-control, and determination; scholarship (completion of high school), moral sense, and physical fitness.

As then stated, since our objective is to train and equip young men to become highly skilled for industry and the arts, it is vital that only those of certain physical, mental, and moral traits be received. We might therefore state the first law of apprentice training as follows:

First Law: Candidates must be physically fit, mentally superior, morally hopeful—hand, head, and heart.

We should include in this intensive apprentice training no boy who has not these desirable characteristics in an outstanding degree. The mediocre candidate displaces one of the promising type who might become an effective influence in industry and the community.

Second, the selection of a supervisor whose heart and mind are in sympathy with the work and with the apprentice and who has the vision, initiative, and organizing ability to devise and carry through a comprehensive training program. We may then state the second law as follows:

---

[1] Engineering Director, Newport News Shipbuilding and Dry Dock Company. Mem. A.S.M.E. Mr. Bailey was born on Aug. 29, 1863, in Greensboro, Vt. His education was received at Worcester Polytechnic Institute, S.B. in M.E., 1888; Dr. Eng. Hon., 1928. He was a machinist, Northern Pacific Railway, Tacoma, Wash., 1888–89; draftsman, Tacoma Foundry and Machine Co., 1889–90; Neafie & Levy Ship and Engine Building Co., Philadelphia, 1890–91; chief draftsman, 1891, chief engineer, 1900, and engineering director, 1918, Newport News Shipbuilding and Dry Dock Company; consulting engineer, Westinghouse Elec. & Mfg. Co., 1915–21; director, Newport News Shipbuilding and Dry Dock Company, Newport News Construction Corp., Newport News Land Corp.; school trustee, Elizabeth City County, Virginia, 1912–16; civil member, member council, 1924, Am. Soc. of Naval Engineers; representative, National Council of American Shipbuilders on Committee to Coordinate U. S. Marine Boiler Rules, 1929–1930; trustee, Worcester Polytechnic Institute; member, Technical Committee, American Bureau of Shipping; trustee and vice-president, The Mariners' Museum; vice-president, Society of Naval Architects and Marine Engineers.

Contributed by the Committee on Education and Training for the Industries and presented at the Annual Meeting, New York, N. Y., December 1 to 5, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.



FIG. 1 THE 235 APPRENTICES, 24 INSTRUCTORS, THE CHIEF INSTRUCTOR, AND THE SUPER-
(The legend on the monument reads: "We shall build good ships here, At a profit—

Second Law: The supervisor of apprentices and the instructors must stand forth prominently in manipulative skill, intelligence, training aptitude, mental resourcefulness, application, and in the high principles of self-control, citizenship, and interest in and vision of the possibilities of the boys' future.

Third, in order to make apprentice training successful, not only must we have the choice, selected personnel, but there must be a realization on the part of the management of the need—an inspiring interest in young men and a willingness to spend time and means to promote such training not only for the good of the company, but for the benefit of the individual, the community, and the country. We may therefore state the third law:

Third Law: The management must be imbued with a hearty interest in the apprentice, visualize his possibilities, recognize the mutual interest existing, and provide both matériel and personnel to foster his training.

ADVANTAGES OF APPRENTICE TRAINING

With care used in the selection of the teaching personnel and the apprentice personnel, the management may confidently look to having eventually a high class of employees imbued with a spirit of loyalty and service who will develop into first-class mechanics and many of them later into supervisors, foremen, superintendents, and even higher officials. It is for this reason that such care should be given to the selection of the candidates.

The management should keep in sympathetic touch with the supervisor and as far as may be with the individuals, lending encouragement to the young men by showing interest and by cultivating a personal acquaintance where feasible. It is vital that the supervisor should follow up all of his students and apprentices throughout the course. Not only should these men be followed during their course, but also, with advantage to both the men and the organization, throughout their service, either with the company or with others. No employer can hope to retain in his organization all of the skilled men which he may train; many will leave, some for a period only and others for their life work in kindred fields. This, however, should not be looked upon entirely as a loss, and many examples might be cited where apprentices have left the company after completing their apprenticeship and have risen to responsible positions in other concerns throughout the country. They thus form a contact and an advertisement for the parent company which is of considerable intangible value. In one's own organization the policy of the careful selection and training of apprentices will encourage and emphasize a study of the history and policies of the company, its founders and leaders, as an inspiration to zeal and loyalty. Such men, so trained, are particularly valuable for promotion to higher positions within the company.

The policy of training is one to be left by the management to the supervisor, who should outline the blocks in each trade which the apprentice should learn and in which he should become practiced and expert. These blocks need not, however, be taught in the consecutive order in which they are naturally grouped. They also serve the purpose of valuable training in emergency for special-purpose men wherein a man may be taught a particular part of the trade and become skilled in this, whereas the apprentice will be trained in block after block until he has completed the educational part of the trade and thereafter will acquire more skill, dexterity, and speed by practice throughout the remainder of his apprenticeship. Such training furnishes the management with a reservoir of potential personnel for the upkeep and development of the organization.

In order to carry on such a course of training it is also advisable to establish in the company's time a school with teachers and trade instructors, and it is here that high-school graduates are eminently fitted to carry on with great interest and value studies directly related to the problems with which they come in contact day by day in the shops. These studies should include physics, chemistry, economics, the principles of accounting, psychology, mathematics, drafting, effective English, accident prevention, and other branches related to the work in hand and to the affairs of the company. In order to develop the morale of the group it is necessary and advisable to make the organization as far as feasible self-governing, which may be done through an apprentice association with committees, officers such as president, vice-president, secretary, treasurer, athletic committee, and honor council, who shall consider all important questions arising as to policies, infractions of the rules of the association, and the discipline to be applied, which may even be as severe as a recommendation for dismissal from apprenticeship. The cultivation of the apprentice traditions forms an incentive; the inculcation, not by direct instruction, but by practice and precept of a broad, correct labor policy is wise and enlightening. The promotion of efficiency as a means of increased earnings for both the company and the apprentice must receive attention, together with improved quality.

CONCLUSIONS

The matter of the apprentice wage is of vital importance. The wage at the beginning and throughout should be liberal, although at first the apprentice may not earn much. This tends to attract the better class and gives a far superior list of applicants from



VISOR OF TRAINING OF THE NEWPORT NEWS SHIPBUILDING AND DRY DOCK COMPANY
if we can; At a loss—if we must, But always good ships."—Collis Potter Huntington.)

whom to choose. It is the policy now in some cases to select only high-school graduates or those who have had the equivalent of a high-school course because of their proved capacity for development.

The inspiration coming to any manager in watching the progress and development of the bright young men is one of great uplift and value. There comes to be a feeling of interest in the boy which is reciprocated by him and which is a great factor in the labor problem of the time. The apprentice should be taught the value of efficiency and given incentives which will develop intense reaction in this respect. These incentives may take the form of the assignment of choice jobs to such as show the ability, interest, and willingness required; the increased interest of the supervisor and the management, without danger of partiality; and later as skill is developed increased remuneration by the adoption of a system of payment whereby the regular hourly rate is supplemented by a bonus or premium based directly upon the character and amount of work performed, which may be established either by contract or in other ways.

Because of mass production in some cases the need for skilled mechanics may seem to be declining; however, one need not be discouraged, since this machine age requires the skilled hand, trained in practical application of the trades, associated with minds trained to straight thinking, with healthy viewpoints of labor policies, which enable young men to advance. The prospects for such a young man thus trained and educated in industry were never before so bright as now. He takes his rank in a different squad, but he takes it just the same along with the college graduate, and often he can outclass the college man. Examples might be cited of these young men occupying positions of supervisors, foremen and superintendents, salesmen, and others. One young man who served in the pattern shop and who is now in a sales department of a high-class machine-building company says that, as he looks at it, there are two types of trained men with whom to compete; first, the purely technically trained man, and second, the practically trained man. The course which he pursued in his apprentice training he considers to be a middle route, and now he expresses his deep appreciation of the advantages which he enjoyed in this training. With the skill gained both by practice and theory he has risen without embarrassment to a leading position.

Another graduate is in a large, successful building corporation furnishing important machinery for marine work. He is a diplomat in adjustment of difficulties, which may be fancied or real, with the purchasers of his company's products. Others who have remained with the parent company have risen to high positions as foremen, as assistant superintendents, and (one) as superintendent. They are now of great value to the company in that they are in sympathy with the broad system of training which is so advanced over that which they enjoyed. These men, familiar with the company's traditions and policies, are the backbone of the organization as representatives of the company in dealing with the inspectors and supervisors sent to the works by its clients.

One apprentice in the civil engineering department lost his job soon after graduation due to drastic curtailment of work in the company. This young man took a minor position in the city engineer's office, and within three or four years he became the city manager, which position he still holds and in which he has made a greater success than any former incumbent—a pride to the company where he was trained and an honor and asset to the city of which he is manager.

A number of letters have been received from manufacturers and operators in the Virginia Section in which is unanimously acknowledged the value of apprentice training. The transportation people express great interest in this and are anxious to progress with it. They admit that it requires much more thought, study, and consideration than it has formerly received.

One Virginia manufacturer says that he considers apprenticeships as becoming more and more a necessity in American industry and that apprentice schools are necessary in the solution of the modern industrial problem; that while mass production has in a large measure changed the need for skilled mechanics, yet its leaders and master mechanics and foremen must be skilled.

One Virginia railway official stresses the importance of examinations physically and along educational lines in order to select the best material for training. Another railroad superintendent states that apprenticeship training cannot be worked out along general lines to meet any working conditions and that no cut-and-dried plan for one works can be adopted in its entirety for some other. He thinks that self-improvement work can be effected and must be largely initiated by the apprentice under suitable encouragement by the management. All of this can be greatly fostered by the management in its dealings with the apprentice.

The vice-president of a large foundry company in the western part of the Virginia Section writes as follows with regard to his newly established apprentice system:

This program has been in practice only about four months, and of course it is impossible to give an intelligent idea of the worth of a program of this nature with the short experience that we have had. We are, however, more than pleased with the progress made thus far

and feel more convinced than ever that our training program will prove to be one of the most worth-while things that we have undertaken for many years.

The products of a well-planned and directed apprentice training course where the attractions offered are such as to draw the brightest, most intelligent material, are greatly improved labor conditions, highly skilled mechanics, an intelligent, happy citizenship, and a supply of potential supervisors, foremen, and even superintendents. When we see this illustrated and stop to consider, we realize the great importance of this training. The supervisor or foreman occupies a dual position of great significance and influence. It is he who represents the management of the company to the men under him. As some one has said, "He must be familiar with the company's policies on the one hand and on the other represent the workmen to the management." Therefore, his own training is of great importance. Not only should he interest himself in teaching the apprentice the trade in all its branches requiring hand, head, and heart, but he also has the rare opportunity of touching the other vital chords of the man's being.

The Chamber of Commerce of the United States considers that foremanship holds the key to production. The foremen can create a sympathetic contact with the men and be an inspiration to them, a service to the company, to the men, and to himself. With such foremen the management gains in spirited team-work, morale, loyalty, lower turnover, reduced accidents, greater output, and an improved spirit of orderliness.

While we honor the outstanding professors in engineering colleges and recognize the great influence which they have upon the development of the engineers of the future and the personnel coming up to take the places of us here today, still many manufacturing concerns today present opportunities which challenge the best in any man to take a large share in this upbuilding of our engineering organizations and personnel. If there can be created in the executives of the many fine industries of our sections an intense love for and interest in the young men whom they are training, they will find that it stimulates the faculties and gives enjoyment and recreation of a satisfying character.

With the right spirit pervading an organization there is a place without jealousies or friction for the development side by side of the well-trained mechanic and the technically but less practically trained college man. Many industries, like the great electrical manufacturing companies, have positions where it is essential that men with eminent knowledge of the sciences are required, but in these same organizations and in most organizations involving the trades are numerous enviable positions for the properly trained apprentice.

In order to carry out such training a company must make a set-up particularly for it. The smaller companies, however, may not be able to do this. In any case, there is one source of great help which should not be left untapped, and that is the assistance granted by the practical application of the Smith-Hughes Act which is fostered and administered in the State of Virginia by the Department for Vocational Education. In the Virginia Section this department has been of great service in helping to develop, establish, and promote the apprentice course and the training of foremen. It is only a step from the training of these well-balanced and bright apprentices to the development of skilled mechanics, supervisors, and foremen which should take its turn.

A few years of a young man's life at this age is not wasted in service before he rises to the position of greater authority. Eugene R. Grace says: "Not only is the hard work, in humble jobs, a test of these young men's endurance, but it serves to give them a background of sympathetic understanding of the workman's point of view which enables them as executives in later years to strike a just balance between their duty to the corporation which employs them and to the men who work under their direction."

In all this training are many opportunities to create loyalty for the company by briefly reviewing in an interesting way the history of the company, dwelling upon its outstanding personalities, past and present, and the policies which guide it.

In any organization or in any large group of young men there is always latent, unnoticed talent in one direction or another, some in special skill, some in leadership, some in individual development.

These qualities are to be looked for and noted, and they can be found in the shop, in the schoolroom, and in the games and sports, which should be largely fostered. The great business of building involves quality in matériel and personnel; the building of men as well as machines or ships.

Throughout the country the most successful manufacturers and engineering concerns are surrounding themselves with staffs of the brightest minds, who by means of conferences, study groups, and lectures are learning the vital principles of the organization and the company's policies.

To many, and particularly to those engaged in small industries, the foregoing may seem impracticable. It has, however, been demonstrated that this is not so. Results obtained under the principles and conditions outlined are encouraging and inspiring and demonstrate that, with the proper spirit of management, the rare supervisor of training, and the selected group of apprentices these results can be realized and improved upon.

Discussion

D. S. Jacobus.[2] Mr. Bailey's paper radiates his enthusiasm and keen insight in the training of young men as carried on in his department. There is a fascination in this work. The principles brought out by Mr. Bailey and his conclusions are in harmony with ideas that have come to the writer in his career as a teacher for over 20 years and his longer contact with the outside field. Hand, head, and heart must strive together in the young man, and the instructors must stand forth prominently in training aptitude.

In viewing teaching in its broad aspects the writer is more and more convinced that the success of any course is more directly measured by the amount of effective work done by the students than in any other way. To learn how to work effectively one must himself work effectively. There is no short cut to success which will bypass hard work. Some may challenge this statement, but where one may claim to have reached the top of the ladder through natural ability rather than effort, there will be a hundred who will acknowledge the benefits of hard work. It is feared that in some cases our modern training is along too idealistic lines. The goal is often set so high that the great majority fail in its achievement. Young men with the sort of training outlined in the paper may outdistance many of those who are looking so far afield that they cannot see what is at their feet.

Some of those reviewing engineering education have laid much stress on character. All know that character is indispensable, but with it should go a fundamental training in the principles of one's profession, and above all a willingness to work hard in applying them.

Character cannot be taught by precept. There are no keener or fairer critics than a class of young men, and any amount of advice on the bearing of character on success by one not having a high character will fall on empty air.

[2] Advisory Engineer, Babcock & Wilcox Company, New York, N. Y. Former President, A.S.M.E.

The writer is happy to endorse all that Mr. Bailey says in his paper and hopes his experience in the training of young men will carry with it as many happy recollections as his own.

E. W. KEMPTON.[3] When reviewing a paper, one might take the stand that there must be something about which he should be critical. The writer thought that he had such a point in Mr. Falk's paper when he stressed the control of the apprenticeship system in Wisconsin through a commission established by law. However, before concluding the paper the author states that the passage of a law is unnecessary, and in this the writer concurs. Mr. Bailey's summation of policies and attitudes for effective apprentice training is well made. It is a pity that there is not a greater appreciation of the benefits to be derived when such advice is followed.

The writer is an ardent believer in apprenticeship, and inasmuch as the papers of Mr. Falk and Mr. Bailey set forth specific plans, principles, and benefits of apprenticeship training, he can but add substantiating evidence.

The reason why there are not more apprentices in training is not because of failure to legislate the control of apprenticeship training, but rather because of the impatience of American industrial superintendents and foremen and their interest in little that does not create immediate results. It takes time to develop an apprentice into a mechanic, and time is the one thing of which we have the least. It requires giving thought and consideration to something that is not a specific production problem of the moment; hence there is too great a willingness to let some one else develop mechanics and then we shall hire them away when we want them.

To secure an idea of the number of skilled all-around machinists (not operators) that exists, one needs only to talk to an employment officer who was in the field for such workers during 1928 or early 1929.

Of machine operators there is no scarcity, but when one looks for the so-called all-around worker such as is needed in the maintenance and repair shops of industry, one does not have to be convinced of the necessity for apprentice training. The reason for this condition is not difficult to find, for in maintenance shops where foremen and superintendents are pressed quite as much for costs and other economical factors as in production shops, there is little immediate interest in taking a man who has developed into a good lathe operator and giving him experience on some other type of machine such as the milling machine or planer. In mass production shops there is no possibility of this.

Furthermore, there is little opportunity today for a young man to go through that process that was formerly known as "stealing a trade." Once a young man could secure a position in a shop, acquire experience, and learn how to operate a lathe, and then if he were not given an opportunity on some other machine, he could leave, secure employment elsewhere, and bluff his way until he had obtained further experience. By repeating this process a sufficient number of times, he eventually became a good mechanic. Today, with well-developed employment departments, it is quickly discovered that the experience of a man has been limited to one particular type of machine, and he is either turned away as not required or is employed for work similar to that which he has been doing, with practically no opportunity to broaden his experience. Even in the limited number of shops where this is still possible, there is no appreciation that this is the most expensive and inefficient method possible. Hence, there has been during the past two decades a very appreciable decrease in the relative number of all-around mechanics, notwithstanding the fact that the total number employed at the machining trade has increased. To secure further corroborative information, one need only make inquiry pertaining to the average age of mechanics employed in any maintenance, repair, or jobbing shop where there is necessity for something more than operators. He will find that the average age is constantly increasing. He will further find that potential material for foremanship positions is relatively scarce, if a foreman is required who has had a wide experience as a mechanic.

The strange part of this picture is that corrective measures are not difficult nor expensive to take. There appears to be an impression upon the part of executives that apprentice training is expensive and cumbersome, but it need not be so. In fact, it may be quite the opposite.

In almost every community there exists some part of the public educational system that may be readily adapted to the training of apprentices, the teaching of that auxiliary knowledge necessary to a good mechanic. Modern-minded educators connected with public educational systems are usually pleased at the opportunity to cooperate. It has been this writer's experience that there is altogether too much said about the unwillingness of the public schools to cooperate in the training of young men and women for business and industry. As a matter of fact, it is usually the other way around. Business men and factory executives will not cooperate with public schools.

With such cooperation and a little conscious effort upon the part of the machine-shop foreman or superintendent or some other executive, in the matter of diversifying experience, the training of apprentices is neither complicated nor expensive.

The apprentice system developed through the cooperation of manufacturers and the public school system in the City of Cleveland is a good illustration. Here the embryonic mechanic attends a school that has been set apart for the purpose, full time, 8 hours per day, for the first year of his apprenticeship. One-half of this day he spends in the classroom studying mathematics, blueprint reading, and other auxiliary material, and the other half of the day in the shop of the school. At the conclusion of this first year the instructors, who are men who have been successful in their chosen trade, are able to distinguish between those who will and those who will not succeed as apprentices. At the conclusion of this first year the successful apprentices are employed by manufacturers in Cleveland, who are always willing to take them, even anxious to get them.

The second year of apprenticeship is devoted to a cooperative plan whereby the apprentice spends two weeks in school and two weeks in the shop of his employer, boys being paired in teams, one being on the job all of the time.

In the third and fourth years of apprenticeship the boy spends all of his time in the shops of the employer, with the exception of four hours a week which he spends in the school on school work.

The success of this plan is evidenced by the fact that it is constantly growing, and those employers who are using it are becoming more satisfied with results as time goes on. However, there are still manufacturers in the metal trades of the City of Cleveland who complain because of the scarcity of all-around mechanics, but who will not exert the effort to participate in this plan. Some manufacturers appear to believe that mechanics simply sprout from nowhere to be plucked at the employment office as required. It is the experience of those conducting apprenticeship courses, whether fully under their own control or in cooperation with public educational institutions, that their best mechanics are from the ranks of their own apprentices.

A discussion of the means for securing a well-rounded training of an apprentice would not be complete without mentioning the fact that apprentices may secure the auxiliary knowledge that goes with their trade, through correspondence school courses.

[3] Educational Director, American Steel and Wire Company, Cleveland, Ohio.

Some employers are today conducting very effective apprenticeship courses and requiring their apprentices to take a correspondence course, which has been developed for that specific purpose, correlated to their trade. Utilizing this facility, no employer need say that he has not the means for properly training an apprentice.

There is just one more idea often expressed against the young men of today that should be exploded, and that is the idea that applicants for apprenticeship are scarce, that young men will not take apprenticeship courses, and that they want white-collar jobs. Wherever a systematic apprenticeship course has been organized, with reasonable guarantees to the young man that he will secure an adequate training, there are always more applicants than can possibly be employed.

It is the experience of the American Steel and Wire Company in the various communities in which it operates that in no case need we take any apprentice who has not graduated from high school unless we so choose. Applications are always on the waiting list.

There is a bountiful supply of good material for apprenticeship courses. Apprenticeship courses are not difficult to organize; they are not expensive to conduct; properly conducted, they easily pay for themselves; and they furnish the best source of material for advancement to higher positions.

Morris S. Viteles.[4] Apprentice training programs are, in most instances, being developed for use in plants employing a large number of apprentices. The small plant employing a few apprentices has, in the main, been overlooked in the formulation of such training programs. This omission has made it practically impossible to exploit the full possibilities of such plants in increasing the corps of adequately trained machinists. This problem has been faced by Philadelphia Electric Company in the organization of a training program for apprentice machinists.

The training program was organized for the generating and operating maintenance section, employing no more than six apprentice machinists at one time. The nature of the job is such that there is no opportunity for production work. There is no way of foreseeing the kind of work in which the apprentice will be engaged at any time in the future and no way of bringing systematic training to the job so as to insure organized practice on the various trade skills in a definite, logical order. There are difficulties in insuring centralized supervision, inasmuch as an apprentice may be shifted rapidly from one location to another. Moreover, the fact that there are so few apprentices makes it impracticable to organize classes for instruction in the related subjects with which the skilled machinists should be familiar.

To overcome these difficulties and still provide systematic training for these apprentices, the following training program has been developed.

In this training program each apprentice works approximately 48 hours per week in the different shops and plants throughout the system over a period of four years. He is transferred periodically from one station to another, in order to familiarize himself with the maintenance work from all its angles.

Each week 4½ hours is devoted to formal training under the direct supervision of an especially assigned instructor. The remaining time is devoted to regular routine maintenance work under the guidance of a foreman or mechanic or as part of a crew working on larger jobs.

The formal training program is divided into main parts, as follows:

(1) *Machine Shop Projects.* This includes a series of shop projects by means of which each apprentice may become familiar with the operation of every machine and with all the bench practices which should be familiar to a skilled machinist. These projects supplement such casual instruction on the machines and on bench-shop techniques which the apprentice may receive from his foreman while working on the job.

The program includes a series of projects ranging in complexity from a simple grinding job to one involving the use of practically all of the machines and all of the bench techniques which the apprentice has practiced in working earlier projects.

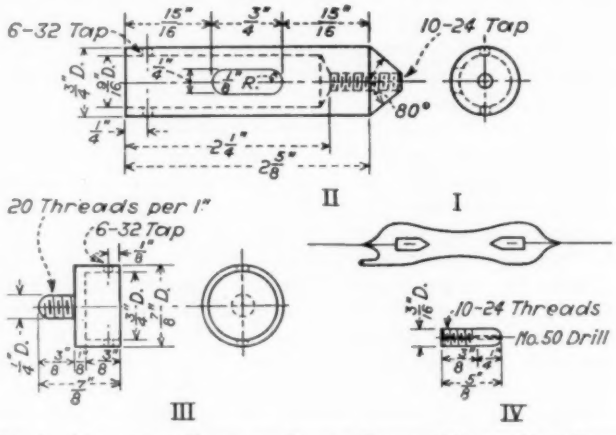


Fig. 2 Apprentice Training Course Project No. 22, High Volt Detector

(Material required: Part I, one Neon tube. Part II, one piece of 7/8-in. round fiber 3 in. long. Part III, one piece of 7/8-in. round brass 1 in. long. Part IV, one piece of 3/16-in. round brass 5/8 in. long. Part V, two 6/32-in. round-head brass machine screws 1/4 in. long. Procedure: Part I: The Neon tube is furnished by the storeroom. Part II: (1) True up fiber in lathe; leave approximately 4 in. extending out of chuck. (2) Square end and turn diameter. (3) Center with center drill and drill 9/16-in. hole. (4) Drill small hole; use extension drill. (5) Cut off at correct angle. (6) Lay out, drill, and file sight hole as per blueprint. (7) Tap small hole. Part III: (1) True up 7/8-in. round brass in lathe chuck; leave approximately 1¼ in. extending out of chuck. (2) Square end. (3) Center and drill hole. (4) Square bottom of hole with bottom drill. (5) Set cutting-off tool ½ in. back of face and feed in to within 17/64 in. diameter. (6) Move tool ahead and make a second slice, leaving approximately 1/8 in. metal in between. (7) Move tool back and cut the metal left between the grooves. (8) Repeat these operations until you have a groove approximately 9/16 in wide. (9) Turn the diameter to 1/4 in. with the same tool. (10) Cut off. (11) Move the bench vise and cut 1/4 in. 20 thread with hand stock and die. Part IV: (1) With hand stock and die, cut thread on 3/16-in. round brass. (2) Cut off to length with hacksaw. (3) Clamp in V-block and center-end with center punch. (4) Drill in drill press. (5) Chuck piece in drill press and round-end with file. Assembly: (1) Assemble parts II and III and drill and tap holes as per blueprint. (2) Mark relative position of parts with scriber and remove cap. (3) Solder wire of Neon tube in cap. (4) Assemble parts I, II, III, and V. (5) Slip part IV over protruding wire and screw it in place. (6) Solder protruding wire to part IV and cut same. Remarks: Considerable care should be taken in the soldering operation, so as not to bend the filament wire of the Neon tube too near the glass, as it will break easily. Questions: (1) How do you grind a drill for drilling fiber? (2) Describe the construction and the use of the Neon tube.)

The character of these projects is shown in the diagram (Fig. 2). The gradual progress from instruction and practice in simple to complex trade skills is evident from a comparison of projects taken from the early and later phases of the training course.

(2) *Correspondence Work on Related Subjects.* A skilled machinist must be familiar with a certain amount of mathematics, with the fundamental principles of mechanics, etc., in the practice of this grade. Inasmuch as it is impossible to organize formal courses in these related subjects, instructions in these subjects are given by the correspondence course method. Arrangements have been made with Pennsylvania State College to furnish these correspondence courses. The arrangement is such that the college furnishes all of the text material and grades all of the lesson assignments. The ground covered by the correspondence work includes courses in mathematics, blueprint reading, mechanics, electricity, and shop economics.

(3) *Conference on Machine-Shop Practices.* In order to familiarize apprentices with the general aspect of machine-shop

[4] Director of Personnel Research, Philadelphia Electric Company, Philadelphia, Pa.

practices, talks are given at periodic intervals by members of the supervisory force of the maintenance section and by representatives of other departments who are asked to cooperate on this phase of the training program.

(4) *Conference on Assigned Lessons.* Provisions are made for conferences between the apprentice and the supervisor of the training program in the maintenance section on problems which arise in connection with the correspondence-course work and on other problems which grow out of the training program.

Length of the Training Program. The training program extends over a period of four years. The first six months of apprenticeship is considered as a period of probation for the apprentice. Accurate record of progress is kept.

L. P. ARDUSER.[5] So much attention is being paid, and must continue to be paid, to the problems of distribution, that it almost seemed that production, the goose that lays the golden eggs, might be neglected. In fact, the writer has heard business men and engineers who ought to know better say: "We do not need to worry about production any more. That job is done. We can forget it. Now we must distribute what we can make." Let us stick to fundamentals. We cannot forget production, for, argue as we will about sales quotas, advertising media, discounts, etc., it still remains true that efficient production is essential to effective distribution. Consequently, the papers by Mr. Bailey and Mr. Falk are significant. We are not going to be allowed to forget where the golden eggs come from even if we have no scientific means as yet for their distribution.

After going over both papers the writer finds very little with which to disagree. He is not quite so familiar with conditions in Virginia as he is with those in Wisconsin, but is sure that both states are on the right track.

There are many ways, aside from our former method of wholesale importation, in which we attempt in America to produce skilled draftsmen and all-around mechanics. We have the mechanics institutes such as Franklin for evening courses, the free trade schools such as Pullman, the two-year day and evening institutes such as Pratt Institute and Ohio Mechanics Institute, technical high schools such as Cass, Brooklyn, Lane, Milwaukee, Omaha, Central Toronto, and Roosevelt at Los Angeles, cooperative high schools and colleges, two-year engineering courses such as exist at Carnegie Institute of Technology in the School of Industrial Arts, a host of company apprentice schools such as at General Motors, General Electric, Westinghouse, New York Central, and Pennsylvania Railroads, R.R. Donnelley, and others too numerous to mention, many Y.M.C.A. day and evening trade schools, college-extension and private correspondence-school courses for employed craftsmen, and finally the nation-wide system of vocational schools operated separately by each state, but guided and stimulated by the Federal Board for Vocational Education largely through the power of granting or withholding federal aid.

Many of these methods and combinations of methods are very effective, but our production of trained all-around mechanics for the various industries is still very spotty. We do need additional organization along state and national lines to provide opportunities to compare notes and share experiences and to add weight to our pleas. The Department of Manufacture of the United States Chamber of Commerce, the various national trade associations, the American Vocational Association, the National Manufacturers Associations, the S.P.E.E., and our own engineering societies are all working on the problems of correlation of effort, while the peculiarly effective set-up of the Federal Board for Vocational Education, with its regional traveling agents, gives it the power to assist all of the other agencies in guiding their efforts in training along proper lines.

Like the U. S. Bureau of Education, the Federal Board for Vocational Education is a great clearing house. The writer has seen technical schools in corporation, private, and public systems here and in Canada modify their methods to conform more nearly to those advocated by the Federal Board. Some of these have had a century of tradition back of them. He has seen much engineering college teaching influenced directly and indirectly by the Federal Board and the American Vocational Association. So whatever agencies are used in apprentice training, do not forget to use state and local departments of vocational education and to see that all agencies cooperate.

As an example of cooperation in Brooklyn, the Brooklyn Union Gas Company sends selected employees to our evening school for a year of preliminary training, after which these men go to the evening classes at the Polytechnic Institute of Brooklyn and then to those at Columbia University. We tend to sift out the impossibles in our first year of preliminary training. In this way men with company trade experience are trained as gas engineers. In addition the directors of all of the important private and public educational systems in Brooklyn are united in one organization which deals as a unit with educational problems.

In connection with apprentice training, the writer's experience leads him to say that it is a relatively easy matter to impart principles, but that intangible something known as trade skill cannot be imparted by a teacher to a learner. Job skill is only acquired on the job itself by monotonous repetitive effort.

Not only skilled mechanics but a host of machine operators (and this applies to office as well as to factory work) are required by modern industry. In fact, as specialization increases, the relative need of trade training as such decreases, but the problem of supervision becomes complicated and acute. And that puts a new job up to the foreman.

When Arthur Mann of the New York State Department of Vocational Education was training a number of Brooklyn business and industrial executives as conference leaders at Brooklyn Central Y.M.C.A., one of the things worked up by the group itself was a table, as follows:

	Plan	Direct	Supervise	Train
Manager.....	75%	15%	5%	5%
Foreman.....	10%	15%	55%	20%

We also agreed that any superior must follow the teaching process in giving orders. Fully 90 per cent of all operator failures are foreman failures if traced back properly. They are largely due to wrong methods of giving orders.

The experience of the group also led it to agree that the study which revealed the following was largely correct:

Fully 98 per cent of all industrial accidents are preventable, and 88 per cent of these are due largely to faulty management. The foreman is an integral part of management. He has some job these days, has he not?

Foreman training has amply justified itself by results. It is believed that the trend in methods of foreman training is toward the conference method. Expert lectures and textbooks have their place in conferences, but they must be subordinate to the conference itself for good teaching.

Before any company institutes conference methods of foreman training, however, the top management would do well to search its heart for sincerity of purpose. The conference is a two-edged sword; it cuts up as well as down. When members of a group of foremen are once started to thinking cooperatively, they may not only think about themselves and their operatives, but they may also think about the top management itself. They

[5] Assistant Director, Education Department, Brooklyn Central Y.M.C.A., Brooklyn, N. Y. Assoc-Mem. A.S.M.E.

may even come to discover for themselves that management can throw more efficiency out of the window in five minutes with a teaspoon than the foremen and men can bring in the doors in eight hours with shovels. So if one starts foremen's conferences, he can be sure that his actions speak louder than his words. He will have to work hard and play the game straight if he uses the conference. Preaching or telling is ineffective in any field.

It is quite apparent, therefore, that when foreman training is discussed, one must necessarily bring in top management training as well. All the closed minds in industry are not in the lower ranks.

Mr. Falk stated that the training of apprentices must become a general institution in as many business and manufacturing establishments as possible. The writer would like to emphasize the word "business." We operate business and technical schools in the Y.M.C.A. in Brooklyn. The problems in each are quite similar. In fact, if one will write for the 1930 pamphlet on "The Necessity of Closer Relations Between Business and the Schools," by Cameron Beck, personnel director of the New York Stock Exchange, he will be struck with the general similarity of the training problems encountered in the office and in the factory. Which leads to the remark that the apprentice method, or the cooperative method if one wishes to call it that, can be applied to training in business as well as in industry, and with decided advantage to business and industry, and the schools. The writer well knows the problems that come up such as coordination, vocational guidance, placement, and the like, but these things have to be done somehow, so why not do them as well as is known how in a cooperative way? The writer's guess is that the trend in education is toward the cooperative or "sandwich" method.

There are many reasons for this, economic, pedagogical, and otherwise, but if it is thought that the writer is wrong just consider M.I.T. with three such courses, N.Y.U., Marquette, the University of Cincinnati, Georgia Tech., Detroit Institute of Technology, and some other schools in the engineering field alone that any one can find out about for himself. The same method is also used in some schools of medicine, liberal arts, and business, and quite effectively, Dr. Abraham Flexner to the contrary notwithstanding.

Bryan Houston, manager of training for the Standard Oil Company of Ohio, led the sectional conference on education and training in the industries at the Conference on Industrial Relations conducted by the Industrial Department of the National Council of the Y.M.C.A. at Silver Bay, New York. Mr. Houston remarked that salesmen, because of their greater articulateness and because they have been beaten into acceptance of the idea that they do not know it all are frequently more ready to accept training than the manufacturing groups. Think that over. But to ward off all criticism, let it be seen that apprentices are taught oral expression. The English language is one of the most effective tools if used properly.

In closing, let it be said that the whole program of apprentice training is just a start. It is not only a means of training skilled mechanics, but also a way of interesting men and boys in continuing their education throughout their lives on a voluntary basis. And that is important for several reasons. An open-minded man is adaptable and can change his job if industry requires it. The man with a closed mind operating on the knowledge that he obtained, say, ten years ago, in high school or college and in industry, is a nuisance to himself and to others. He cannot change jobs to save his life and he is an important element in the social lag that complicates so many problems of the day.

Support schools. It pays to do so. We need a Federal Department of Education that will cover every branch of this great field, with a separate cabinet officer at its head. In this way we may get business, industry, and education coordinated as they must be to secure the balanced progress we so sadly need.

Let the writer close with the words of Harvey N. Davis in his opening address at the Silver Bay Conference: "The stake that industry stands to lose if a balanced progress, material, cultural, and spiritual, is not attained may be its own existence, but the stake that industry may play to win is an honored place in a civilization far finer than any of which this generation can even dream."

JOHN A. MCCARTHY.[6] While interested in this type of industrial training, the writer feels that he cannot with justice discuss any of the statewide problems in this development in Virginia since we in New Jersey have many problems in the same field which we have not successfully solved. Mr. Bailey presented an apprenticeship-training program which has been established under ideal conditions in the Newport News Shipbuilding Company. The main features of this program were outlined in a wonderful manner by Mr. Via at the American Vocational Association Convention in New Orleans just a year ago.

The writer is interested in the three laws which Mr. Bailey sets up for the guidance and administration of a successful apprentice program. The factors used in the selection of apprentices, such as initiative, self-control, and determination, are factors that are not easily measured in a selection process. It is true that these factors may be observed while the boy is still in school and before he enters his apprenticeship in the industry, but it must be remembered that these factors can be observed only through behavior, which is much influenced by environment. Those who come from industry realize that the environment of the pupil in the school is entirely different from that of industry. The things which are prohibited in school and which produce one set of behaviorisms are required in industry with probably a different kind of behavior resulting.

The writer is not sure that it is advisable to set up high-school graduation as a requirement for apprenticeship in all trades. Even in a shipbuilding organization there are trades which can be successfully mastered by boys who have completed only the seventh and eighth grades. Perhaps high-school graduation is set up as a measuring device which will insure a selected group with high intelligence. There are more scientific means of measuring intelligence, however, than merely requiring high-school completion, and many boys who have been retarded in school because of illness or migration of parents will be found to have an equally high I.Q. as those who have graduated from high school. In using high-school completion as a standard, we are confusing schooling with intelligence and are missing a group of boys who may have more desirable aptitudes for trade training than those who have had greater educational opportunities.

The second and third laws which Mr. Bailey establishes are fundamentally sound—particularly the third, which sets up management's responsibility in the program. At the present time it is unfortunate that many large manufacturing organizations are dropping their apprentice-training program because of a business depression. These organizations are not releasing their mining properties or other sources of raw products, as they know that there will be a return to prosperity with a demand for such raw material for the factory production. Only two years ago there was a decided shortage of skilled tool makers along the Atlantic Coast, and the very industrial organizations which have now dropped their apprentice-training programs were suffering because of that shortage. The shortage will be more acute when prosperity returns.

[6] State Supervisor of Trade and Industrial Education, New Jersey Department of Public Instruction.

Employers in the building trades are notable offenders in this respect, and as a result of the present depression in the building trades, hundreds of so-called "half-baked" mechanics will be found on building operations when building is resumed. Boys are engaged as apprentices and held on the job during a building operation to be dropped at the first ceasing of work on that operation. When a new job is started, in most instances new apprentices are engaged.

The writer is interested in Mr. Bailey's outline of studies for an apprentice group, particularly since such an outline included chemistry, economics, principles of accounting, and psychology. There is a responsibility for determining just what trade skills are involved in an apprentice-training program and then determining what mathematics, science, and drawing are necessary for these trade skills. Anything beyond that must be classified as broadening or cultural subjects. Too many apprentice programs are spoiled because the objectives of this program are to develop trained mechanics, and the technical training program set up is far beyond the needs of the mechanic, being intended for developing engineers. Of course eventually a far-seeing organization is looking to the development of supervisors and minor executives through their apprentice program, but the first job should be to develop the mechanic and then to set up another training program for those mechanics who give evidence of ability for supervisory positions.

The Federal Board for Vocational Education has made many wonderful trade analyses for the purpose of determining what a trade-training program should include. The most notable of these is Bulletin 52, an analysis of the machinist's trade. If attention is given to such analyses when a trade-training program is being developed, a more effective training program will result.

Mr. Bailey emphasized the development of the heart, as well as of the head and the hand. Recently those in trade and industrial education have considered this from the viewpoint of morale. It is expressed in the Allen-Richards formula, which also is outlined in Bulletin 52 as the factor "Mo." Morale has to do with the spirit of the individual or the group and should not be confused with morals. Morale may include some factors which are found in morals, but it is possible for good industrial morale and poor morals to be combined in the same individual. A study of failures of young persons in industry indicates that most of these failures are due to poor morale rather than to any deficiency in trade skill or technical knowledge. Most employers are ready to accept young workers with good morale even if they have limited trade ability.

Mr. Bailey also mentioned the importance of foreman training if a successful apprentice-training program is to be carried out. Foreman training assists the apprentice-training program because the foreman himself has a better view of his own job and a better appreciation of the difficulty of other workers in his organization. Then, too, the foreman is the individual with whom the apprentice has his industrial relationships. Unless the industrial organization is a very small one, the apprentice never meets the superintendent or representatives of the management. It is important, therefore, that the foreman know his job, and he should be trained in the best teaching methods to use during the learning period of the apprentice on the job.

The writer's chief comment on the paper prepared by Mr. Falk on "Apprentice Training in Wisconsin" is that we in New Jersey are envious of the Wisconsin apprentice law. We wish that we had a similar law; we are certain it would materially assist in solving the difficulties now experienced in apprentice training. The Wisconsin law makes it possible to properly organize and supervise apprentices. Without such a law, apprentice training is usually unorganized. No adequate training standards are established, and the ethical practices of both employer and apprentice are questionable. We have been attempting to coordinate the apprentice-training activities within the state, but unfortunately such an important training program is left to local districts and employers without any legal means for supervision and control.

We are hopeful, however, that some day we may be able to profit by Wisconsin's example and place our apprentice training on a more sound basis.

MAN-53-2b

Apprentice Training Movement in Wisconsin Industry

By HAROLD S. FALK,[1] MILWAUKEE, WIS.

The apprenticeship movement in Wisconsin has been made a concerted organized effort. In developing the work important considerations and tendencies have been taken into account, such as the shortage of skilled mechanics in American industries at the present time in spite of considerable unemployment, the requirements of production as well as of jobbing industries for trained men, and the importance of the human factor in all industry. Employers and employees have cooperated in the promotion and support of an apprenticeship law in the state which stabilizes and dignifies apprenticeship by preventing exploitation of boys and by setting standards of workmanship in the trades. As a result of public favor and the enthusiasm of influential individuals, apprentice training has developed rapidly in the state. There are 3350 apprentices employed and 2567 have graduated in the last 15 years. At the present rate of graduation, this number will be increased very much in the next 15 years.

However, neither the Industrial Commission nor the industrial leaders are satisfied with the progress which has been made. The number of apprentices in the state is inadequate compared with actual requirements for skilled men in the trade. Nevertheless, an excellent foundation has been laid upon which greatly increased activity may be built in future years.



THE apprenticeship movement in Wisconsin industries has attracted much attention in recent years, not so much perhaps because of the number of apprentices in actual training, although the number is considerable, but more particularly because apprenticeship has been made a concerted, organized effort. In most industrial and commercial communities there are and always have been employers who have given young men apprentice training on their own initiative and according to regulations and customs which they themselves have established, largely upon a basis of tradition. Such independent apprentice training programs include some of the finest institutions of the kind to be found anywhere. However excellent as these training programs may be, they can afford opportunity only to those young men who happen to be employed in these organizations. In order to make apprentice training a social factor on the one hand, whereby organized experience may be provided for the maximum number of those who wish to follow skilled trades, and on the other hand to provide a proper quota of skilled mechanics for the industries of the community, the training of apprentices must not be carried on merely by one or the other random employer, but must become a general institution in as many business and manufacturing establishments as possible.

An attempt has been made in Wisconsin to build an apprentice-training system which shall satisfy these requirements. Employers usually resent what is ordinarily called meddling in their affairs on the part of state government. However, Wisconsin employers were themselves largely instrumental in the passing of the apprenticeship law whereby the Industrial Commission of Wisconsin is authorized and enjoined to direct and supervise the training of young men in the industries. In many trades and industries committees have been appointed which cooperate with the Industrial Commission and direct the apprentice affairs of their respective organizations and promote the movement in every way practical. As a result of all this, apprenticeship in the state has achieved a dignity and a stability which would not be possible otherwise.

In the development of this apprentice-training work, certain fundamental considerations had been recognized and taken into account. For instance, organization seems to be a keynote of all modern activities. Business men depend upon organization, not only for success, but even for survival. They have organized their purchasing, their finance, their accounting, their manufacturing, and their selling. Is it not reasonable, therefore, that they should organize the training of men for industry? Men are the most important factor in industry. Accordingly, the training and development of men should be even more highly organized than other business and commercial departments.

Again, in spite of considerable normal unemployment at all times and a serious unemployment situation at the present time, the fact remains that there are not enough highly skilled mechanics in our industries. In fact, in some parts of the country, according to reliable reports, there is a distinct shortage of skilled mechanics even at the present time. The term highly skilled mechanic is here taken to mean an artisan who is fully qualified in all branches of his trade and not merely in one operation. It is taken to mean a man who can act on his own initiative, plan his own work and carry it out, and solve his own ordinary problems without assistance. According to this viewpoint, a machinist is not a man who knows how to operate a milling machine or a boring mill; he is a man who thoroughly understands the operation and the maintenance of all standard machine tools and is sufficiently familiar with their fundamental principles to grasp the construction and operation of all kinds of special machines.

It is sometimes maintained that men of this type are no longer needed because of the rapid introduction of automatic machinery in modern manufacturing plants. While it is no doubt true that the relative number of skilled mechanics in modern industries is less than formerly, nevertheless the skilled mechanic cannot be dispensed with and is required in great

---

[1] Vice-President and Works Manager, The Falk Corporation. Mr. Falk was graduated from the University of Wisconsin in 1906 and entered the employ of the Falk Corporation as assistant to the superintendent. In 1923 he was appointed to his present position.

Contributed by the Standing Committee on Education and Training for the Industries and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

numbers. Even the most highly developed mechanical manufacturing plant requires trained men for the planning and supervision of work, for the adjustment and repair of machinery, and for the building and maintenance of the many fixtures and appliances without which automatic manufacture is impossible. Moreover, many essential industries cannot be established on an automatic basis. In the manufacture of locomotives, ships, machinery for power plants and steel mills, for instance, skilled mechanics will be employed for years to come just as they have been in years past. After all, only a limited number of commodities are in such demand that thousands of units must be produced every day so that automatic production may be justified.

In the development of apprentice training for Wisconsin industries, these various factors and many others were kept in mind and they served as a basis for planning the work.

The Apprenticeship Law

The apprenticeship law of the state has a twofold purpose: to prevent exploitation of boys and young men and to establish and maintain high standards of skill in the trades and industries.

The decline of apprentice training with the close of the last century was no doubt due in large measure to certain abuses which had developed. Boys and young men were engaged as apprentices, but were actually employed as common laborers. In many cases no systematic instruction was provided and they were given no change of work. If a boy happened to become proficient in the operation of a screw machine or in the making of a certain mold of which a large number were required, he was kept on those operations indefinitely, regardless of the restriction thus imposed upon his training, merely because the foremen or department head was too indolent to break in another operator.

The law is intended to do away with this evil. It provides close supervision of the work of apprentices by the state, so that conditions of this kind are practically impossible.

Again, the standards of skill in a trade or industry will depend almost exclusively upon the training which is given. In those trades in which high standards of apprenticeship and instruction are regularly maintained, high standards of workmenship can be found. On the other hand, when training is neglected, a low grade of skill prevails, such as exists at the present time in only too many industrial and building trades.

In order to secure high standards of training, the law requires that the Industrial Commission of the state must approve all work programs for apprentices, and if these work programs are not sufficiently broad and not sufficiently strenuous, they are rejected, and the employment of an apprentice under the plan is forbidden.

The various provisions of the law have been established in order to carry out its twofold purpose. In the first place, the law requires that all apprenticeship agreements must be executed in writing in three copies, of which one copy goes to the apprentice, one copy to the employer, and one copy to the state, so that an official record of the apprenticeship and of the work to be done shall be found on file with the state. The agreement, or indenture as it is called, must contain the names of the parties and the age of the apprentice, the trade or occupation which is to be taught, the number of hours to be devoted to work and the number of hours to be devoted to school instruction, the schedule of work or processes, the schedule of compensation to be paid, a statement that a certificate will be furnished the apprentice upon completing his course, and any special provisions which may exist, such as the payment of a bonus for the completion of the course.

The law requires that every apprentice must be 16 years of age or over and that he must attend school four hours per day during the school year until he has completed 400 hours of school work. Hours of overtime work are limited. Also, a penalty is provided for violation of the indenture by either the employer or the apprentice. The Industrial Commission is appointed by the law to supervise apprenticeship in the state, and all public school officers are required to provide facilities for the instruction of apprentices in their respective communities.

The apprenticeship law was passed at the urgent request of representative groups of employers and mechanics in 1911. It has been revised a number of times since then.

Although the Industrial Commission of Wisconsin is charged with the administration of the apprenticeship law, only important and final decisions are left to the commission itself. The practical management of the apprenticeship affairs of the commission is intrusted to a supervisor and several deputies. It is the function of the Industrial Commission to issue such rules and regulations as may be necessary for carrying out the provisions of the apprenticeship law, to approve indentures or contracts which are not valid without such approval, to adjust grievances which may arise between apprentices and employers regarding the fulfilment of the provisions of the agreement and other matters, to issue certificates of graduation to apprentices who have completed their course, and to cancel apprenticeship agreements when it appears necessary to do so. After a probationary period of three months has been passed, the indenture may not be canceled except by the Industrial Commission and only for valid reasons.

The commission carries on a promotional campaign at all times in order to develop apprenticeship in the state as much as possible.

In the administration of the apprenticeship law, the Industrial Commission has been eminently far-sighted and reasonable. Decisions have been made carefully and with due regard for the attitude and position of all parties concerned.

The regulations and decisions of the Industrial Commission regarding apprenticeship are based very largely upon the advice and assistance of joint trade committees which have been established. These committees are made up of three or four representatives appointed by the employers' association of the trade or industry in the state and an equal number of representatives of the mechanics. These committees meet under the auspices of the Industrial Commission and determine, first, the length of apprenticeship which shall be standard in the trade and, secondly, in a general way, the course of work to be pursued by the apprentice during this time. Thereafter, the committee meets at suitable intervals to advise the commission in the adjustment of the problems which arise in the administration of apprenticeship and in the correction and adjustment of the apprenticeship policy of the commission.

In addition to these various trade committees, there is a general advisory committee, composed of representatives of employers and employees, which consults with the commission in those matters which affect apprenticeship in all trades, but which do not involve the peculiar problems of any one.

School Requirements

Compulsory school attendance for apprentices is one of the most important features of the apprenticeship law and probably the outstanding characteristic which distinguishes modern apprenticeship from the apprentice training courses which began to disappear about 30 years ago. This legal requirement places school attendance where it belongs as an essential and adequate training and not merely a finishing touch which the employer may add to his course or omit as he wishes. The law recognizes that schools for apprentices in the ordinary

shop are out of the question because of lack of organization of the work and lack of suitable instructors and equipment. Moreover, the atmosphere of the manufacturing plant is not ordinarily conducive to effective study.

Accordingly, the apprentices are required to attend the local vocational school four hours per week during the school year in order to receive the related instruction necessary for their apprenticeship, and the school authorities are made responsible for proper teaching of mathematics, blueprint reading, elementary science and mechanics, industrial subjects, and even some cultural subjects.

Individual instruction has been instituted in most of the apprenticeship classes so that each apprentice may progress as rapidly as his ability permits. Although the law requires that each apprentice shall attend school until he has completed 400 hours of school work, the Industrial Commission reports that the majority of apprentices are required by their employers to attend the vocational school during their entire apprenticeship, which is satisfactory evidence that the value of school work is appreciated by the employers in the state.

In those communities in which no vocational school has been established, special classes are arranged under the supervision of the State Board of Vocational Education, and part-time instructors for these classes are secured from the local trade or industry or from the members of public school faculties. In other cases a number of communities together engage an instructor in a given trade who divides his time between the communities. The salaries of these itinerant teachers are paid partly by the federal government, partly by the state, and partly by the local communities, just as in the case of regular vocational teachers.

Growth of the Movement

Favored by public interest and the enthusiastic support of influential individuals, apprentice training in the state of Wisconsin has developed rapidly in recent years. A total of 3350 apprentices are employed in the state, distributed in 10 building trades, 11 metal trades, 15 printing trades, 6 garment trades, 5 automobile trades, 11 railroad trades, and 22 miscellaneous trades. Between August 1, 1915, and August 1, 1930, a total of 2567 apprentices had completed their courses. This may not appear to be a great number, but two factors must be taken into account. In the first place, there were very few graduations in the earlier years of the movement. At the present rate of 325 apprentice graduations per year, the next 15 years will show a great increase in this number. Also, it must be remembered that Wisconsin is not one of the outstanding industrial states so far as total value of product and number of employees are concerned.

By far the largest group of apprentices is employed in the metal-trades manufacturing industry, and as this industry in Wisconsin is centered very largely in and about Milwaukee, the development of apprenticeship in the shops of the Metal Trades Association of Milwaukee may present a fair picture of the growth of the movement in the state. Apprentice training of one kind or another had been carried on in a limited way throughout the machinery-building industry of the state metropolis. It appears that the first concerted effort for the development of apprenticeship in the industry was made in 1901 when the Milwaukee Foundrymen's Association (no longer in existence) appointed a committee to investigate apprenticeship, and this committee recommended that all members of the association adopt a plan of apprenticeship which had been established by the National Metal Trades Association. Apparently there were very few apprentices in training at the time, and the training must have been haphazard and disorganized.

In 1905 the important machinery builders and foundries of Milwaukee united in an organization known as the Metal Trades and Founders Association, and one of the purposes of this organization was the promotion of apprentice training. A survey made at the time disclosed 217 apprentices employed in the shops of the members, although it is not known among which trades these apprentices were divided. In 1908 a new committee of this organization was appointed, and it recommended a uniform apprenticeship contract and the awarding of a diploma. In the following year this committee adopted 11,000 hours as the term of apprenticeship and established pay rates of 10, 11, 12, and 15 cents per hour during each of the successive four years of the apprentice course.

It was at this time that the need of a law was felt in order to stabilize and dignify apprenticeship work, and members of this committee took an important part in the deliberations which preceded the enactment of the law in 1911.

In that year there were 460 apprentices in the shops of the members of the National Metal Trades Association in Milwaukee. This number has grown to 1068 at the present time. The total number of employees in the member shops is ordinarily in the neighborhood of 20,000, so that there is about one apprentice for every 16 or 17 employees.

In these machinery-building plants, apprentice graduates have become an important factor in the development of strong organizations of mechanics, foremen, inspectors, engineers, and other highly trained personnel. In many of the plants, the apprentice graduates have become the principal source from which important members of the organization are recruited. In one large organization which operates three foundries, no molders have been engaged for a number of years other than graduates of their own apprentice-training program. This may be considered a singular achievement in view of the attitude of American people toward the foundry trade. In another large organization no foremen have been hired for ten years or longer other than graduates of apprentice courses in the plant, with the exception of department heads who have become members of the organization when smaller manufacturing plants were absorbed.

Evidence of the prestige which apprentice training has acquired in the community is the attitude toward apprenticeship of the people in the community in general. For instance, public school officials cooperate in the work, which is somewhat surprising in view of the fact that the traditional attitude of the school principal is that higher education is or should be the chief goal of all primary school and high school students. In three of the city high schools, apprentice supervisors have been invited to address the students on the benefits of apprenticeship in metal trades shops. Supervisors have also addressed students in three suburban high schools. In one of the high schools, which is located in a very fine residential district, the principal has permitted a foundry representative to recruit steel-foundry apprentices from the members of his senior class at the close of the school year. High school students are frequently conducted through shops under the supervision of apprentice directors in the plants. The result of this public attitude is that there are many more applicants for apprentice training in the shops of the city than can be accommodated, and practically every plant maintains a considerable waiting list from which the best applicants may be selected.

The apprentice-training work of the members of the Metal Trades Association of Milwaukee is carried on under the direction of a major committee of five plant executives who meet once or twice a year to direct policies and decide important questions and a general operating committee made up of apprentice directors and supervisors which meets monthly and

carries out the plans and directions of the major committee. These committees cooperate with the Industrial Commission, and through their efforts apprentice training in the Metal Trades Industry has been standardized to a large extent.

Moreover, excellent progress has been made in other trades and industries. The Industrial Commission of Wisconsin has available considerable information regarding the work in a number of the important trades. In the building trades, for instance, the plumbers have made great strides. Nine years ago there were only 20 indentured plumbing apprentices in the state. At the present time there are 458. This excellent showing has been the result largely of the influence of certain persons in the business who have taken an active interest in promoting apprentice training. The state advisory committee of the plumbers has been very active, and the state plumbers associations are in close contact with the Industral Commission. Four itinerant instructors for plumbing apprentices are engaged, and their services are available to 16 different cities. In 15 of these cities there are local joint apprenticeship committees for the plumbing trade.

There are 46 indentured apprentices in the painting and decorating trades. The promotion of apprenticeship in this trade has been rather difficult due to the fact that the majority of the employers are not members of any single association and many of them have not themselves completed a full apprenticeship in their trade. Three itinerant teachers are employed and serve 12 cities, although their services are not limited to the teaching of apprentices, but classes are held also for others in the trade. In an effort to develop apprentice training in this field, the Industrial Commission has held a number of mass meetings for employers and journeymen, and these meetings will be continued in the future.

In the electrical trade, 62 apprentices are employed in the state, and three itinerant teachers are provided for 12 different cities. Interest in apprenticeship has been stimulated largely by the itinerant teachers, as electricians themselves have as a rule shown little interest.

Although the carpenter trade is the most important of the building trades and has the largest number of workers, there is such a diversion of specialties and lines in the trade that apprenticeship has not flourished as it should and only 73 apprentices are indentured. In the bricklaying trade, conditions are much better, with 173 apprentices. Most of these are employed in closed shops, and the promotion of apprentice training in the trade has been the result almost altogether of the cooperation and energy of the labor organizations.

In the printing trade there have been labor troubles in recent years which have had a detrimental effect upon the development of apprentice training. Nevertheless, 72 apprentices are employed, and it is expected that this number will be increased very shortly.

Work Only Begun

It must not be inferred that the Industrial Commission of Wisconsin or the leaders in any single trade or industry of the state are satisfied with what has been accomplished in apprentice training. The extent and development of the work appear strikingly inadequate when it is compared with the number of persons engaged in the various trades and industries and the requirements of the trades and industries resulting from natural growth. In speaking before this society in 1924, Dr. Magnus Alexander, of the National Industrial Conference Board, pointed out that 5 per cent of the skilled employees in every industry are lost every year as the result of death, retirement, transfer to other occupation, and other normal causes. Accordingly, there should be graduated in every trade a sufficient number of apprentices per year to replace the 5 per cent who are thus lost. And to graduate 5 per cent of the total skilled mechanics per year, it would be necessary to employ a number of apprentices equal to at least 20 per cent of the total skilled mechanics in the trade. It is perhaps unnecessary to explain that such a large quota of apprentices is probably not employed in any single trade or occupation in the state. However, the movement has been started, methods have been established and proved to be successful, and upon this foundation it must be possible to build in the years to come a constantly increasing apprentice-training activity which should in time be adequate.

Law Not Essential

Also, it would be a mistake to assume that a state law is in any way essential for successful apprentice training. Excellent proofs to the contrary are the very fine apprentice-training programs which have been carried out in important industries for many years in states in which no apprenticeship law has ever been passed and in which none is contemplated. After all, the individual employer can establish apprentice training if he wishes to do so and is willing to give to the subject the amount of time and attention which is necessary. The law, after all, merely standardizes and recognizes the apprentice training and provides an agency through which apprenticeship may be promoted and controlled. Undoubtedly this is of great benefit to the apprenticeship movement, but it would be absurd to state that apprentice training cannot be carried on without a state law. After all, successful apprenticeship must always be the result very largely of the enthusiasm and energy of far-sighted individuals, and unless these are to be found in the trades and industries, a law would never come into being in any case. The Wisconsin apprenticeship law is the result and not the cause of such energy and enthusiasm.

Discussion

Fred H. Colvin.[2] An outstanding feature of the paper is the statement at the end of the first paragraph, "In order to make apprentice training a social factor," etc.

While more and more machine operations are bound to be done on automatic or semi-automatic machines, there always will be a need for skilled mechanics to set them up and keep them in operation. Likewise the need for, and the possibility of, improved machines is as likely to be realized by bright men in the shop as by engineers in the designing department.

The sooner it is realized that this is an age of cooperation in such matters rather than of haphazard individual effort the better.

There is, however, a place for individualism in apprentice training; in the handling of the apprentices themselves. Boys differ widely in temperament, which may reflect home conditions that vary greatly. This does not mean favoritism, but rather an intelligent study of individual characteristics and possibility. The diffident boy will benefit by encouragement and the overaggresive lad by judicious repression, to the improvement of both and the development of first-class material.

[Further discussion of this paper jointly with that of C. F. Bailey, by E. W. Kempton, Morris S. Viteles, L. P. Arduser, and John A. McCarthy, will be found following the Bailey paper.]

Author's Closure

In reading over the discussion and comments which were made concerning the papers by Mr. Bailey and myself, it is gratifying to find so many people, representing so many different attitudes and fields of activity, who apparently are fully in agreement with us. In fact, it seems remarkable that the movement does

[2] Editor, *American Machinist*, New York, N. Y. Mem. A.S.M.E.

not grow more rapidly when it very seldom if ever happens that any organized, coherent arguments against the apprentice-training system are formally presented under conditions which make it possible to answer them effectively. For instance, it is very seldom that a speaker appears at a meeting of this kind and opposes apprentice training.

Here and there something appears in the discussions which is startling in view of our experience at Milwaukee, but local conditions may be so different in other places that they may justify what would be considered radical procedure in Wisconsin. For instance, the question of limitation of apprentices to high-school graduates is raised once or twice. In Wisconsin it is felt that boys of common-school training can find a place as skilled mechanics in industry and may be trained for this work with much success. In fact, the majority of Wisconsin apprentices are not high-school graduates, and employers may even be found who refuse to engage high-school graduates, as they very much prefer the graduate of the eighth-grade course. It is generally accepted, however, that the high-school graduate apprentice is more likely to develop into the kind of man who can assume a responsible position.

Mr. McCarthy refers to the tendency of certain manufacturers to discontinue apprentice training at the present time because of the depression. Apprentice training, of course, must be a continuous institution. We would not close our public schools during a period of depression in order to relieve the taxpayer, because we fully realized that public education must be a continuous process. The same argument should apply to apprentice training.

Moreover, it is not merely important to maintain apprentice training during periods of depression, but it is even true that a period of depression is a better time to institute apprentice training in a plant than any other. In the first place, officials and foremen can give to the work the time which is required; secondly it is easier to find desirable candidates for apprenticeship; and finally, the public will appreciate that apprenticeship is a legitimate enterprise and not a method to get cheap labor.

MAN-53-3

A Generation of Progress in Human Engineering

By WILLIAM A. HANLEY,[1] INDIANAPOLIS, IND.

In this paper the author has considered two manufacturing plants of about the same size, and in the same state, and contrasted conditions under which labor worked in these factories 30 years ago and conditions as they are today. The subjects covered deal with nearly every phase of factory employment. Some interesting conclusions are drawn.



IN THE LAST 30 to 35 years there have been fundamental changes in the status of factory workers, many of which are not fully known to the younger generation, nor to those not intimately connected with factories. Some of these changes are so striking that conditions in the nineties will seem to many as representing a period much earlier. The object of this paper is to show the conditions which surrounded the worker at that time and those under which he works today. By worker is meant not only the mechanic or artisan, but also the man who has no trade and who is not skilled in any craft. They are the men who sweep floors, use picks and shovels, run wheelbarrows, unload cars, haul trash, handle coal or ashes, and do such other tasks requiring no especial experience nor training. Many men from the farms fall into this group. They have no unions to help fix working conditions. They are the first class reduced in numbers and wages in hard times and their employment is never very stable.

Because comparisons are sometimes unfair, the author in this instance takes two concerns with which he is very familiar. Both of these concerns were or are located in Indiana, not very far apart. The first concern manufactured iron and steel, and the second manufactured drugs and chemicals. The author's family has been intimately connected with both companies, so that the knowledge is first hand.

Both corporations had as workers a large percentage of native Americans, very few foreigners being in either organization. The concerns were about the same size, each employing about 1500 people. Both companies have been successful, although the iron-and-steel property was merged at the close of the century, and because of fuel conditions, later dismantled. Both industries were operated by men of high ideals who believed in the policy of fair play. Each concern had loyal and faithful employees.

The place of labor in industry, even of the poor common laborer, has changed and changed greatly in the last 35 years as these comparisons will show.

Wages

In 1895 the wage of the common day laborer was 12.5 cents per hr. About the close of the century this was increased to 15 cents per hr. The wage today is 45 cents per hr., or just three times what it was 30 years ago. Now 45 cents today buys about $1^7/_8$ times as much in useful and necessary commodities as 15 cents did in 1900. The purchasing power of an hour of labor has increased in just this proportion.

In 1895 all labor was paid either by the hour or by the piece. Today every job, where at all possible, is placed on a standard minutes basis and the employee is paid a bonus for all produced above the standard. This is true even of porter work.

Hours

The hours for common labor in the nineties were from 6:30 a.m. to 5:00 p.m. with 30 min. for lunch. A day was of 10 hr. and a week of 6 days This made a total of 60 hr. per week. Shopping and business transactions were limited to Saturday night; recreation and rest was limited to Sunday. In winter the worker went to the factory in the dark and came home in the dark.

Today the worker puts in $44^1/_2$ hr. per week; that is 8 hr. on Monday to Friday, inclusive, with $4^1/_2$ hr. on Saturday. Plant operations start at 8:00 a.m. and are over at 5:00 p.m. The lunch period at noon is 1 hr. The plant closes at 12:30 Saturday afternoon. It is interesting to note that the worker goes to his job $1^1/_2$ hr. later in the morning than he did in 1900, has 1 hr. at noon for lunch, where he had $^1/_2$ hr. before, and that he has Saturday afternoon where he formerly worked all day on Saturday. On a recent referendum (something unheard of 30 years ago) as to whether the employees preferred a 30-min. lunch period instead of the hour with 30 min. earlier closing, the majority vote was to make no change and to keep the lunch period 1 hr.

Hiring and Discharging

In 1900 the common laborer would be at the works seeking a job between 6:00 a.m. and 6:30 a.m. He would apply to one of the labor foremen, a man receiving about 25 cents per hr. Physical characteristics were usually his only recommendation. If hired at all, he might be kept for a long or short period depending on how he satisfied the foreman. He made no written application, his age and previous employment were not recorded, and he either satisfied the foreman or was dismissed by the foreman. It was a very simple affair, about as harvest hands are hired in the grain-producing states, yet many men advanced under the old system, crude as it was. Its principal drawback seems to have been that there was no real selection, no opportunity for the thinking worker, that many filled positions for which they were unsuited for years, and that foremen were given great powers of

---

[1] Director of Engineering, Eli Lilly & Co. Mem. A.S.M.E. Mr. Hanley was born in Greencastle, Ind., on December 13, 1886, and received his education at St. Joseph's College, Rensselaer, Ind., and later at Purdue University, where he received the degree of B.S. in Mechanical Engineering.

Mr. Hanley's first position was with the Republic Iron and Steel Co. in Muncie, Ind., in their rolling mills. He became associated with the Broderick Boiler Co. as assistant to the manager. In 1911 he joined the staff of Eli Lilly & Co., manufacturing chemists, at Indianapolis, Ind., for the purpose of developing a gelatin-capsule manufacturing plant. Some time later Mr. Hanley's duties were broadened to include the supervision of all engineering, construction, maintenance, mechanical design, power generation, and plant equipment, with the title of director of the engineering division. This is his present position.

Contributed by the Management Division and presented at the French Lick Meeting, French Lick, Ind., Oct. 13 to 15, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

discretion, and personal differences with the foreman often controlled the destiny of a worker instead of his value to the company. The records of the company showed no difference as to whether the worker was discharged for drunkenness or laid off because of slack business conditions.

At the present time the man seeking employment makes his application in writing from 24 hr. to a month or more before he expects to go to work. The application shows his age, family conditions, parentage, education, previous employment, and his own desires as to type of employment and wages expected. The man has a personal interview with an employment supervisor, especially trained in human relations. If the man is employed, this employment supervisor becomes his guide, counsellor, and friend in helping him to success in his new work. He will advise the employee on any subject from getting along with his wife and children to his family budget or buying a home. The employment supervisor has nothing to do with the actual duties of the man once he is assigned. He is then under the foreman of his department, but the employment supervisor is responsible for the employee being started in work for which he is suited, being transferred when the work is unsuited, and in getting a fair deal with the foreman. The employment supervisor is a staff officer rather than a line officer, and his rank is higher than the rank of foreman of a department. No workmen can be discharged except through and by the employment department.

Seasonal Changes

In times past there would always be a part of every year when business was very brisk and another part when it would be slack. The general practice was to hire and to lay off employees to meet these conditions. Two distinct advances have been made from the old scheme. The production demands are now anticipated, and merchandise at special terms is offered during slack seasonal periods, so that the variation in monthly factory production has been practically eliminated. This means steady work for the employees. It also does away with temporary help. The second change covers times of depression. Formerly departments were closed and employees forced into idleness. The new method is to reduce working hours instead of reducing numbers of workers. This gives all employees some income and keeps the whole organization ready to function as soon as normal conditions return.

Rest Periods

Formerly people worked 8 or 10 hr. per day continuously, stopping only at noon. Studies of fatigue, particularly on fast repetitive operations, have shown that rest periods are very beneficial. Accordingly two rest periods of 10 min. each, one in the morning and one in the afternoon, give the worker the relaxation which is so helpful to both muscles and nerves.

Safety, Hospital Facilities, and Insurance

Thirty-five years ago very little attention was paid to safety. It was expected that every man would take care of himself, regardless of the conditions under which he worked or the hazards which might prevail.

In the steel business there were several men who lost the sight of one eye and one or two who lost the sight of both eyes due to the explosion of the hot metal. There were no regulations as to the wearing of goggles, and no written instructions as to hazards which should be avoided. There was not even any provision for taking care of injuries. The author recalls very well witnessing an accident wherein a blacksmith striking at his anvil had a piece break, which made a very ugly wound on his leg. The man was bleeding profusely and the only bandage which could be provided was the napkin out of a fellow workman's dinner basket. Another injury to which the author was a witness left the workman in such a state that he could not walk. He was carried home, about two blocks away, on a door, as there were no cots or stretchers available. Another man had one arm severed on an alligator shears. No guard of any kind was provided on this shears, and when a guard was supplied it was taken off by the operator as being a nuisance to him.

The theory of the employee was that he was to take care of himself, and to this theory the employer agreed. There was no safety committee, no first aid equipment, and no medical attention provided.

Great progress has been made in 30 years, and now the plant has a well-equipped hospital with a graduate nurse on duty at all times. People who come to work in the morning with a cold or who feel indisposed from any cause are encouraged to go to the hospital and receive preventive remedies. A constant system of education is maintained in an attempt to keep the health of the workers on a higher plane. Warnings are given about living conditions which will result in tuberculosis, typhoid fever, pneumonia, etc. Minor casualties, such as scratches, cuts, or burns, must be treated immediately. A practicing physician is on hand 5 hr. out of every day to give advice and to treat the employees for minor ailments. The theory, of course, is not to replace the family doctor, but rather to help him and to keep the employee from falling into the hands of incompetent practitioners. There is no charge for any of this service.

A safety committee carefully combs the plant for hazards. Any accident causes a complete investigation, with the result that there are virtually no repetitions of accidents of the same type. Machinery is guarded carefully when it is installed, and it cannot be adjusted except when shut down. Goggles are provided where eyes are endangered, and gloves are provided where possible poisoning is involved, window washers have safety belts, and scaffolding is always erected with the thought of protecting the worker. Sometimes these goggles, gloves, etc. slow up the worker or cause him inconvenience, but the management has an iron-clad rule that these precautions must be followed out or the workman cannot go ahead with his duties.

In times of epidemics every employee in the plant will be vaccinated, have his throat swabbed or given such other preventive help as the occasion demands. The employer seems now to realize that it is just as necessary, or maybe more so, to keep his employees in good condition on the job, as it is to keep his machinery well maintained and ready for use.

Thirty-five years ago when a man died his family buried him, and unless he had provided for this contingency the family budget was given a severe blow from which it took many months to recover. For some years the company with which the author is connected has had group insurance for every person in the factory. The major cost of this insurance is borne by the corporation. For those receiving the smallest wage, there is provided $1000 for ordinary death with $2000 for accidental death. In addition there are weekly sick benefits, accidental dismemberment insurance, total- and permanent-disability benefits, and non-occupational accident benefits. The employee pays for all of the above about $1 per month. This covers the job of the least-paid employee in the organization. Those receiving higher wages go into classes Nos. 2, 3, or 4. The last class provides for $4000 ordinary life insurance with an additional $4000 in case of accidental death, and dismemberment and disability insurance. All this is at a cost to the employee of $3.20 per month.

This, however, does not tell the whole story. In the case of a person employed by the company for 5 years, who is head of a family, the company pays 6 weeks' wages to the family of such employee to cover the period in which the family are adjusting themselves and providing another income. Where the employee has been with the company 10 years, 10 weeks wages are paid,

and this keeps the family during the readjustment period from having to draw on what savings they have accumulated.

Luncheon and Recreation

Thirty-five years ago the workman had his lunch packed in a bucket or basket and brought it from home with him. The campaign of 1896 had as its popular cry, "The Full Dinner Pail." No facilities were provided around the establishment for the employee to get any food or even to keep food which he might bring with him. The corner saloon, and it is to be hoped that it is gone forever, was usually the only place he could get recreation and refreshments in or near the factory. The workman could use no recreation facilities in connection with the factory as the 30 min. provided for lunch was hardly more than sufficient to give him time to consume his food.

At this time a fine cafeteria provides food at a nominal cost to all employees, and they have one hour in which to eat, rest, or recreate. Different departments go to the cafeteria every few minutes, so that there is a steady line passing before the lunch counters and to the inviting tables. In the old days people had to eat what they brought in the department where they worked. Now a clean, well-ventilated cafeteria is the only place where lunches can be eaten around the plant.

For the interval after lunch there are provided recreation rooms for both men and women. The employees play cards, read, or relax. Each rest room is equipped with a piano, radio, or phonograph. In connection with the girls' recreation room, there is a room with cots where those who are indisposed can rest. In connection with both recreation rooms there are branch library facilities from the city library. This enables employees to use the facilities of the public library through the instrumentality of the factory.

Wash Rooms and Lockers

Thirty-five years ago scant provision was made around factories for taking care of employees' clothes and providing them with adequate wash-room facilities. Now every employee has a safe, clean place to put his clothes, and people leaving the factory are just as clean as when they come to the factory in the morning. Wash-room facilities have not only increased in numbers and been improved in type, but they are so located as to make them convenient to the employees.

The self-respect engendered in the individual, who can go on the street at the close of a day's work without the factory grime, is well worth considering.

Vacations and Holidays

In the nineties the only people who enjoyed vacations were those who worked in the office or who owned the industry, that is, salaried workers. Those who did piece work or who worked by the hour received nothing except when they were on the job.

At the present time every person, no matter in what capacity, who has been in the employ of the company one year, receives one week's vacation with pay. When they have been in the employ of the company 5 years, they receive 2 weeks' vacation with pay.

In addition to this (and the author knows of no parallel case) every worker is paid for 6 holidays in the year; that is, New Year's, Memorial Day, Independence Day, Labor Day, Thanksgiving, and Christmas. These are real holidays to the employees, and their pay envelopes do not reflect the fact that they have not worked on these holidays.

Pensions

In the old days when a workman could no longer carry on his regular task he was given some comparatively light work to do. He was compelled, however, although he had reached the age when he should retire, to continue to work.

Today a pension plan covering all employees that have reached 60 years of age is provided. This does not mean that all employees must retire at 60, because many are very active and can give good service for some years beyond that age, but it does mean that any one in the employ of the company 20 years or more, who has reached the age of 60, can retire if he so desires and receive an income for the rest of his life.

Promotions

In former times every job had a certain rate. The persons on such jobs got such a rate no matter how long they were on it. To get higher wages it was necessary to go to another job, either in the factory where they were employed or in some other establishment.

Much progress has been made, and now every employee is given a starting wage, and it is anticipated that this wage will be raised in 3 months if the employee has been satisfactory. Every 6 months the foreman of each department, with the head of the division, considers each individual case, as to whether the employee should be continued on his present job, moved to a better job, or be given a wage increase. Thus semi-annually the management carefully considers the status of each employee.

Attendance Bonus

Another feature of the modern factory is the attendance bonus. This is an extra payment over and above wages paid to those who are on time and present each day at their post of duty. This amounts to $4.50 per month and tends to create promptness and regularity.

Automobiles

In the old days employees came to work in three ways—that is, they walked, rode bicycles, or came on street cars. In a recent survey it was found that there was one automobile being driven to the plant for every four factory workers. These cars usually carry 3, 4, or 5 workers, so that at least two-thirds of all employees travel to and from the factory in automobiles. To care for these cars, parking and garage facilties have been provided by the management so that the owner can feel that his car is protected while he is performing his duties during working hours.

Methods and Standards

Another division in the organization not heard of 30 years ago is the group that visits departments, studying the duties of the different workers, and determining the one best way to do each job. Usually the one best way is both the simplest way and the easiest way. This department makes layouts, locating machinery and tools, so that the worker can perform his duties most efficiently and with the least exertion. Mechanical power, conveyors, and hoists are substituted in every case possible where human power was used in the past. Drudgery has been reduced to a minimum.

Sentiment

It has been said that there is no sentiment in business, but wherever the human heart is, there is always sentiment. On Christmas eve each year all employees are guests of the company at a turkey dinner in the cafeteria. Christmas decorations have changed the room to one of feast and pleasure. Christmas carols by trained groups of employees sing out the glad tidings. Those in the service for 25 years have a special table where they reminisce and dine with the president of the corporation. Each of the 25-year group is given a Christmas check equal to 10 per cent of his year's earnings. Following the dinner there are department parties as the afternoon is given over to recreation.

Happy faces portray happy hearts and all leave the company premises with the true Christmas spirit.

Conclusion

Having enumerated in detail many of the industrial changes, it is interesting to list the results both to the employee and the employer.

To the employee there has come: (1) The higher standard of living, (2) greater respect for himself and from his fellowman, (3) better health, (4) no uneasiness about the future because he is insured against accident, loss of heath, and old age, and (5) greater happiness and contentment.

Surely these things are worth while because they tend to raise the standards of health and citizenship, and to give greater happiness to workers and their families.

The results to the employer because of these changes have been: (1) A more highly selected class of employees, (2) fewer interruptions due to sickness, (3) no absences except for unavoidable reasons, (4) a real interest in company affairs by intelligent and interested workers, (5) a minimum of labor turn over, (6) a decrease in avoidable wastes, (7) better quality of merchandise, (8) more output per employee, and (9) no strikes and no labor problem.

The question will arise in the mind of the skeptic as to who pays for all these innovations. Are they taken out of corporation profits or are they added to the cost of merchandise? The answer is that nobody pays for them. The difference in output per employee of a factory having selected, contented, healthy, and interested employees is 25 to 30 per cent greater than the factory which regards its employees as human machines only, and this increased output pays the bill. Cost sheets for over 20 years verify this conclusion. After many years of study and experience the statement can be made that one of the most valuable asset of any corporation is a corps of loyal, intelligent, and satisfied workers, and that not only the corporation profits thereby, but a distinct service also is rendered to the community and the nation.

MAN-53-4

Management by Exception

By CHESTER B. LORD,[1] NEW YORK, N. Y.

The exception principle, as approved by the first leaders of industrial management, is expounded and advocated. It is described as a natural method of scientific management that directs when direction is sufficient and that compels when compulsion is called for, but that does not interfere with matters which are progressing in orderly compliance with a planned procedure, but which has positive methods of detecting and correcting variations from that procedure. Four rules of management by exception are stated.



WITH scientific management came the problem of its application to industry. The principles laid down by Taylor were epochal, but a positive method of applying them to industry was lacking. As a result each one who accepted the principles interpreted them and applied them according to his own experience or understanding. It is not surprising, therefore, that there sprang up almost as many schools of management as there were engineers. Neither is it strange that, in some cases, system became paramount and in the course of time was considered a major factor of management control instead of a mere expression of individual opinion or preference as to the method to be followed in applying management principles.

In his day Taylor was accused of not knowing how to handle men, while Gantt disagreed more or less violently with some of the higher executives and was called intolerant. It seems impossible today for any one to review the work of these two men and say that they did not understand other men or that they did not have the executive viewpoint.

Taylor and Gantt were practical. The principles they advocated were the result of many years' labor and experiment. Therefore they spoke a language different from that of many who attempted to apply Taylor's principles and who inferred the principles were at fault when the application failed. It seems more probable, in the light of the later work and utterances of both Taylor and Gantt, that both men were exasperated by opposition and disappointed because there was no existing mechanism into which they could permanently transfer the skill and knowledge they possessed. Hence Taylor's impatience and his insistence upon his own interpretation being followed. Gantt was similarly situated. He believed that many of the higher executives were opposed to scientific management. The real trouble, however, was that the executives could not visualize cause and effect on such a scale as Gantt himself could.

Robert T. Kent described the situation in his paper on the "Use of Economic Manufacturing Quantities" in the following words:

"In introducing scientific management methods, one of the prime requisites of success consists of so regulating the dose to be given that all the medicine will be effectively and completely absorbed. In other words, it appears essential that those who have delegated themselves to the job of introducing the application of science to industry must prepare themselves to be satisfied with accomplishment which falls far short of the goal which they are able to visualize. . . . Some of our most learned and accomplished disciples of scientific management have fallen far short in the accomplishment of results by not being willing to be satisfied with a 30 per cent accomplishment of the goal rather than 100 per cent accomplishment."

This is what both Taylor and Gantt failed to realize.

That Taylor, Gantt, and other leaders in industrial-management methods recognized the deficiencies in systematic management is proved by their continued advocacy of the exception principle. Taylor asserted that the extension of the principle to every field of industry was a vital necessity. Gantt labored continually to make it possible. Gilbreth visualized its necessity in management and the benefit to be derived from its use even more definitely. As far as can be learned, he was the only one who visualized its extension to minor executives. He declared that it apportioned the load to the shoulders which must bear it, that it made good executives of mediocre ones, and gave good ones more time for constructive planning.

Under the circumstances, the natural question is: If the importance, and even the necessity, of using the exception principle was recognized by these and other leaders who had the power and the opportunity to use it, why was not some definite action taken toward its introduction? The same circumstance that handicapped the introduction of scientific management and caused the teaching of Taylor and Gantt to be futile in many cases, also prevented the introduction of management by exception. This was the lack of the proper mechanism for its introduction.

The striking thing about the whole matter is that these men should have been so insistent upon the use of a method of management that had no history, no rules for its application, and not even a clear definition. In spite of these deficiencies, the desire for exception management has always been inherent in people of the executive type, and it is used by them to the limited extent that their businesses will allow. This desire for its use undoubtedly arises from an instinctive knowledge that there must be some natural method of management that directs when direction is sufficient and that compels when compulsion is called for, one that does not interfere with matters which are progressing in orderly compliance with a planned procedure, but which has positive methods of detecting and correcting variations from that procedure.

It is possible that the knowledge of such a type of management is inherited from a line of ancestors who used that method of bringing up their children, and it may also be possible that such a method of bringing up has been responsible for the executive type.

When exception management is contrasted with systematic

---

[1] Associate Editor, *The American Machinist*. Mem. A.S.M.E. Educated at University of Omaha; two years in railroad and harbor works, Brazil and Argentine; assistant foreman, R. Hoe & Co.; foreman, General Electric Co.; assistant superintendent, Victor Talking Machine Co.; general superintendent, Wagner Electric Corp.; plant manager, Advance Rumely Co.; partner, Wallace, Delaney & Lord; superintendent of shops, Newport News Ship Building & Dry Dock Co.; permanent member, Inter-American Council.

Presented at a meeting of the Metropolitan Section of The American Society of Mechanical Engineers, Feb. 7, 1930, sponsored by the Management Division.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

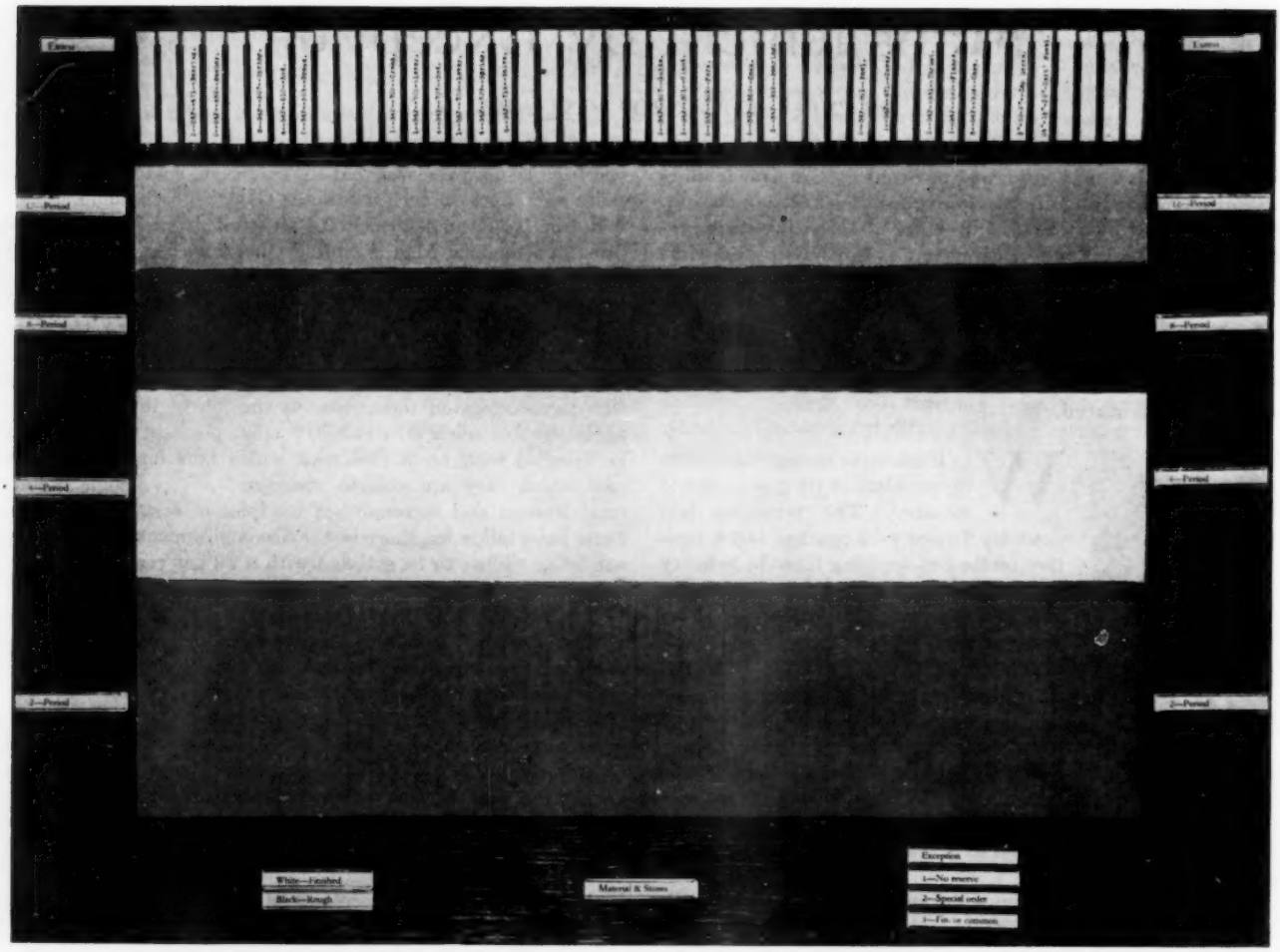


FIG. 1 PIN CHART ARRANGED TO DEMONSTRATE HOW ITEMS ARE PLANNED IN UNIFORM TERMS OF A COMMON DENOMINATOR
(The static chart without indicators is the permanent plan of the items shown across the top. The faint titles are planned exceptions, typed in red.)

management, it is no longer strange that the former has not been extended to every field of industry, as Taylor advocated. Systematic management assumes theoretical conditions and seeks to control exceptions by making those conditions paramount. Exception management, on the other hand, makes system subordinate to an exception, and must therefore visualize actual conditions in order to function.

The very first rule of exception management is that *"Consistency of procedure must never interfere with an obvious exception to that procedure or to its system."* Because it cannot be theorized or systematized, it has been utilized, as stated, only by the higher executives, and by them only as far as they can depend upon their immediate subordinates for accurate information.

Alford, in his "Laws of Manufacturing Management," reflects the former limitations upon its use by the way he expresses the law. The law of exceptions, as given in his book, reads:

"Managerial efficiency is greatly increased by concentrating managerial attention solely upon those executive matters that are variations from routine, plan, or standard."

It is obvious that, if the use of the exception principle is confined to the manager alone, and cannot be extended to include his subordinates and their subordinates in turn, it becomes merely a rule for managerial efficiency. Such is not the intent or limitation of the law of exceptions.

Because of the vagueness surrounding the history of the exception principle, the lack of rules for its application, and the apparent limitation of its use, it is natural to question whether it can add to systematic management anything that it does not at present possess. Briefly, the answer is that it gives, in fact compels, an entirely different management—one of exceptions instead of details.

In practice, the use of the exception principle separates manufacturing management into two distinct types, and these again into components having a common denominator. It divides the elements of manufacturing and management into constants and variants, and eliminates the constants from further consideration. It separates the variants into planned and accidental exceptions and ignores these also unless, and until, they vary from plan.

It plans permanently in terms of denominators that make all elements comparable regardless of their quantity or nature. It divides material, makes one portion a constant, and automatically balances inventory with much less investment than is usually necessary. In manufacturing, it eliminates schedules, progress charts, and calendar dates, and makes the use of work and material orders optional. It automatically separates the duties, directs the actions, and limits the responsibilities of different ranks of individuals in proportion to their capacity and authority. Further, it furnishes a basis of comparison for every item and provides a definite measure of individual and collective management by visualizing together the planned condition

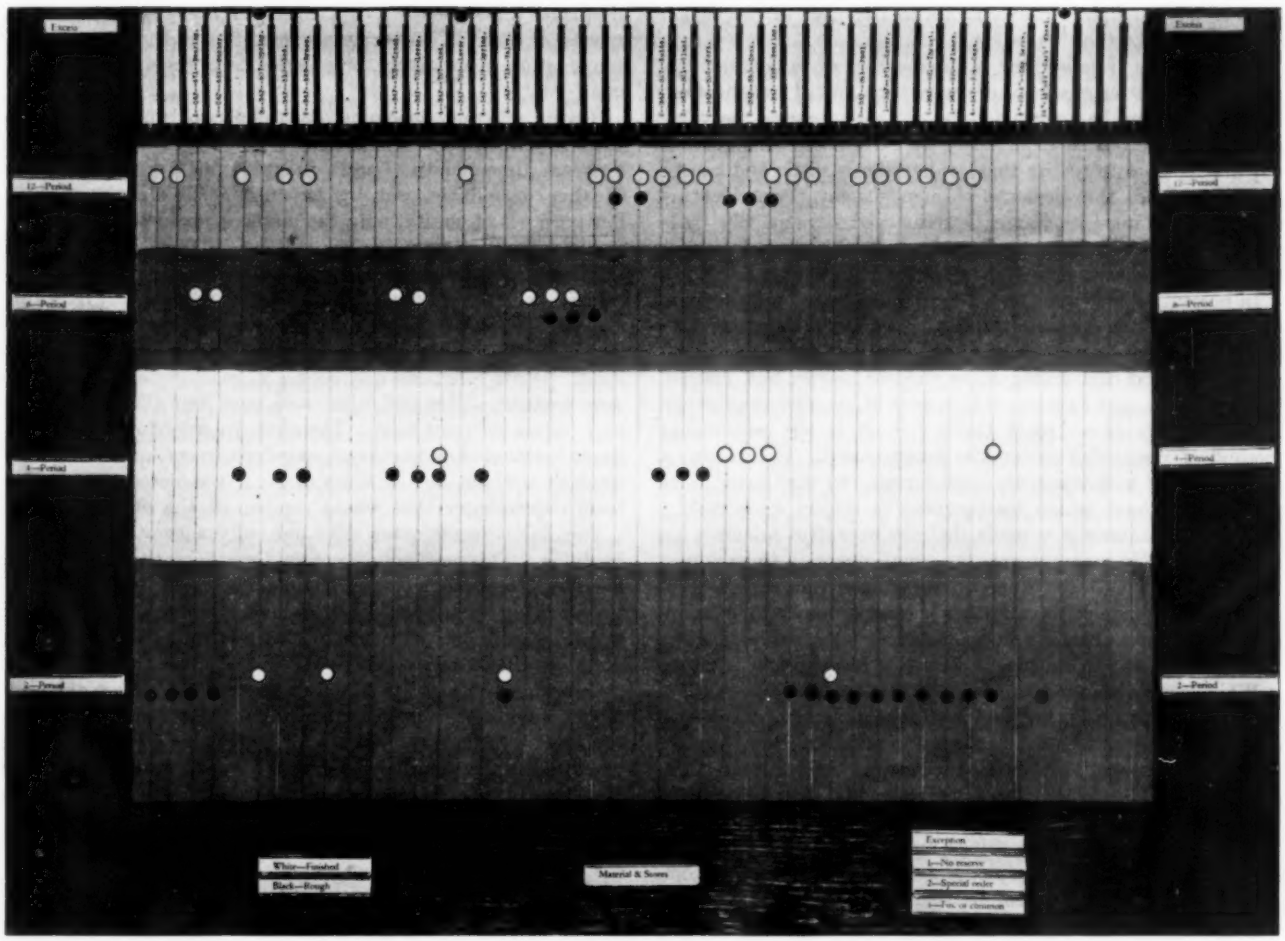


FIG. 2 THE BLACK AND WHITE PINS CORRESPOND TO THE INDICATORS USUALLY USED
(They contrast the actual condition of an item with its planned condition. The first two items from the left show planned exceptions which require no action. The next two show emergency conditions of roughly but eight weeks' supply of finished material.)

and the actual condition, emphasizing any difference between the two in definite terms. It does this by means of mechanisms into which skill can be permanently transferred, by a method of visualization that covers the entire range of possible condition, and by a forced coordination impossible under other circumstances. Summed up, it furnishes a self-corrective, compulsory method of management that functionalizes the processes of management by automatically apportioning the duties to those best able to perform them.

No longer is it necessary for the management engineer to be satisfied with a smaller percentage of accomplishment than a mechanical or accounting engineer can reasonably expect. Exception management simply cannot operate without controlling the function to which it is applied 100 per cent.

The natural reaction to such strong claims is to question whether exception management has any weak spots, and if so, what they are. Briefly, the exception principle is more dangerous in the hands of careless or mediocre executives than is system. There is such a thing as a weak system, but no such thing as weak control—it is either control or it is not. Exception management is an example of the permanent transfer of skill into definite mechanisms, similar to that transferred into an automatic machine. It follows the "law of the transfer of skill" in that "the attention and skill required to use a tool or operate a machine is inversely as the skill transferred into its mechanism."

Here also, as in the case of the machine, the "creator" must have had the skill to transfer. While it does, as Gilbreth said, "make good executives from mediocre ones," it does so because the skill of the planner has apportioned the mediocre executive a suitable load and a definite path along which to carry it.

Another drawback to exception management is that it cannot be applied theoretically. Systematic management concentrates upon influencing the mass of elements to follow a given path. It cannot visualize the stragglers. Exception management, after pointing the direction, ignores the mass and concentrates upon any items which show even a tendency to stray.

Experience shows that probably 70 per cent of manufacturing and management items will follow a planned path if not molested. Control consists in anticipating which 30 per cent will stray and when.

Reduced to its final terms, the art of management is to plan each element that enters into or affects its application. The problem of management is to forecast which of the elements will vary from its plan, when it will vary, and what will be the degree of its variation. The task of management, therefore, is to minimize and prevent variations from the planned procedure.

To do this effectively, three things are essential: First, a basis of comparison for each item; second, a common denominator that makes related elements comparable; and third, a method of visualization, in terms of the common denominator, that will compare and evaluate the different elements in such a

manner as to emphasize exceptions to the plan in proportion to the degree of their variation.

Gantt clearly visualized the requirements of exception management in the matter of a basis of comparison and of visualization, but so far as can be ascertained, he did not recognize the value of planning permanently in terms of a common denominator, nor did he seem to realize that, while plan is static and can be thus visualized, the elements of management themselves are dynamic and must therefore be visualized dynamically by a method capable of advancing or receding with the advance or recession of the element.

In presenting his chart, Gantt again emphasized the supposed opposition of the higher executives to scientific management and offered his chart primarily as a measure of management. Had he realized that management was not hostile, but, instead, was blind, he might have carried his work of visualization further. It is generally agreed that Gantt's chart is the outstanding method of visualizing systematic management. It furnishes a yardstick for measuring accomplishment, as contrasted with the plan, without which management is largely guesswork.

When the statement is made that the exception principle has not been used in management, the usual exception to rule must be allowed. H. Carter Harrison recognized its "supreme importance" in disclosing "variations between actual and standard costs," which, as elsewhere in management, means the difference between plan and execution, and applied it to cost accounting.

While accounting may be called a reflection, or result, of management, it is of prime importance as a means of visualizing conditions in systematic management. It will therefore be instructive to learn how the application of the exception principle to accounting was made possible, and why its general extension did not follow.

As has been said, three things are essential to the application of the exception principle: a basis of comparison, a common denominator, and emphasized visualization. Of these three essentials, accounting already had two—a common denominator and a means of emphasized visualization. The common denominator of accounting is value. Every item entering into accounting may be expressed in terms of the dollar or its fractions, and such items are comparable and intelligible regardless of whether they represent labor, material, overhead, or waste, because value is the product of time and quantity.

The results of accounting may be selectively visualized by grouping items into totals, by placing them in different positions, and by using different colored inks. Harrison found, however, that accounting lacked a basis of comparison and that it was not wholly effective as a means of visualization. He proceeded therefore, to set up "cost variation formulas" to compare and measure the difference between an estimated cost and the actual cost of the same item, or, as before stated, between plan and fulfilment. This made it possible to restrict the items demanding attention to those that were excessive or otherwise exceptional.

Like causes must produce like results, and if a common denominator, a basis of comparison, and a means of selectively visualizing them are applied to any other branch of management, the results obtained must be as definite as those secured in accounting relatively speaking.

As in the case of accounting, production management already had two of the three essentials. It had a basis of comparison in its planning, and it had its denominators in time and in quantity, but it lacked a means of dynamic visualization and could not therefore use its denominators.

Given these three essentials, the first thing their use does is to demonstrate that there are two distinct types of production management: one, where time is conserved at the expense of quantity and another, where quantity is conserved at the expense of time. These types are repetitive and contract type of manufacture, respectively, or as some term them, large lot and small lot.

Having different denominators, these two types of management require different treatment. There is a wide difference between time-quantity and quantity-time. Either type is equally controllable, but by an entirely different procedure. For purposes of clarity, only the methods necessary to the control of repetitive manufacture will be outlined in this paper.

The use of the common denominator of repetitive manufacture is expressed thus: *"Since time is the basis of repetitive manufacture, all elements must be presented in their relation to time."* This denominator applies only to production management. Other functions and phases of management have other denominators, and in still other cases, each item that is visualized may have a different base. Therefore, in order to visualize any single function of management, any department, or any management as a whole, on the same chart, it was necessary to use a basic denominator that would resolve all the others.

The basic denominator that naturally suggested itself was, *condition*. This not only resolves time, quantity, and value but any other denominator, factor, or element, tangible or intangible, in management or elsewhere. A moment's reflection will show why this is so, as well as how appropriate and universal condition really is.

Our whole outlook on life is in terms of the effect that different actions or lack of action will have on conditions that concern us financially, socially, or personally. When it rains, we visualize the rain not in terms of water, but in terms of the condition the rain will bring about. It is a good rain or a bad one, according as it affects conditions in which we are interested. When we refuse to eat a hearty meal just before retiring, we do not consult our desire: we reason in terms of the condition that will be brought about if we do eat it.

In manufacturing management while we plan in terms of time or quantity, we are really concerned with the conditions that will be brought about by a lack of time or quantity, or a surplus of either one. In short, from the time that we are able to reason, we do so in terms of condition. Condition, therefore, as a basic denominator, is not an innovation, but is the application of a universal basis of comparison and measurement. Everything is in some state of condition. To be able to visualize present conditions is to be able to forecast probable conditions, and thus forward or prevent them.

Condition being all inclusive, it was necessary, in order to use it as a definite measure of comparison, to have several different degrees of condition. Four were segregated and relatively evaluated. One extreme of any condition is normal; the other extreme is always called vital. Between these two extremes, two more or less arbitrary zones have been designated as attention and emergency condition, respectively. These four degrees of condition—normal, attention, emergency, and vital—are the standards of evaluation and measurement used for visualizing purposes.

The third essential of management, emphasized visualization, requires an unusual type of chart. It may be made in any desired form, provided it fulfils the requirements. The one used is of wire screen of any desired length and width. The indicators are permanently attached.

The chart that must be used has as its base the four degrees of condition mentioned above, arranged as horizontal zones. Each zone has a distinctive color, and its relative importance is emphasized both by the intensity of its coloration and by its width. The effect of this arrangement is to attract attention first to the vital zone and next to the emergency zone.

The colors used are green, blue, yellow, and red, for the normal, attention, emergency, and vital zones, respectively. The two upper zones are the operating zones. They are inconspicuous both in color and width, because they represent normal and near normal conditions. The two lower are the exception zones and are therefore emphasized.

These four zones not only indicate different degrees of condition, but also the relative rank of the individual who functions within each of them. The operating zones fall naturally into the classification of clerical and supervisory. A normal item is purely clerical. If for any reason it ceases to be normal and requires attention, a supervisor is the logical one to give it attention at that stage of variation.

The original planning does not contemplate that any item will go into the two lower zones; thus only exceptions to plan are visualized there. Such exceptions receive, first, the attention of a general supervisor or of an assistant to the executive that is responsible for the application. If this fails to arrest the variation or to restore the item to normal, it receives the attention of the executive himself, at a time and in a condition provided for in the original plan.

The four zones, therefore, when used for manufacturing management, are called normal-clerical, attention-supervisory, emergency-staff, and vital-executive. These are the basic characteristics of the chart. They are always used and understood regardless of the application and of whether they are indicated by legend or not.

Arranged thus, the chart serves a threefold purpose; first, the bare chart, with appropriate legends, acts as a common denominator and visualizes the planning. The top of the normal and the bottom of the vital zone are the extremes of the allowances of the plan. The two central zones represent stages in the plan or increments of time, quantity, or progress between the two extremes that are proportioned to the authority and capacity of those responsible for them. Legends denoting what denominators are being used and what proportion of the whole plan is assigned to each zone are placed in their proper holders. The titles of 100 per cent of the items belonging to a particular department or function are placed across the top, and the whole represents a complete plan with three different degrees of exception, each apportioned to a different degree of authority.

The third rule of management by exception states that: "*Positive control demands the visualization of 100 per cent of all factors, 100 per cent of the time.*" Such visualization is compulsory and necessary, because an exception cannot be detected until it happens, but if it is caught at the beginning of its variation it can usually be either controlled or discounted. The tendency to stray is shown by the chart, which therefore directs what action shall be taken and how long it shall continue.

The selective zone principle apportions the responsibility accurately, not by dividing the items numerically, but by dividing each item and observing the laws of specialization, division of effort, and responsibility. The net result is that the law of exceptions is amplified so that it may be read: "Efficiency in management is greatly increased by confining the attention of each individual solely to those matters within his own jurisdiction which vary from routine, plan, or standard."

The second function of the chart is to visualize the actual condition of the two elements which constitute each posted item, and to contrast them with their planned condition. Actual condition being dynamic, the means of visualizing it must also be dynamic. For this purpose, two separate but related elements of each item are indicated—one by a white indicator, the other by a black one. These related elements may be rough and finished, current and cumulative, time and progress, or any two dependent factors.

The reason for indicating two factors is the simple rule that, "from two known terms, the third, or unknown, term may be found." It is of no value to an executive to look at a chart and see that sales are above the average for the month, unless he knows what they have been for the year to date and also what they were planned to be at different intervals. Neither does it profit a production manager to be informed that a certain rough stock is exhausted unless he knows also how many finished parts are on hand and how long they will last. The "two known terms" principle is automatic and universal in exception management. Obviously it must be, if only exceptions are to be recognized.

This double indication does not require additional work or space, because they have but one title. The two indicators may have the same denominator or different ones, depending upon the application. In fabrication, for instance, they represent tools and material, respectively. In contract or similar work, one represents location or progress and the other the percentage of scheduled time that has been consumed, while on a coordinating application, where every item has either a different denominator or none at all, one indicates specific condition and the other cumulative or average condition.

A hospital is about as different from manufacturing as can be imagined, but the chart visualizes each with equal clarity. One indicator shows the temperature of the patient, 98.8 being normal and above 104 being vital; the other indicator shows whether his degree of condition is normal, below normal, emergency, or vital. Here, as elsewhere, the two known terms are necessary to arrive at an intelligent conclusion. Here, also, the zones divide themselves logically into tasks apportioned to the shoulders that must bear them. The different zones become normal-nurse, attentive-interne, emergency-house physician, and vital-consultation. In this case, the chart would be cleared twice a day to check the attention given each patient.

In applications where a deficit of time or quantity will eventually become vital, the maximum allowance or requirement is shown in the top or normal zone, and the indication is in terms of receding quantities. The chart has, when used thus, the inherent characteristic of automatically increasing the importance of an item in proportion to its decrease in time or quantity. In other cases, such as contract sales, the indicator for unsold capacity starts up from the vital zone, while the "time consumed" indicator moves down from the normal zone. The same is true of starting a contract or assembling an army. In other words, vital indications, whether receding or increasing with time, are in the vital zone at the proper time.

One of the results of indicating in terms of receding quantities is that exceptions stand out clearly. Possibly 2 per cent of the indicators get as far as the emergency zone, and less than one-half of 1 per cent reach the vital-executive zone. They are, therefore, quite conspicuous.

The third function of the chart is to act as a measure of management. The actual condition of the two related elements of an item, as shown by the two indicators, contrasted with the original plan, which is the chart itself, gives the efficiency of execution. Not only as a whole, but in four different degrees that represent the relative and actual efficiency of the individuals responsible for the different zones. This is the feature that Gantt considered so desirable. Any method of visualization that uses a basis of comparison is a measure of that part of management to which it is applied, but the visualization of 100 per cent of the items, a common denominator, and definite zones of action make measurement both comparative and definite.

The three functions of the chart—the permanent plan, the actual condition, and the efficiency of execution—constitute

a simple method of visualizing the requirements and results of management that is as plain to the messenger boy as it is to the manager.

Visualization of conditions, simple as it is, makes the use of the exception principle possible. The chart and the permanent methods it makes possible are the mechanisms into which skill is transferred to such an extent that management becomes directive and, when necessary, compulsory.

The foregoing epitomizes the difference between the two methods of management. Exception management takes any plan and reduces it to a common basis on a chart. The dynamic indicators show whether the plan is consistent, and, if not, which items need correction. Thus, over a cycle of time, the method is self-corrective, because all items having been planned to a common basis with all differentials allowed, they should normally follow a uniform procedure. Any item that lags behind or that proceeds toward the vital zone too rapidly receives the individual attention of clerk, supervisor, or staff, and each must explain to his superior the reason for the exception to plan unless he rectifies it himself. It is impossible to have a surplus or deficit of any item at any time without its being conspicuous as an exception. It is also impossible to neglect an item unless three people neglect it in succession. This and the fact that those responsible for the two exception zones are also notified by form when items are indicated in their zone constitute the compulsory feature of the method and insure action being taken to prevent exceptions as well as to rectify them.

Another decidedly important point remains to be emphasized. No method of visualization can be effective if it is used only for tickler and informative purposes, and no system of management can be effective unless it has within itself the elements of compulsion. Neither can it be automatic or definite without being compulsory, because the three terms are synonymous. No man, regardless of his station, does anything without an incentive. The only difference is that, with some men, the incentive is in front and with others it is behind.

The creative instinct of the executive type rebels at repetition. Only the detail man is content, or even able, to follow a routine day after day, and he is not considered executive material. As pointed out, this was part of Gantt's trouble. Executives, when given a task, would attack it with all the enthusiasm that they possessed as long as they could see a reason for so doing, but they failed five times out of ten in the attempt to follow a routine that was unnecessary.

The last of the four rules governing exception management covers this point effectively. It reads: "*A department shall not post any chart that directs its activities or that visualizes the results it achieves.*" This rule, like the others, is so simple that it requires no effort to observe it.

The chart applications, then, provide direction without comment and record failure without criticism. They also provide a pitiless publicity, by an independent agency, which is both an incentive and a compulsion. There are no rules as to what must not be done or what shall be done, except the four referred to. The compulsion lies in the inevitable progress of the indicators from zone to zone when actual condition varies from plan and in the notification of the staff or the responsible executive when an exception occurs.

Thus it may be truly said that exception management makes it possible for a few girls to direct the activities of a large plant. The uninformed accept this statement with wonder or skepticism, but a knowledge of condition shows the assertion to be without point because the skill of management has been transferred into a permanent mechanism, and the attention and skill required to operate the mechanism are inversely as the skill transferred into it.

At this point, three questions naturally arise. They are: How much does it cost in time and labor to post the charts? How can a large number of items be visualized? How much of the executive's time is required?

The answer to the first is that in the shop the departmental charts are the only records necessary. Because schedules, definite quantities, and dates are ignored. The production department, therefore, has practically no records to keep. It posts the departmental charts for dispatching and visualizing purposes.

The material charts usually require part time of one girl, who posts them, together with the storeroom records. The items showing on the chart are not posted every time the stock records are changed. As stated, definite quantity is ignored in planning and visualizing, and the indicators are changed only when the balance between the planned condition and the actual condition is disturbed beyond the planned limit. The visualization of material is in terms of periods of time, with an ultimate reference, of course, to quantity in the form of economical manufacturing lots.

The second question was asked by a manufacturer who had 110,000 items of material. Another asked how it was possible to visualize material with a turnover once a week. The answer in both cases was the same. Exception management has no system. It simply visualizes any system that can be consistently planned. Therefore, if a system is used, the chart will visualize it in terms of a common denominator, balance it, and forecast exceptions to it. If a plan is not followed, its adoption will make planning compulsory, and whether the material is kept track of in terms of individual parts, groups, or sub-assemblies, it will visualize the difference between actual and planned condition 100 per cent of the time.

It is not concerned about whether the turnover period is one day or one year. It merely makes the method used positive by forecasting shortages and excesses in time for them to be remedied. Preparation for a weekly turnover must extend further back than one week.

This leads up to the question of room necessary for visualization. One cannot expect to receive something for nothing, and while the method of visualization used in exception management requires less linear space than any other, it still requires room. The only application that requires any considerable space is for material, which is 20 linear feet for 1000 items. The charts, in any desired length, are usually placed against the wall or back to back in the aisles, with the bottom of the chart high enough not to obstruct the vision of persons seated. The reason for the small linear space required is that the two elements of an item are visualized under a single title instead of separately.

The third question is: With 100 per cent of the items visualized and any or all of them likely to go into the executive zone, will not the method require too much executive attention?

There are three reasons why items go into the executive zone. They are: Error in planning, failure of subordinates to rectify or prevent, and an uncontrollable exception. The first two are the responsibility of the executive, and they can be remedied by him.

Experience shows that, with material turnover periods of four, eight, and twelve weeks, less than five uncontrollable items out of a possible 1000 reach the executive zone after the material has made one cycle of turnover.

It still holds true, however, that there can be no management without a manager, and no transfer of skill without ability; but, granted these, there is no comparison between the systematic and the exception methods in economy of labor, overhead, and material.

Discussion

WM. H. GESELL.[2] There can hardly be any doubt that the modern factory that is run along lines of approved scientific-management methods does not in some degree make use of the principle of management by exception. The principle which advocates that executives and sub-executives should not be burdened with a mass of routine details connected with the execution of any function which is smoothly proceeding according to plans seems to be the quintessence of common sense.

A practical chart which would enable the executive to visualize variations from planned routine would naturally be of invaluable assistance. However, the question naturally comes to mind, How much does it cost to prepare these charts and to keep them in such a shape that they reflect the current conditions in the plant? The author contends that the use of these "visualization charts" will practically do away with records normally kept in the production department. It seems to the writer that the condition emphasized in the visualization chart can be determined only by knowing all the factors which bring about that condition. Using the author's illustration, one must first record the temperature of the sick man, and by comparison with the normal temperature then determine the condition of the patient. Similarly, one must record all the facts pertaining to the execution of a given manufacturing function. If that execution of function proceeds according to plan, there is no need to burden an executive with them. Yet they must be recorded for possible analysis and comparison should a function not proceed according to plan. In other words, records must be kept. Possibly the writer has missed the point of the author's contention because he never has used one of the proposed visualization charts.

Another thought in connection with the charts is that while they may be suitable for large businesses where thousands of employees are concerned, yet in the small factory of only several hundred employees it seems rather unnecessary to employ such a method. The introduction of the principle of management by exception into an organization is largely a matter of training the employees right down the line.

KENNETH MCGRATH.[3] The author has stated the benefits of exception management with such precision and logic that those who doubted its advantages compared with the conventional or systematic type of management are at least willing to try it.

For the last few months the writer has been considering how he could better portray certain angles of a production-control system through the use of charts.

Considering one form against the other, in the case of exception management through the aid of charts there is something which is an open book at all times and which every one from the highest to the lowest can refer to and pick off at a glance the items for which they are responsible. In the other type of management no one except the clerk actually responsible or working on the particular file in question really knows what the high spots are. Of course, he probably has been instructed to send a note to his superior when the danger point of any item in his file has been reached, but this does not have the same effect on the executive as the visual examination of charts at stated intervals.

In the early part of the paper the author spoke of the opposition that both Taylor and Gantt experienced in their installations. This is something with which one will always have to contend. Every one feels that he can give advice on management matters. Probably this is because it is not an exact science, and furthermore because of the experience all have had in managing their own affairs. Ask a man to drive a car or figure a problem on a slide rule, and if he has had no experience he will say so, but not so in matters of management.

We should not be disappointed on this account, because it is only natural when one considers that systematic management has existed as long as industry, and for that reason companies that have grown from a small beginning where their entire organization is acquainted with their existing system are naturally hesitant in what appears to them a radical, doubtful change.

The writer would appreciate it if the author would say where additional information on the make-up of the charts referred to could be had. In fact, it appears to him that the actual physical make-up of charts would be a worth-while topic for some future meeting.

GEORGE A. PAGE.[4] Historically the system presented by the author seems to be a routine successor to an executive function or individual common in factories, jobbing shops, and on construction jobs for more than 20 years in the writer's own experience and still in use in flexibly organized work. Indeed many of us have been "it."

The unconventional users of the fundamental method of management by exception were frequently men who were opposed or indifferent to Taylorism, and in one instance of the writer's own experience such an unconventional advocate of the principles of Taylor and Gantt retained the latter to promote this work only to have the alliance rudely broken because they "couldn't make sense of Gantt" and because Gantt on the other hand found them to be impervious to new ideas. Yet they were working in closer harmony than either realized.

Such "outsiders" would probably be classified in this meeting as using Taylor ideas and Gantt methods in a general way. But from the paper it appears that some would charge that to apply the principles set forth under his title through a human agency instead of as a statistical or signal system would be archaic and without method. Here the "radical" and the "mossback" began to concentrate on their differences and ignore their common views. One was charged with being "method minded," as would be said today, and the other with "standing pat" on results through inertia, prejudice, and ignorance.

But the "archaic" or "man power" application of all management principles had their particular virtues, and in the case of the "exception principle" such virtues are illustrated by four natural characteristics that the human agency possessed that were not inherent in any formal routine system. These four items are:

Versatility
Judgment
Adaptability to small jobs and concerns
Yielded more information.

For the report included in condition, for example, not merely position, as with a military map, but direction, rate of change, predicted position in advance interval of time, reported application of corrective measures, and current result.

Since this managerial detail has been known by many names, and its personnel by many more—most of them picturesque and none of them descriptive—the writer will merely recall that the personnel or individual charged with this work was usually a man thoroughly informed as to requirements and personally experienced in all the details involved.

Is it surprising that a conservative executive, competent but

[2] Vice-President, Lehn & Fink, Inc., Bloomfield, N. J. Assoc-Mem. A.S.M.E.

[3] Eclipse Aviation Corporation, East Orange, N. J.

[4] Consulting Engineer, Freehold, N. J. Assoc-Mem. A.S.M.E.

busy—or an executive board, the larger number of whom were less familiar with the internal operation of their business than with its external aspects and contacts—should look askance at a mere system without brains as a substitute for a man or a department that was able to help them with their thinking?

The writer submits that able scientific management experts frequently enter a concern when they are unhampered and provide systematic management while they remain with the outfit, and leave behind them when they withdraw their presence and services—merely management system. This seems to be an unhappy chapter in the history of rational management. Where the established management has not hospitably extended unlimited cooperation with the incoming change or to its ambassador, the story has been still more unfortunate.

Is it not true that much new light might be shed on ideal management by a thoughtful consideration of the examples where the application of modern principles of management has been hastily conceded as failure, with special regard for the institutions to be displaced and a sympathetic analysis of the human situations? Is this not, after all, the utility of detailed history of a subject like management?

DAVID B. PORTER.[5] It would be difficult to trace the origin of management by exception. It is used by every one of us in the control of many of our every day affairs. If we are in the habit of being home for dinner at a certain time, we do not telephone from the office every day that we will arrive at the usual hour; but only in case something has arisen to prevent the usual event from occurring do we find it necessary to notify our domestic headquarters. Of course some of us may be so irregular that there is no habit of being home for dinner, and consequently no established routine; therefore a non-appearance at this meal is not an exception, and consequently no notification is necessary.

It is perfectly natural, therefore, that this principle should be extended to the more complex problem of management in industrial and commercial enterprises. Taylor, Gantt, and Gilbreth recognized its importance and devised means for its application. It was the writer's privilege to be associated with Mr. Gantt at the time when he was developing his system of charts which give visual expression to the exception principle. He was able to do this by reducing the various factors to the common denominator of time which is the field of the chart, and by using bars to show their condition with reference to time. The normal condition is represented by bars which are up to or moderately ahead of the current date. Exceptions to the normal state are represented by bars which lag behind or are excessively ahead of the current date, and the degree of variation from normal is measured by the amount of this lag, or lead.

The author, in this paper and in earlier writings, has presented his tell-tale mechanism for visualizing exceptions from plan which appears very effective in its application of the exception principle. He accomplishes this by reducing all of the factors to the common denominator of condition which is the field of the board, and by using indicators to show their degree of condition. This technique compels the use of several fundamental principles of good management.

First, it depends upon 100 per cent planning of all the factors. This includes the determination of the allowed limits of variation from plan for each of the four degrees of condition.

Second, it transfers intelligence into the fixed plan in the same way that skill may be transferred into a machine tool.

Third, it transfers judgment into a routine by notifying the proper individual designated for taking care of each degree of variation from plan.

Fourth, this latter provision necessarily apportions and delegates responsibility in accordance with the ability to assume it.

Fifth, it conserves the time of all executives by bringing to their attention only those matters which vary from plan and over which they have jurisdiction. This liberates the higher executives from the useless attention to routine detail which ought not to concern them. In this connection it tends to develop executive material by training those in subordinate positions in the art of executive technique.

The writer would suggest that the use of too many denominators on one chart be avoided. The effectiveness of any method of control lies in its degree of simplicity. This was emphasized by Gantt, who hoped to establish through the medium of his chart a simple and universal language which could readily be understood in any of its applications.

JAMES C. TUCKER.[6] The subject seems too exhaustively treated to offer additions or criticisms. Management for the past thirty years or more has been very much talked about—systems and methods experimented with and variously tried out. Are we getting anywhere? Doubtless we are, for large industrial organizations must be better managed than formerly, else utter confusion would result, unbearable expenses would be involved all ending in sad failures with disrupted society and untold suffering among the workers thus deprived of their livelihood.

Better management, then, has become as essential as better machinery, better materials, and higher grade output.

The author has reviewed the exponents of various management systems scientifically evolved during the past thirty years, and now proposes renewed attention to "management by exception," as the inevitable and the most workable next step toward more efficient results. He notes that both Taylor and Gantt could not eliminate the law of exception from their advocated systems, and then develops a plan whereby the "law of exceptions" may be tied into a system so that management may be graded and assigned to executives of progressive rank.

Ten years from now will one wonder, as did Taylor and Gantt, why the idea meets opposition or indifference among leading managers, and will later students turn the pointer of better direction back to system or to some new plan yet undiscovered or to be evolved?

The writer ventures to say that some day it will be discovered that there is a fundamental or perhaps a divine principle so basic that upon it may be evolved a system of management as complete, unchanging, and unexceptional as the law of gravity and the law of centrifugal force, which keep our universe in consistent harmony.

The weekly system of labor and rest which, according to Scripture, was set up by the Supreme Being when He did all His creating in six days and rested on the seventh day, has been pretty much adhered to ever since. Only lately have some of our prosperous fellow workers discovered that in five days can they earn what they need to spend in their two-day week-end. Mr. Alford, in his reference to early instances of applied management principles as recorded in his new book, "The Laws of Management," apparently overlooked that remarkable example of "management by exception" set up by Moses under his father-in-law's direction, as recorded in the 18th chapter of Exodus.

One writer concludes that, after all, there is not much new to be known about the principles of management.

The author infers that we may have inherited, and therefore

[5] Assistant Professor of Industrial Engineering, New York University, New York, N. Y. Mem. A.S.M.E.

[6] Plant Superintendent, Plant No. 2, Carrier Engineering Corporation, Newark, N. J. Mem. A.S.M.E.

come naturally into the acceptance of management by exception through its being the basis of family government.

Another writer, after discussing various systems of wage incentives, concludes that, after all, the gross results are not much different than those obtained by the ordinary day-work-pay method.

Are we talking in circles, getting nowhere, but perhaps enlarging our horizon of vision, or is it the inevitable triangle in which we are brought up sharply at each angle to try a new direction of procedure, profiting by our experiences on the last leg, returning again to discover that the human element is eternally involved, and that our progress in management procedure barely keeps pace with the changing demands of business and industry?

The author, after presenting his plan with charts, whereby exception management may be operated infallibly by clerks, supervisors, staff executives, and managers, has cautioned that no plan can function successfully without capable managers.

Capable leadership is essential in successful management. Then we must have leadership. Ability and intelligence, plus training, will make leaders: then we must have training.

A paper entitled, "Training Minor Executives in a Rapidly Growing Organization," presented before this Society in May, 1928, by Mr. A. J. Beatty, Director of Training, The American Rolling Mill Company, and the discussions led by R. C. Forbes, C. S. Coler, J. O. Keller, and others, have ably outlined plans and methods for training leaders.

Craig and Charters, Tead and Metcalf, and many other writers on management have supplied us with ample source material to develop our potential leaders into trained managers.

Then, it seems, any good, workable plan of management, whether it be Taylor's, Gantt's, Emerson's, Ford's, or any other, will be adapted by trained managers to the particular type of business to be managed, with a reasonable, if not a high degree of success.

The author states that "exception management has no system." He also emphasizes that it is an effective aid to a system. That, it seems, is the point to keep clearly in mind. One also must be sure that any mechanism adopted by which the managerial line will function effectively must not be so complicated, expensive, or cumbersome as to absorb too great a percentage of management effort for its operation.

In spite of the author's assertion that the "skepticism" of the "uninitiated" is "without point," it is found that some are still skeptical as to the possibility "of a few girls to direct the activities of a large plant, by exceptional management." One may feel as did a certain gentleman who said of another he had eloquently presented an idea he wished to "sell," that "his argument is persuasive but not altogether convincing."

With the aid of "the professional manager" and with a better understanding of the "exception principle" reduced to terms outlined in the paper, amplified in future discussions, papers, and probably books on the subject, one should be prepared to meet every intricacy of management that modern business and industry will demand.

GEORGE G. BERGER.[7] The author is undoubtedly prepared for a difference of opinion in the discussion of so highly a controversial subject as management. His paper would have been clearer if it included a case example to illustrate just what he means by the term "element," what and how many elements are to be classified as variants, and how the zones for compelling various types of executive attention are set. These are points which require illustration by example in order to determine the practicability of his technique of management and determine the "standards" of discussion.

As the writer understands this technique of management by exception, which is designed to relieve the executive and subexecutive of numerous details and functions of management and to call their attention to those situations which require higher degrees of judicious action, the various elements or factors affecting the industrial organization, its operations, earnings, and growth are classified as constants and variants. It is taken for granted that the constants will be adequately controlled by standardized management. The variants are subdivided in turn into accidental and planned exceptions, and classifications or zones are established for these exceptions. These zones are intended to call for various kinds of clerical and executive action. If, by elements, the author means all the factors, functions, details, and phases affecting industrial operations, the making of an inventory of variants and constants will prove quite a task. Many exceptions cannot be forecast and are only known when they first confront the management personnel. Then to establish the limiting values for these voluminous elements in order that they fall into classifications calling for various types of executive action gives the work of organization for action a fearful aspect.

The author presents the idea that Taylor and Gantt were arbitrarily insistent upon the "use of a method of management that had no history."

The work of Taylor was to establish standards of the tool and of the worker in order to control the utilization of factors that had been previously neglected. This work following the period, which may be called standards of the product, is part of the evolution of management and the standardization trend for control. The evolution extending back some thousands of years is ample history to back up a method which is part of the evolution.

In the efforts to develop the science of management, there have been used the deductive process in endeavoring to hit upon general principles and apply them to the specific case. In the author's words, there is "an instinctive knowledge that there must be some natural method of management that directs when direction is sufficient and that compels when compulsion is called for, one that does not interfere with matters which are progressing in orderly compliance with a planned procedure, but which has positive methods of detecting and correcting variations from that procedure." This instinctive, deductive process is an example of the mental attitude of the medieval ages, when the search was on for the magic formulas and the inductive process for developing science was neglected. Bacon's masterly "Organum" was designed to curb this attitude, and John Locke in his "Essay on Human Understanding" warned that future investigators of science at all times would tend to this method.

There have been financial ratios, economic-lot formulas, and other general principles which completely conceal the fact that the science of management has for its object establishing the science for the specific organization which has its own set of standards (derived directly from its standards of the product, tool and worker, and which represent the particular set of facilities and attendant arrangements). Taylor assumed this same operating point of view when he decided to set standards of the worker by the time study, taking into account the conditions surrounding the specific situation, rather than set the standard by the average for that operation as shown by statistical data of output in his plant or other plants.

The object of management (considering the term in its broadest sense) is control. That is the object of any science—control over the phenomenon or objects it seeks to understand by cata-

[7] Consulting Industrial Engineer, New York, N. Y. Jun. A.S.M.E.

loged laws of cause and effect. To fix the laws of cause and effect in a science, there must be standards to fix the conditions surrounding both cause and effect. Management searching for control in all phases requires an increasing number of standards. Practice is now up to the standards of the management in the trend following Taylor's work; their development is logically compelled by the present status of industrial operations. On the basis of standards of the management, one may secure by some twenty-five or thirty operating policies the measurement of management for the industrial unit. In this rating there is incorporated the exception principle, with the "exception" that the industrial manager chooses what requires his attention, whether it is a constant or a variant, whether an accidental or a planned exception, and can determine in terms of the rating (a numeric of the standard) the importance of the variation of result from standard. This is in accordance with present practice in industry. It is also in accordance with the principle that the human factor is the most important in the future of the organization and that the measurement should be directly placed over the human factor. By taking the operating point of view, setting the standards for these policies on the basis of standards of the product, tool, and worker, one may have outstanding standards which not only serve as a basis for evaluating results, but as the basis of the practical measurement of management. This attainment is an object of scientific management.

Author's Closure

It will require at least two, and probably three, more papers to describe exception management and the method of visualizing it. Management is divided into four distinct branches, each of which has a different denominator, and therefore a different technique. The denominators are time for repetitive and quantity for contract type of manufacture, value for accounting. and condition for those factors of general management that cannot be resolved by the other denominators.

To describe the applications of these denominators and the different technique necessary in each case will require several more papers.

It is hardly to be expected, therefore, that the discussors can pass judgment upon the whole, from reading an introductory portion. On the other hand, it is not to be expected that the author can give a clear view of the whole in the same portion. The technique applied and the results achieved are so radically different that a general statement covering the whole of industry would be classed as exaggeration if not romance, and in the light of our present technique would not be understandable.

Most of the questions, other than those that are matters of personal opinion, will be answered in succeeding papers. There are some statements, however, that indicate that certain parts of the paper are vague. Mr. Tucker, for instance, states that "possibly some time a fundamental method will be discovered." It has!

The technique which is described in the paper constitutes management by condition. Condition is a universal state than which nothing can be more fundamental, except the factors that cause condition. Creation began with a nebulous condition; it will end with a chaotic condition, probably. Management must be based upon things as they affect our senses. Our imagination creates conditions; our intellect reasons conditions; our taste, smell, and touch define and measure condition. Our whole life, then, is a series of conditions which we visualize and emphasize according to our interests. In management, the tell-tale chart visualizes any condition that can be defined. The reaction to a condition, of course, varies in different individuals, and it is the nature of the reaction that indicates the degree of efficiency.

Mr. Gage thinks that an executive "will look askance at a mere system without brains as a substitute for a man or a department that was able to help him with their thinking." Mr. Gage has the wrong conception of the technique. The men and departments help the executive more, because the combined thinking results in a plan that is permanently and exactly recorded and statically visualized. The plan and technique are self-corrective over a cycle of time, because being planned in terms of a common denominator, any difference between the plan and the actual practice is shown conspicuously by the dynamic indicators. If the variation is slight, it is brought to the attention of a subordinate. If important, it is brought to the attention of an executive. Management has long sought a method of functionalizing duties. It has not fully succeeded, because importance increases with the passage of time. A detail that is a routine matter today, a month hence may be a vital factor. Functionalizing, therefore, must be in terms of condition, not items.

The medium of visualization that makes exception management possible is an automatic machine that takes the raw material of management and converts it into a finished uniform article by the law of transfer of skill, just the same as any other automatic machine. Alford, in the "Law of Transfer of Skill," quotes Roland as follows: "Intelligence may be transferred from the brain of the machine maker to the machine itself, to such an extent and with such permanence that the machine which the toolmaker has thus imbued with a portion of his own wits can subsequently, so long as it is in working order, supplement with this endowment the small mechanical skill of its tender." Any degree of skill in planning may be transferred into the mechanism of the chart, and the skill necessary to execute the plan "will be inversely as the skill transferred into its mechanism." What greater help could an executive have?

Finally, Mr. Beyer thinks that the paper would have been clearer if it had included a case application. It does. There is no base from which to theorize. Everything that has been said has been descriptive of the original installation in 1921. It is not necessarily the best technique, nor is it necessary to agree with it. The author is a practical shopman with 30 years' mechanical and management experience. As is usual in such cases, practice preceded theory. The things that the author has described were found out after the installation, not theorized before. Primarily the chart is based on that of Gantt, with the addition of a common denominator, dynamic indication, and two-factor visualization. It is a mechanic's idea of what Gantt would have eventually evolved from his line chart.

MAN-53-5

Training Apprentices and Foremen

BY FRANCES PERKINS,[1] ALBANY, N. Y.

The experience of the author in labor problems and in legislature affecting workers provides a background for recommendations as to the proper training of apprentices and the development of foremen. Foremen generally are ignorant of the labor law, the health requirements, or of safety. The personnel manager and the safety engineer of the plant should work in closer contact with the foremen. Under the Smith-Hughes Act, the University of the State of New York has a course for training trade, industrial, and technical teachers. The courses are offered in large centers. Should labor management be taught in addition to these other subjects, there would be provided a good course for foreman teaching.



THE far-seeing employer is building today not only his physical properties but his man power. At this moment, when we are in the slough of depression, none but the far-seeing does such things as these. Experience and insight tell him that the future holds expansion, as the present holds depression; he knows that now is the time to prepare for expansion. Industry is profoundly affected by the present surrounding gloom; for the manufacturer there is no tomorrow; there is nothing in the adage "In time of peace, prepare for war." Some think that now is no time to train men in industry, because unemployment gives us the pick of the market. But the time inevitably comes when the whole scene will change; factories will hum, sales will go up, and there will be the usual shortage of labor.

At such a time something must be done; high profits can be secured if only production can be obtained, but the employer perhaps feels that labor is not doing its share, either through reluctance or ignorance. The training of workmen is then considered hastily, but it is too late; the "system" only gets in full swing just as the pendulum begins to swing the wrong way. Many of our industrial problems follow the same course. We are now concerned with the condition of unemployment; ameliorative agents that should have been at work two years ago are now only blossoming just when the unemployed, by the mere passing of events, are getting back to work.

Let us consider industrial training from the broad point. It has not been too successful in the past. There is no integrated scheme which recognizes that most human beings are destined to work almost from the time school has been vacated till death takes them away.

Theoretically, every child who leaves school is educated; practically, it is not so. There is an effort to make the pupil mentally and physcially fit for his life's work, but the effort is defeated not only by the desire of parents to get their children on "the payroll," but the desire of some employers to get youth into industry at too early an age. How many employers there are who complain about the inefficiency of clerical help, for instance; yet they hire workers whose term of school life has been shortened by the call to the office.

It seems that the great limitation of industrial training, whether in apprenticeship or in shop, is due to the fact that it has been only haltingly, partially, and haphazardly applied by industrialists who are not teachers, to their employees from the production point of view. What would be thought of our teachers should they venture on any scheme to get out production; they would fail; they could only use a small part of their knowledge in the effort.

It is sometimes held that, if you stop production to train men, production is thereby irrevocably lost. But is it not apparent that if you should render a worker unproductive for a month while you train him, and by training increase his producing capacity by 10 per cent as long as he is in employment, you will have made a substantial gain? All this presupposes a certain stability of labor, but it is certain that if you train a man for your sole profit, turnover may be expected, and the whole scheme of training will be for the benefit of your rivals. If training is to have any significance at all, it should mean more profit for the employer and more wages for the worker than he can obtain elsewhere. Thereby you will have stabilized him by the most secure bond.

A Government document has been published recently; it gives the names of the trade associations of the country. Many famous and powerful organizations are listed among them. They are the leaders in their respective industries; they direct policy and are decisive factors in guiding their forces to success. I recollect few of them together, or any one alone, going to the great teaching institutions of the country and asking for guidance in the problem of training the workers under them, except in the most limited manner. What is more serious is that never has the problem of the transference of the youth from the school to the industry been planned out in anything like a comprehensive way. The result is, of course, chaos. The boy, released from

[1] Industrial Commissioner, Department of Labor, State of New York. Frances Perkins was born in Boston, Mass., educated in private and public schools in Massachusetts, graduated from Mount Holyoke College, studied as a post-graduate student at the University of Chicago, University of Pennsylvania, and Columbia University, specializing in sociology and economics. In 1910 she became executive secretary of the Consumers' League of New York City, conducting a successful campaign for the passage of the fifty-four hour bill, the law which limits the hours of labor of women to fifty-four hours a week or nine hours per day in the State of New York. She also urged successfully amendments to the labor law in regard to the protection of women and minors and regulations of tenement homework manufacture. In 1911 she conducted an investigation into cellar bakeries in the City of New York, in cooperation with the Commissioner of Accounts, which led to a thorough-going regulation of the sanitary conditions in cellar bakeries by the city authorities. In 1912 she became secretary of the Committee on Safety, an organization of private citizens formed after the Triangle fire, to promote various measures designed to prevent factory catastrophes, either from fire or other accidents. Representing the Committee on Safety, she cooperated with the Factory Investigating Commission in drafting the new provisions of the labor law, particularly those relating to the prevention of life hazards from fire and accidents. She is a member of the National Safety Council, of the Committee on Safety to Life of the National Fire Protection Association, and a director of the Consumers' League. She was Commissioner of the State Industrial Commission from February 18, 1919, to April 15, 1921. She was appointed a member of the Industrial Board in January, 1923. On January 19, 1926, Miss Perkins was appointed chairman of the Industrial Board, and on January 9, 1929, was appointed Industrial Commissioner.

Contributed by the Committee on Education and Training and the Management Division and presented at the Hartford, Conn., Meeting, June 1–3, 1931, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

the school, must wander objectless around until by chance he finds "a job."

An integral part of this work is vocational guidance, but that science is in its early infancy, and it will not be for many more years, unless some comprehensive studies are made, that we may expect any improvement.

Many of the larger industries of the country have been forced rather than led into sporadic efforts to train their workers, but whether it may be considered an evidence of strength on their part or of weakness in our teaching system is not so very clear. Our large industries cover only a very small proportion of the workers of the country. There is a vast number of small plants to which the torch of light never penetrates. Evidence of competence is demanded of doctors, chiropractors, and stationary engineers, but not of those who undertake manufacture; and there is no organization in the country to prevent fools from entering where angels fear to tread. The industrial structure is thus weakened from the start.

Now, when you have amateurs in industry shouldering the burden of manufacturing, what is to be expected from their ability to train the workers who have the misfortune to work in their shops? With very few exceptions men undertake to manufacture goods with only the most indifferent knowledge of the demand. They are perhaps setting out against insuperable competition and are wholly unaware of the fact.

The defects we find in the educational field in industry are not due to industry; they are principally due to the faulty structure of education itself. We, in America, are firmly of the belief that specialization is the secret of success. Foreign educators hold to exactly the same belief, but they attack the problem differently and with more success. We believe that the way to attain business, and even professional success, is to begin specialization at an early stage of our college careers. In Europe, it is believed that without a full general training it is not possible to begin specialization with any hope of success.

Our colleges turn out some specialists; they are now getting on the band wagon of research in industry; and because of this radical defect in our educational viewpoint, it is feared that there will be a great slump in the present vogue of research in industry—not because research is undesirable, but because with our present ideas in training, our efforts in research may be unproductive. It is noted, too, that research is generally confined to manufacturing and not to personnel problems.

It would undoubtedly seem that, if economically impracticable by plants, it would certainly be practicable for industries to consider the matter of training workers, especially in repetitive work. No man who has any appreciation of the value of time can endure doing repetitive work without a study of "the one best way," yet they nobly endure this in the workers under them. But training for such work involves careful selection. Certain types are fitted for such work and can make a success of it, while others would fail entirely.

The great stumbling block against the introduction of any sort of training in industry is the question, "How can the thing best be done, and what does it cost?" Now, training is not like a pound of butter; its cost cannot be obtained without effort. One would fancy that industry, asking "what does it cost," is really interested in the answer to the question, but that is not so; it is interested in putting off the issue by asking a question to which an answer is not available. The object of knowing costs is to determine whether training, on the dollar and cents basis, is worth while, because, whether one likes it or not, that is the criterion.

There are very few cases on record, it is believed, where a cost comparison has been instituted between the ordinary method of getting out production and that after a period of training. It is quite generally held that every man who is put to work is, in fact, being trained, but for the sake of clarity, two classes of training are recognized in industry: training by absorption and training by intention. As regards manual workers, at any rate, "it is apparent that the vast majority of American industries give the training of their manual workers little definite attention;"[2] that is to say, they train by absorption, which is really no training at all.

This brings us to a consideration of the foreman of whom we are told that "teaching ability is an essential qualification for the foreman," and if it be so, then it is apparent he has not exercised the essential qualification to any extent.

There is, in fact, an apparent difficulty here. If teaching ability be an essential, then how is it to be ascertained? That is an embarrassing question because it raises the whole issue of training in industry, yet out of the millions of foremen who have been operating in industry the past quarter of a century, no system of ascertaining teaching ability has been evolved.

We have, within the last two decades, gone from one extreme to the other and back again with regards to the position of foremen in industry. At first he was the supreme autocrat; he had the full power to hire and fire, and he exercised his prerogative to such effect that management then swung to the other extreme. The prerogative was pitchforked into the lap of the newly hired personnel manager. This plan, in theory conceived to be perfect, did not work out quite so happily in practice, but for a time it was endured, and in the meantime shop committees became the fashion, and with their introduction the foreman suffered a further eclipse. The main defect of shop-committee management was its failure to consider the foreman. In fact, it is now clear that, had the foreman been trained, many of the conditions that were responsible for the introduction of shop committees would have been obviated. The shop committee was mainly engaged in settling questions that ought to have been settled by the foreman.

The result of this situation was the movement toward foremanship training, and, in the main, it has been productive of good. The foreman has been reestablished in much of his former glory.

Training the foreman is a rather peculiar problem. There are colleges of pharmacy, schools for nurses, seminaries for clergy, police schools, dancing schools, mortician schools, hair-waving schools, and vocational schools and the apprenticeship system, but very few schools for foremen. All of the academic courses mentioned are for the youth who hope to become trained at their respective trades, but it is different with the foreman. The former have left school only recently; the foreman may never have graduated from high school, and even so, there will have been a long interval between the time he left school and the time when he has reached foremanship. His whole career will not have made him very receptive for any kind of formal training, although it is generally believed that foremen are willing to learn provided the teaching is interlarded with plenty of tact and diplomacy.

Since foremen are trained in industry and seldom in any kind of regular, full-time school, there is a difficulty in finding trained teachers to train foremen, if such a class has ever existed. Most teachers have, themselves, to get a diploma to teach whatever subject is to be taught, yet there is no regular school for foremen teachers, but of course there are many training courses throughout the country, unfortunately too limited in character. Nor, even if such a development were possible, would it be possible to bridge the gap between Mahomet and the mountain unless our present notions on the subject undergo a change.

As has been intimated before, production is one thing and teach-

[2] "Training Manual Workers," R. N. Keppel, Production Executives Series, No. 72, American Management Association, 1928.

ing is another, and unless there is a reconsideration of the whole teaching problem in industry, and some form of alliance established between our academic system and our industrialists, no solution of the problem is very apparent.

Looking at the problem from the viewpoint of the New York State Department of Labor, it is recognized that we have a long way to go before the training of foremen and apprentices is likely to materially assist in a solution of the many problems that confront us.

We have looked, from time to time, to the personnel manager for help without any very satisfactory results. The author has always felt that he, in the large plants of the State at least, should be the one person who should be acquainted with the legal aspects of our labor problems and that with him, as regards the administration of the Workmen's Compensation Law, should be the safety engineer. Neither of these influences has made itself palpably felt, either to us or to those with whom they come in contact in industry. Some attempts are being made along this line, and such a development is highly desirable.

We are still attempting to force on industry by means of the Labor Law established truths which aim to the betterment of the worker and, incidentally, the employer, whereas had the personnel manager and the safety man the right conception of their positions, we should find them seeking our aid rather than occupying a neutral or indifferent position. Their situation often forces them to take a stand which they conceive to be acceptable to the employer, but the very opposite is the truth. They misinterpret the position of the employer, as often as not, and merely because they have only a misty knowledge of the situation. Take a case in point. The law requires that the workshop shall be free from dust, fumes, and gases when they exist in harmful quantities, yet we have seldom had a single enquiry from a personnel manager or medical department in industry as to whether the conditions were harmful for workers or not, and, what is worse, we have never had any enquiries from such functionaries when a building was in prospect as to the proper sort of ventilation.

In view of these considerations it is not to be wondered at that the foreman is ignorant of the labor law, the health requirements, or safety.

One feature of industrial training might be brought to the attention of industrialists in the State of New York, because with comparatively little extension of powers, the same system might be extended to include the training of foremen in industry. If, as has been established, the worker gets no regular training, and the foreman, who is supposed to teach him, gets little regular training, then the whole hierarchy of training in industry is faulty, and some scheme is necessary to establish a real system of value to industry.

Under the Smith-Hughes Act the University of the State of New York, through the State Education Department, has a course for training trade, industrial, and technical teachers. The course requires 480 clock hours of work and 4 years of attendance at evening school in order to secure the certificate and license granted to graduates.

These courses are offered in New York City, in Buffalo, and in Oswego during the summer session, and elsewhere by arrangement.

Candidates should not be less than 21 or more than 35 years of age. They should have 5 years or more trade experience in the occupation in which they desire to train; they should at least have 1 year of high school and the other ordinary qualifications.

Should labor management be taught in addition to the other subjects, this would provide a good course for foreman teaching which might be accomplished through the Smith-Hughes Act.

It would seem that with the development of some such plan the plane of industrial training would be greatly improved and the employers would obtain incalculable benefits from this departure.

Discussion

FRED H. COLVIN.[3] Training should be general enough to give a general idea of the trade, but modern production demands considerable specialization. It seems best to teach the rudiments of machines in a school, either a public trade school or a shop school, and get the boy beyond the stage where he is a nuisance and a hindrance before inflicting him on the foreman. But the final training can best come in the plant itself, on regular work.

One railroad shop superintendent reversed the usual process of his time. Instead of starting a boy on the bolt cutter and disgusting him with the whole shop idea, he put him in the erecting gang as a helper. In this way he learned the names of parts, where they went, and what they did. He learned what parts had to be carefully fitted and where a loose fit was permissible or desirable. He also learned to work with men. Then when he went on the bolt cutter or the lathe he knew just what parts he was making and had a greater interest in the job.

Superficial training or too specialized training makes it difficult to shift men from one kind of work to another. They fear to tackle a new kind of job.

A. E. CROCKETT.[4] About this training of apprentices and foremen—the training of workers to do skilled jobs and of men to lead them—the Department of Education and Department of Labor should see if they themselves are not more at fault than is the industry. They forced laws upon the statute books to prevent the exploitation of young people by the unscrupulous. That was proper. They then went farther, and had laws extended to prevent employment and training in order to protect the adult from juvenile competition.

The inevitable has happened. The Department of Education and the Department of Labor, disturbed over conditions which they helped bring about, are complaining that industry is not doing the training that it ought to do. In fact, they are so disturbed that they are offering to go down into the plant and do the training themselves. And they will continue to be disturbed, because as soon as business picks up, there will be a shortage of persons capable of doing skilled jobs, and at the same time, as in the hectic days of 1928 and 1929, there will be a milling mass of unemployed, most of them being between 20 and 25 years, incapable of employment.

They will be incapable of employment largely because the State Department of Education and the State Department of Labor have helped put laws on the statute books that prevent the necessary training, and not only will do nothing about correcting the situation, but quietly or openly oppose any effort by any person else to correct it. The laws in New York State are such that boys and girls, instead of getting the practical education and training in employment that they need when they are young, get jobs, when they get any at all, only as errand boys, clerks, etc.

In New York State, up to 18 years, they cannot even sharpen the tools they work with. There is a long list of machinery at which they cannot be employed. There is the double compensation feature, too.

The result is that instead of the employer trying to get young people to exploit, he cannot be coaxed even to give them jobs at which they can acquire skill—in the precious years of their life, so far as acquisition of skill is concerned.

No wonder these complaints about training. The Education

[3] Editor, *American Machinist*, New York, N. Y. Mem. A.S.M.E.

[4] Manager, Industrial Management Council, Chamber of Commerce, Rochester, N. Y.

Department and the Labor Department do not seem to have realized the revolution that is going forward in industry; they seem to think of training in terms of 35 years ago. Preliminary schooling for college is not what the prospective skilled job worker needs either. It is not doing good to help make the young feel that if they are headed for work they are failures.

Parents are in part to blame. So many of them hate to think of the son working like the father did and they feel the bottom drop out of the world if the son does not want to be a doctor, a lawyer, a clergyman, or at the worst a mechanical engineer. It is bad enough to have parents that way, without having the educational system of the country operated on the same theory.

We shall have to go back to where training of body (the touch, the hand, the eye in a productive way) will go hand in hand with the training of the mind.

The writer believes we are approaching days when every youth will be taught some skilled job, even though we do not go as far as to give him the exact equivalent of our more or less obsolete trades.

If the schools get busy with our youths in the interest of that great proportion of them that must go into industrial or other manual employment, and if our State Education and Labor Departments will make it possible for them to get training on the job—on some part-time basis for those in their last public-school years, for instance—the better it will be for America and most everybody in it.

MAN-53-6

Value of the Safety Movement in the Industries[1]

By JULES PODNOSSOFF,[2] BROOKLYN, N. Y.

The paper outlines the various aspects of the industrial safety movement, its attractiveness, economies, and inherent possibilities. The safety movement has been long ago divorced from altruistic conversation and put on a basis of cooperative helpfulness. Accordingly, the subject has been treated from a "cold" business standpoint in terms of dollars and cents; that is, in the approach to the problem the center of gravity has been shifted to the economic rather than humanitarian aspects. A brief historical review of the growth of the movement is first made, followed by some statistics of the losses in life and property caused by industrial accidents. The principal emphasis in what follows has been placed on the thesis, at present conclusively proved, that efficiency in production and safety are inseparably correlated. To afford a better perspective of the actual values involved, illustrative examples taken from the experience of different industrial companies are introduced. The role of engineers in accident prevention is next considered, and the discussion is concluded with several recommendations.



THE essence of this paper may be condensed in the following: 1 Because of the tremendous burden of economic and human waste, safety becomes a problem of supreme national importance, in view of the increase in the frequency and severity of accidents with an increasing intensity of industrial activity.

2 It is possible to control industrial accidents under modern conditions of highly efficient productivity—"engineering revision" being the most effective means.

3 Efforts to improve safety performance do not interfere with production; on the contrary, the experience of large groups of industries throughout the country indicates that material reductions in accident rates can be attained simultaneously with an increase in production rate.

4 Apart from ethical, moral, or humanitarian motives or obligations, it is imperative that safety be made a major executive objective and responsibility, because:

(*a*) It is cheaper to prevent accidents than to allow them to take place

(*b*) Maximum industrial efficiency can be secured only when the accident performance is brought down to an irreducible minimum.

5 Engineers, thanks to their particular professional training, are singularly well suited to assume leadership in the safety movement, and thus effectively to cope with industrial accident prevention—a vast field of activity with tremendous possibilities.

The United States is rapidly forging ahead of other countries, mainly because of her industrial supremacy. In her industries the miraculous achievements and inventions of the scientific and engineering genius are reflected as in a magnifying glass. It is because of the large scales employed and because of the tremendous and ever-increasing intensity of production that American industry can afford to have a working class that commands higher wages and maintains a standard of living superior to that of the workers in any other country, and at the same time carry on a successful competition in the world markets. This enormous increase in intense industrial productivity, however, necessarily leads to the consideration of a problem of first magnitude—the problem of safety.

The Problem of Safety

In spite of the remarkable achievements of American industry—achievements which are the basis of unparalleled national welfare and a source of national pride—a study of the performance of the most representative and even of the best plants throughout the country indicates that great improvements can still be brought about; and most important of all, that, at the present stage of development, highest efficiency in production and operation will be attained only when the amount of waste shall be brought down to a minimum.

Of all industrial waste, human wastage is the most inexcusable from every point of view. The industrial and economic factors involved in human wastage, apart from the humanitarian aspects, make the question of plant accident performance a problem of first-class national importance. But before proceeding to a discussion of the basic aspects of safety, it may not be out of place to make a brief review of the development of the safety movement. (1)[3]

Development of the Safety Movement

The beginning of the safety movement as such may be said to date as far back as some nineteen years ago when the first workmen's compensation laws were passed. However, the first organized effort in accident prevention was made much earlier, viz., in 1867 in Alsace-Lorraine, and the idea began to spread rapidly in the most progressive European countries.

In this country the first effort to prevent accidents was undertaken on a large scale by the United States Steel Corporation in 1907. In 1910 the railroad companies adopted the slogan of "Safety the First Consideration" as their operating rule to reduce their appalling accident rate. On account of the great number of casualties in the iron and steel industries, a group of engineers connected with the latter arranged a conference in Milwaukee in 1911, followed by a second meeting in 1912, at which many representatives from other industries were present.

The organization of the National Safety Council was a result of these meetings. The National Safety Council is a rapidly growing organization, non-sectarian and non-commercial, having at

[1] This paper received the Charles T. Main Award in 1930.

[2] Student, Polytechnic Institute of Brooklyn, at present graduating in Mechanical Engineering. Born in 1908, brought up and received both elementary- and high-school education in Moscow, Russia. Wanted to study engineering, but could not do so in Russia. Taught himself English, and arrived in the United States in September, 1927. Immediately upon arrival enrolled at the Polytechnic Institute of Brooklyn, where he was last year designated honor student. Interested in Student Branch of A.S.M.E.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

present a membership of 4000 industrial concerns—the flower of industrial and railroad service—and is doing magnificent work in carrying on safety service and propaganda throughout the United States.

What Price Accidents

It is essential to a clearer realization of actual values involved in the material and human waste which accidents entail briefly to recount a few statistical data, although it must be mentioned that owing to the variability of the different factors it is impossible accurately to estimate the enormous economic loss caused by industrial accidents.

To take, for example, the year 1929, as a rather typical year, the mortality statistics of the United States Bureau of Census show that 23,000 men were killed during that year in industry. If each death be taken as representing a loss of only $5000, this source alone accounts for a loss of $115,000,000. (2)

According to Mr. Beardsley, (3) on the basis of data obtained from the industrial centers of the country, 50 out of every 500 men who are engaged in industrial work[4] are eventually hurt. Of these 50, 3 are seriously hurt, 1 dies, and some of the others are permanently injured. It has been estimated that, on the average, approximately 3,000,000 men are injured in industry every year, and that in terms of wages industrial accidents for but one year resulted in a loss of $827,000,000, (7) the burden falling not only on the workmen but on society in general.

The foregoing survey does not tell one-half the story. The mind is fairly staggered when one begins to consider the losses due to the following:

1 The disorganization resulting from additional labor turnover
2 Increase in damaged machinery and spoiled material because of the replacement of the killed and injured by new men
3 Loss of time connected with each accident
4 The reaction of accidents on the morale and psychology of the workers, foremen, and company officials
5 A yearly loss of half a billion dollars due to fires alone
6 Occupational diseases resulting from unsanitary or otherwise improper working conditions.

These facts become still more striking when it is realized that the major percentage of accidents are preventable. This has been established definitely beyond question and is proved by the vast amount of information now available.

However, it would seem that as far as the employers of labor are concerned an emotional, ethical, or moral approach to the problem is inadequate; it does not suffice to picture the wrecked homes and lives, ruined hopes and ambitions, the suffering and the misery and the sorrow which grimly follow in the trail of accidents. That is, it is impracticable to set the "purely human" against the "money" values. From this standpoint the following thesis assumes a profoundly significant importance: *The safe factory is the most productive and the most efficient factory.*

Relation of Safety and Efficiency

At the request and with the financial aid of the National Bureau of Casualty and Surety Underwriters, the American Engineering Council undertook two years ago a study of the relation between industrial safety and efficiency of production. The results of this investigation are embodied in a most thoroughly exhaustive and singularly well-organized report. (4) Before stating the chief conclusion at which the authors arrived, it may be of interest briefly to outline the work accomplished, to give the reader an idea of the unique thoroughness with which the research had been conducted.

The investigation was nation-wide in scope and covered the following industrial groups: Cement, coal, mining, coke, electric railways, mineral mining, quarries, steam railways, chemicals, electric light and power, gas, iron and steel, machine building and metal working, paper and pulp, telephone and telegraph, textile, woodworking, and a large miscellaneous unclassified group.

There is a wide range in the size of the companies included, varying from 8 to 82,000 employees. Through the field investigation information was obtained from 1145 individual plants located in 40 different states. Investigation centers were established in 15 industrial cities, and definite territory was assigned to each field engineer, with local committees composed of industrialists, engineers, and safety men in many localities.

The vast wealth of detailed statistical information obtained, as well as the thorough analyses of the levels of plant performance, brought the authors of the report to the following outstanding conclusion:

"There is a close relationship between safety and production—a relationship so pronounced and consistent throughout as to indicate a fundamental of management, which may be stated as follows: Maximum productivity is dependent upon the reduction of accidents to an irreducible minimum. The data for the 13,898 plants consistently show that those plants and industries which have the highest production performance also have the lowest accidents rates, which fact sustains the fundamental stated. This fundamental clearly indicates that if the management desires maximum productivity it must initiate, direct, and control ways and means of keeping accident frequency and severity rates at an irreducible minimum."

In order to enable one to have a better appreciation of the possibilities of the safety movement and of its attractiveness and economies, particularly with reference to the foregoing fundamentals, it may be timely to introduce a few examples.

Illustrative Examples

1 The following experience of the Du Pont company is very instructive:

During the war the Du Pont company built a smokeless-powder plant and a shell-loading plant for the United States Government. On the basis of their previous accident experience, accidents on the smokeless-powder plant job would have cost them at least $452,752. For economic as well as humanitarian reasons the Du Pont company decided to carry out a program of intensive safety work. The total cost of safety work plus compensation and medical expenses amounted only to $116,000—i.e., the company had a net saving of $336,652. The insurance rate on the shell-loading plant was $103,545. Thanks to efficient safety work, the safety and accident expenditures amounted oniy to $16,233—i.e., on this job alone safety work resulted in a net saving of $87,312. (5)

2 An interesting illustration of the psychological effect of safety work on the workmen's efficiency is reported by the California Industrial Accident Commission: (4)

"At the Moccasin power plant of the Hetch Hetchy project a surge chamber 40 ft. in diameter by 160 ft. deep was being constructed of concrete. After the concrete had reached a height of 80 ft., it was decided to stretch a life-net between the walls of the chamber to serve the double purpose of saving the life of any workmen who might fall from the top and also to provide protection for the men below from falling timbers. While the net was installed solely with the idea of affording protection, it actually resulted in reducing the cost of the work, it is stated, due to the added sense of security afforded the workmen, which enabled them to speed up all operations in connection with the

[4] Figures are specific for the electrical utility industries.

concrete lining, with the result that the time required for completing a 5-ft. lift was reduced from 56 hours to 48 hours, an actual saving in labor cost of nearly 20 per cent."

3 In his recent book on industrial safety (6) Mr. L. A. De Blois says the following:

"Safety and efficiency are unquestionably linked together by what is best described as 'continuity of operation.' For continuity of machine operation—that is, useful operation—we must have continuity of feed and stock removal, as well as continuity of power supply. The joint effect on safety and production wrought by common sense changes in the methods of feed and stock removal is well illustrated in the operation of punch presses."

A case in the point is the Simmons company, related as follows:

"In 1919 when there were 700 presses operating in one department, 36 fingers were taken off in accidents. With a change in the methods of feed and stock removal the injuries decreased to 7 fingers during the first six months of 1920. In the four years following not a single finger was lost, and the production per press had increased on the average 60 to 65 per cent. Our experience has been that whenever we made a job safe we also increased the production from 15 to 150 per cent."

Here are two more illuminating examples cited by Mr. De Blois:

4 At the Newburgh plant of the Du Pont company accidents at the presses used for embossing Fabricoid, a leather substitute, led to a development of guards, and later to additional safety installations. Through the elimination of accidents, production per press was increased 80 per cent.

5 In the manufacture of dinitrobenzol by the dye works of the Du Pont company production has been increased 100 per cent in one year coincident with a reduction in the number of injuries (including poisoning gases) to 50 per cent.

6 In concluding the presentation of the illustrative material it may be of interest to cite Mr. Albert W. Whitney, (4) as follows:

"The question arises: 'Can they (accidents) be controlled? Perhaps, after all, with an increasing intensity of industry, an increasing hazard is inevitable?'

"The answer is a simple one, and not only conclusive, but suggestive of the solution in general; the answer is, namely, that individual cases exist, characterized also by high industrial efficiency, in which this control has been secured.

"The United States Steel Corporation in 13 years has decreased its accidents 86 per cent; the Union Pacific Railroad has a safety record for shop employees that is five times as favorable as the average of the other large railroad systems of the United States; the Clark Thread Company has a record of nearly 10,000,000 man-hours without an accident; one of the plants of the Du Pont company with 65 employees has a record of 11 years with only one accident, and that a relatively minor one."

A close consideration of the cases presented in the preceding, as well as the inherent nature of the subject, would lead one to contemplate the relation between engineers and industrial safety.

Relation Between Engineers and Industrial Safety

The foregoing examples indicate that the technique of safety organization involves essentially:

1 Planning and layout of plant, equipment, and operations with a view to maximum safety.

2 Inclusion of the mechanical and automatic safeguarding equipment in the original machine design and construction.

3 Location and elimination of hazards by superimposing and adjusting safeguards; i.e., fundamental changes in or redesign of operating practice, processes, and structures.

4 Substitution of industrial railways, conveyors, hoists, lifts, chutes, cranes, automatic feeds, and other machine devices for the human element.

5 A psychological approach to the workingmen, such as:

(a) Systematic education in safety
(b) Consideration of physical and mental fitness of operator for the particular job
(c) Placing the right man in the right place.

In the modern literature of the subject the first four points are referred to as "engineering revision," although the latter is ordinarily employed in the sense of item 3. Item 5 has been generally termed "human engineering."

In the early stages of the development of the organized safety movement, "human engineering" was particularly emphasized. But in the course of time experience has shown "engineering revision" to be of surpassing importance. Mr. Beardsley (3) is authority for the statement that, according to accident statistics,[5] the more highly trained men are the more seriously injured. It can thus be stated without any fear of contradiction that, thanks to his practical and theoretical training, the engineer is the man best equipped to tackle the problem of safety organization—a large, new, and interesting field, full of great possibilities.

Conclusions

On the basis of the foregoing discussion of the various aspects and possibilities of the safety movement, it may be well worth while to emphasize the following practical recommendations:

1 It is necessary that safety be made a primary executive interest and a managerial responsibility.

2 The states should adopt uniform safety codes and establish universal periodic health examinations.

3 The safeguarding equipment should be made an integral part of machine design by the manufacturers.

4 Engineering colleges should include safety as an important part of their curricula.

When society is freed from the inexcusable burden which industrial accidents impose upon it, the entire human life will be elevated to a higher level of comfort and happiness, and by an active leading participation in the industrial safety movement, engineers will add credit to the profession, and human life will be made still richer and deeper for the engineers' predominant influence upon it.

Bibliography

(1) "Industrial History of the United States," by Katherine Coman, 1916, Macmillan Co.

(2) "Handbook of Safety and Accident Prevention," Section on Statistics, by Fred G. Lange, The Engineering Magazine Company, New York, 1926.

(3) Address to Student Branch, A.S.M.E., Polytechnic Institute of Brooklyn, N. Y., Nov. 22, 1929, by C. R. Beardsley, electrical consulting engineer, Brooklyn Edison Company, and chairman of the Accident Prevention Committee of N.E.L.A.

(4) "Safety and Production," a report by the American Engineering Council, Harper & Bros., publishers, New York and London, 1928.

(5) Proceedings of the Eighth Annual Safety Congress, F. L. Hurlbutt, E. I. du Pont de Nemours & Co., Wilmington, Del.

(6) "Industrial Safety Organization," by L. A. De Blois, McGraw-Hill Book Company, New York, N. Y.

(7) "Accident Prevention and the Engineer," National Safety Council, reprint from 1925 Annual Congress proceedings.

References 2, 4, and 6 were those found of particular value. Very helpful in the preparation of this paper were also the current issues of *Mechanical Engineering*, *Safety Engineering*, *Industrial Management*, as well as the various publications of the National Safety Council.

[5] Of the electrical utility industries.

MAN-53-7

Symposium on the Stability of Employment in Industry

LONG BEFORE unemployment became the problem of the hour, the Management Division of The American Society of Mechanical Engineers had planned to have a Round Table conference on the subject upon Dec. 3, 1930, during the Annual Meeting, with the American Management Association cooperating. It was proposed to have brief outlines informally presented by numerous representatives of industry rather than a program entirely of formal papers. In order to secure the utmost freedom of discussion, it also was proposed to consider both these outlines and the impromptu discussions that might follow as confidential; that is, they were not to be printed without the consent of the speakers in every instance.

Mr. Ralph E. Flanders, Vice-President of the Society, presided and opened the meeting by recalling the challenge which Dr. Wesley C. Mitchell[1] had given engineers to face economic problems. Mr. Flanders stated that this meeting, although arranged far in advance of Dr. Mitchell's address, was nevertheless an acceptance of the challenge. He then called for the presentation of the three formal papers and the general discussion that should accompany them.

Manufacturing in Advance of Orders in Seasonal Industries

By EDWIN S. SMITH,[2] BOSTON, MASS.

The methods used by American business concerns to reduce seasonal slumps have formed the basis of study for the opinions of the author. Efforts to reduce seasonality may take the form of striving for a more uniform sales demand throughout the year, an attempt to get orders further in advance of the season, or a more level production curve. The experience of a number of manufacturers is related in the effort to show preferred methods in many lines.

THE data and opinions set forth in this paper are the result of a study by the author of methods used by American business concerns to reduce seasonal slumps. The study was made under the auspices of a committee of New England business men consisting of Henry S. Dennison, President, Dennison Manufacturing Company, Framingham, Mass.; John H. Fahey, Publisher, *Evening Post*, Worcester, Mass.; A. Lincoln Filene, Chairman of the Board of Directors, Wm. Filene's Sons Company, Boston, Mass.; Henry P. Kendall, President, Kendall Mills, Boston, Mass.; and J. Franklin McElwain, President, J. F. McElwain Company, Boston, Mass.

Effort to reduce seasonality may take the form of striving for a more uniform sales demand throughout the year, an attempt to get orders further in advance of the season, or a more level production curve, or all three. Numerous instances can be found in American business of these various types of procedure.

The author's belief is that the solution of the problem of seasonality by the first of these methods, which takes the form of filling in the valleys of a normal seasonal dip in sales, is the most difficult to achieve. Sometimes new uses can be developed for a product which will appeal in the off season. Again, by slight alteration a seasonal product may be made into a year-round seller. But the cases when this can be accomplished appear relatively rare. To stimulate sales in the off season by making a special advertising and sales drive may or may not result in new business at a profit. If the effort keeps down overhead and keeps up the morale of salesmen, such drives for off-season sales are no doubt often justified. In general, reluctance of the consumer to buy certain products at certain seasons represents a deep-rooted habit whose force the manufacturer is bound to respect.

There are many cases where a manufacturer has helped to fill the hollows of a seasonal sales curve by cultivating a foreign market in the Southern Hemisphere, which has seasons opposite our own. Seeking a diversified domestic market, both geographically and by trades, may afford opportunity for extending sales through the dull season.

To introduce into the line a new item which enjoys a busy season in the dull season of the company's regular product seems to offer to many manufacturers an attractive avenue out of the slough of seasonality. Several have followed this plan with success. However, if the new product requires a substantial departure in sales or production technique, it may prove a risky undertaking. Also it may grow at a faster rate than the regular line and so create a seasonal peak of its own.

A common method of trying to regularize seasonal production is to endeavor to get orders in earlier. The retailer's marked present preference for buying close to the season makes this a

---

[1] Director of the National Bureau of Economic Research, New York, N. Y.; address, "The Economist's View," delivered on President's Night, Dec. 2, 1930, at the Annual Meeting of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. See *Mechanical Engineering*, February, 1931, p. 105.

[2] Director of Research, Committee to Study Methods of Reducing Seasonal Business Slumps.

Contributed by the Management Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

difficult policy to pursue. Even special discounts and datings many times will not tempt them. This is particularly true in industries where style plays an important part. Until the trend of fashion is known, the retailer prefers not to order. Chains, cooperative buying groups, and mail-order houses, which buy in large quantities, are obliged to indicate their requirements earlier in order to obtain prompt delivery. They are therefore a stabilizing influence on the flow of orders.

An inquiry among department stores revealed the fact that in almost every line the larger department store, at least, is placing orders with fewer manufacturers. Although up to the present this policy may not have resulted in more advance ordering, it seems that this should be its ultimate result. The more closely individual manufacturers and retailers work together, the more aware they become of each other's needs and the more confidence they acquire in each other's judgments. If the retailer has been taught by experience to have faith in the salability of a certain manufacturer's merchandise, he should be more willing to place orders with him before the direction of demand for the next season makes itself definitely felt. The simplification movement, by eliminating slow turning items, has for many kinds of merchandise made advance ordering by the retailer less risky.

The brief summary just given of some of the topics covered in this study indicates their complexity. In a brief paper delivered to an audience familiar with the details of management practice, the author believes that to isolate one phase of his study and treat it more comprehensively would be the most satisfactory procedure.

One phase of regularization can, in a short period of time, be discussed with much greater satisfaction than many. For purely selfish reasons also the author is anxious to get the reactions of a technical group on the particular subject picked for presentation, namely, manufacturing in advance of orders. Time does not permit treatment of the subject of budgeting production in a seasonal business, but budgeting quite properly precedes the manufacturing to stock.

The attempt to achieve level production despite a seasonal sales curve is perhaps more often made by manufacturing to stock than by any other method. Under present conditions of close-to-the-season ordering by wholesalers, manufacturing in advance of demand is virtually a necessity in most seasonal industries. Even though a concern may not deliberately set more level production as a goal desirable to achieve for its own sake, it must ordinarily do some manufacturing in advance of orders, so that it may have merchandise ready to deliver when active buying commences.

Instances of Advance Manufacturing

The May Oil Burner Corporation, whose business is distinctly seasonal, feels it necessary to build up a certain amount of stock in dull seasons to meet later delivery demands. However, it attempts to achieve level production for about eight months only in the year, two months going extremely high and two months about 25 per cent higher than normal. Finished stock is not in excess of 90 days' shipment in the dullest time of the year, or two weeks' shipments in the busiest. Employment is much steadier than production because overtime is resorted to in the peak months. This policy serves to cut down lay-offs in the dull months.

The Gruen Watchmakers Guild of Cincinnati sells as many watches in four months of the year as in the remaining eight, the peak seasons being at graduation and the Christmas holiday season. It employs about 2800 persons and states it has never let anyone go because of lack of work to do. During the eight months of lower sales it manufactures about twice the quantity of watches which it sells.[3]

[3] Radio talk by Dr. Francis H. Bird, University of Cincinnati. March 24, 1930.

The Crompton Company, manufacturer of velvets, plushes, etc., has a selling season which is largely concentrated in August, September, and October. As it takes two to four months to prepare the fabrics, it is obliged to undertake manufacturing in advance of orders. If, then, the season turns out unsatisfactorily, production is curtailed. This sometimes necessitates a carry-over of finished goods until the next season. "Obviously," the company states, "this requires large amounts of capital and considerable patience. So far we have found no better solution of the problem."

The Norton Company, Worcester, Mass., maker of grinding machines and wheels, manufactures to stock in accordance with sales forecasts balanced against variation in incoming orders. Stock is classified by items on the basis of turnover, and when stock is being accumulated, it is made up of items which are most active. This is of course the soundest method to pursue in any program of manufacturing to stock.

Keeping the Labor Supply Intact

Keeping the labor supply intact is a reason which frequently determines a policy of advance production in months of dull sale. This may be a sufficient motive, even though to let manufacture fluctuate with demand would be in other respects cheaper.

The refrigeration department of the General Electric Company has a sales curve which reaches a high peak in the warm weather. This company stated that it would be more economical to have its monthly production rate vary with sales, particularly as improvements by competitors may affect the sale of models which are manufactured in advance of demand. However, the machines used in the manufacture of refrigerators are built to very limited tolerances and require a high degree of precision. To make sure of keeping a properly trained working force, the company has felt it necessary to produce about the same quantity of refrigerators every month.

In a small community the need for maintaining a low employee turnover may be particularly acute since the labor supply cannot be readily replenished. The American Optical Company, Southbridge, Mass., has this problem. It feels obliged to do much advance manufacturing, although it finds this more and more difficult, as style appeal plays an increasingly large part in the sale of its products. It avoids this difficulty in part by carrying advance manufacture only up to the point of actual finishing. Concerns which are able to take this form of insurance against style change are in a favored position when they undertake to manufacture in anticipation of orders.

Factors Assisting Advance Manufacture

It is often possible for a manufacture of products which are fairly standardized to make up parts in advance, even though the final assembling of the product has to wait until specific orders are received from the customer. The Barrett-Cravens Company, manufacturer of lift trucks, portable floor cranes, portable elevators, etc., makes up in dull times various parts of the lift trucks and other products such as crossheads, platform tops, wheels, handles, and even frames.[4] The device of making repair parts for stock is employed by many concerns in an effort to get more level production.

The practice of standardization of parts and products and of simplification of lines is assisting many concerns to undertake advance manufacturing with less risk. The movement for limiting the variety of colors has assisted paint companies in knowing the proportions of certain colors that it is safe to manufacture in advance. The Sherwin-Williams Company finds that 30 per

[4] "Steadying Business: No. 2, The Metal Trades," published by the Policyholders Service Bureau of the Metropolitan Life Insurance Company.

cent of the demand for house paint is for white, with ivory, colonial yellow, and grays following. This enables that company to make up stock in advance and ship it to the warehouses.

Many concerns which do not dare to take the risk of manufacturing in advance of the season on their style lines are in a position to fall back on certain staples which can be made year in and year out, during dull buying seasons, feeling reasonably assured that those articles will prove salable.

The Hickey-Freeman Company, men's clothing manufacturers, states that it is able to offer workers year-round employment even though the men's clothing business is notoriously seasonal. One means used to achieve this result is to manufacture for stock blue suits, tuxedos, and other staples.

The Stehl Silks Corporation, broad silk manufacturer, is in a business which is drastically affected by styles and is highly seasonal. Also the process of manufacture takes three months from the time of shipment of raw silk to the throwing plant until pieces are received back from the dyers. Under these circumstances there is nothing to do but forecast styles as well as may be and to build up inventories in the slack season, in order to make prompt deliveries later. There are certain staples, such as georgettes and crepe de chines, which can be woven "in grey" before decision is made as to the design or colors which will later be applied. This removes some of the risk of advance production.

An important part of a well-thought-out program for manufacturing to stock is the continual effort by the sales division to get customers to order stock items instead of clinging to the often trivial variations which in many lines characterize special orders. The Eaton, Crane & Pike Company reports that it continually strives to interest customers in stock items because this gives a better chance to plan production ahead. This company carries a large stock which does not move for several months, but it believes this policy warranted by its results of steadier production and employment.

The Dennison Manufacturing Company reports that by studying a number of special orders, stock items which meet the requirements of most of these orders can frequently be worked out. This not only permits manufacture in large lots at times which balance the production schedule, but helps to eliminate rush orders and makes immediate deliveries possible.

The Link-Belt Company, of Chicago, states that it is always looking for items which can be grouped and carried in stock. This has been done successfully with crane parts such as loaders, vibrating screens, speed reducers, etc.

The Carborundum Company, of Niagara Falls, N. Y., maker of abrasives and refractory materials, reports a larger percentage of finished products as manufactured to stock and that more and more special orders are from time to time changed over into stock items. The concern, which has only a few customers whose wants are well known, may even in an industry much affected by style, successfully anticipate future orders in its desire to produce on a more level basis.

Because the Lumb Knitting Company, of Pawtucket, R. I., women's rayon underwear manufacturer, has only four customers, and its connection with each of those customers is one of long standing, it has made it its business to understand its customer's needs with such thoroughness that it has often successfully anticipated what quantities of certain items will be ordered, before the customer himself has ventured to enjoy any such confidence.

"In-Stock" Departments Necessitate Advance Manufacture

Manufacturers who have changed their distribution from jobbing channels to dealing direct with retailers, necessarily carry a larger stock. Their goods, which in pre-war days would have been shipped out in large quantities at certain seasons of the year in response to heavy ordering by wholesalers, are now more likely to be accumulated gradually with an eye not only to the customers' needs, but to the economy of level operation. Under an intelligent policy of inventory control, many manufacturers have found that the warehousing and jobbing functions they have been obliged to assume have had a silver lining in their effect on their production program.

Hand-to-mouth buying, with its emphasis on prompt delivery, has forced many manufacturers to develop elaborate "in-stock" departments. These are to be found today in almost every line of business. By means of these stock departments, the customer can re-order constantly during the season in small quantities. The need to maintain such departments has forced many manufacturers into a program of manufacturing in advance of orders.

E. T. Wright & Co., Inc., Rockland, Mass., maker of Arch-Preserver shoes, reports that it has increased the business done on stock shoes until this represents approximately 75 per cent of output. As it is necessary to anticipate busy stock-shipping months the company finds that its stock-manufacturing program "has helped amazingly in ironing out the seasonal valleys."

An extreme instance of manufacturing to stock is furnished by the United States Cotton Company, of Fall River, Mass., makers of Cape Cod shirts. The men's shirt business is highly seasonal. Thirty per cent of the year's business of this company is done in the month of December. Seeking to attract the department-store trade, it aims to keep a stock of shirts at all times sufficient to cover the demand for immediate delivery. It has no orders in advance and does not seek to get them. Naturally it cannot achieve entirely level production, but by specializing in a standard broadcloth shirt in a limited range of colors, it is able to manufacture in advance of demand with a minimum risk.

Building up in-stock departments has of course necessitated enlarging warehouse facilities. The existence of these facilities has undoubtedly made it possible for many concerns to manufacture for stock which they could not previously have done because of lack of storage space. Bird & Son has steadily increased its warehouse space in the last ten years in more than direct proportion to increased business, for the purpose of maintaining more nearly level production. As a result of this policy this company reports that no production department faces a complete shutdown from a lack of orders or storage space.

Research an Aid to Manufacturing in Advance

Many times when a manufacturer desires to spread his production more evenly, he has to undertake research to make it possible to store his product without danger of deterioration.

The New England Confectionery Company manufactures some 500 varieties of candies, some of which can be stored and others not. At one extreme are fudges which deteriorate rapidly, but lozenges at the other extreme can be stored for years. In between are various lines with different keeping qualities under storage conditions. Chocolates can be stored in properly conditioned rooms from two to six months. As a matter of fact certain candies are improved by storage.

The Procter & Gamble Company, soap manufacturer, acquired some cotton-seed mills which at the time were running only four months a year. This part-time operation was characteristic of the industry. The company promptly proceeded to equip every one of the 15 mills which were taken over with large storage tanks to store seed, and through research found a way to keep the seed from molding or heating. It now runs three of the mills 11 months of the year and could run the others also if there were enough seed in the locality.[5]

[5] From article in *Printers' Ink*, Apr. 17, 1930, "Steady Work—Management's Job," by R. R. Deupree, vice-president, Procter & Gamble Co.

Special Arrangements With Suppliers to Facilitate Advance Manufacturing

Carrying high-priced products in stock is expensive, and this cost deters manufacturers in many lines from advanced manufacturing. Sometimes the cost of manufacturing relatively expensive products for stock can be reduced by favorable arrangements with parts manufacturers. The Silent Automatic Corporation, maker of oil heaters, manufactures on an almost equal monthly schedule throughout the year despite the fact that 60 per cent of its business is obtained during August, September, and October. In order to produce in such large volume in excess of sales, this company took advantage of the fact that the motor manufacturers and control manufacturers from whom it bought had dull seasons at the same time as the Silent Automatic Corporation. Arrangements were made with these suppliers to furnish equipment on a basis of equal monthly shipments, the material not to be paid for until sales had been made by the Silent Automatic Corporation to its dealers or branches.

The Meyercord Company, maker of decalcomania transfers, has made an interesting arrangement with certain veneer manufacturers which permits the Meyercord Company to get into production early and to advance its delivery date at very little expense to itself. During October and November it permits the veneer manufacturers to ship anywhere from 40 to 70 carloads of veneers that are standing in their warehouses, arranging to settle for these veneers as they are used, the veneer companies paying the insurance; in other words, the Meyercord Company offers the veneer companies free warehouse space at certain seasons, but as their own warehouses are fairly depleted of finished products at such times, this arrangement can be made at no economic sacrifice. The veneer manufacturer is also able to clear his warehouses and start production earlier, an arrangement for which he is quite willing to receive deferred payment on the veneers which he has shipped out.

There are probably many instances of seasonal business in which the maker of the finished goods could use his warehouses for the storage of raw materials on which he might draw as needed to the mutual advantage of himself and his raw-material supplier. This type of arrangement offers a method by which production in advance of demand can be undertaken at considerably less expense than when raw materials are bought and paid for in advance of use.

How Food Producers Have Extended Production

Industries which prepare fresh food products for immediate sale, or which can and preserve these commodities, are governed in their productions by current supplies, and these supplies vary with the season.

Concerns in these industries frequently try to meet the problem of achieving more level production by attempts to spread the growing season of crops or the season in which animals are ready for slaughter. With canners manufacture for stock is automatically resorted to because the selling season does not correspond with the production season.

In addition to spreading the season in which supplies are received, meat packers are hoping to achieve more even sales by introduction of frosted meats which can be held in stock.

Canners in general are obliged to manufacture their products at a time when the quality of the main ingredients is at the best. The Campbell Soup Company was regularly faced with a large production load in the early part of the tomato season. However, through its research department it was able in this instance to harmonize the operations of nature with its own needs. It selected varieties of tomatoes which ripened at a later date and persuaded farmers to grow them, thus extending its tomato-packing season.

Through the activities of its laboratory, field-production department, and seed department, the California Packing Corporation (Del Monte products) is constantly endeavoring to secure new strains to extend the packing season. The California canners have interested the Federal Government in trying to develop new varieties of fruit to fill in the intervals when present fruit crops are scarce.

Difficulties in the Way of Manufacturing in Advance of Orders

Despite the most intelligent efforts of management, through standardization, simplification, anticipation of styles, etc., there are bound to be many cases in which a policy of manufacturing for stock, at least on an extensive basis, will be too risky and too costly to undertake.

Lack of certainty of demand for standardized product is a difficulty in the way of advance manufacture which confronts even manufacturers of producers' goods into which style, in the sense ordinarily conceived, does not enter. Due to improved design, the products of the industry which they serve may be changing constantly. Naturally their own products must keep pace.

The Black & Decker Manufacturing Company, maker of portable tools which are used in the automotive and aviation industries, is obliged to alter its product in accordance with mechanical and design changes in various makes of automobiles and airplanes. This situation would make it very risky for it to attempt a great degree of manufacturing in advance of orders. Moreover, style enters even into the tool industry. This company has found it necessary constantly to improve the design and accuracy of its tools in order to hold the interest of garage mechanics who are accustomed to changes in the products on which they work and the implements which are used.

The competitive situation is constantly engaged in making everyone impatient with products to which he might otherwise become accustomed. As long as this force is at work, few manufacturers without much justified trepidation can count on sufficient stability of demand to undertake manufacture in advance of orders.

The Edgemoor Iron Company attempted to manufacture in advance the principal parts of a number of different sized boilers. Changes in the design and construction of boilers made it impossible to find ready sales for this product, much of which had to be carried so long that any saving in manufacturing costs was offset in interest charges.

Even when the risk of a change in demand is not a factor which need cause the manufacturer particular concern, the expenses connected with holding stock in anticipation of orders may be greater than he feels it wise to meet.

The Edward G. Budd Manufacturing Company, of Philadelphia, is a large manufacturer of automobile bodies. Even if it could anticipate with safety, which it cannot, the type of bodies which the automobile companies are likely to demand in any season, it would not manufacture in advance because of the heavy storage expense. This same consideration is distinctly operative in the policy of unwillingness of automobile companies to manufacture for stock.

The value represented in stored goods is a burden which most manufacturers will avoid unless there are powerful considerations impelling them to undertake manufacture in advance of sale. Even though a manufacturer of heavy products concludes that he can save storage space by manufacturing parts and assembling them later, he has to bear a considerable cost in putting the material into storage and getting it out again, a cost which must be weighed in considering the desirability of manufacturing to stock.

Style Fluctuation Hinders Advance Manufacturing

In a vast range of consumers' goods, the probability of change in style or mechanical improvement has become so great that many manufacturers refuse to accept the responsibility of advance production, preferring the expense of expanding their plant and working forces to a point sufficient to handle peak orders.

The women's garment industry, both outer wear and inner wear, goes through decided shifts in color, material, and design from one season to the other, and often during the season, so that with the exception of some reasonably staple items few manufacturers in this industry can afford to take the risk of advance production.

Manufacturers of women's hosiery, like shoe manufacturers and underwear manufacturers, are obliged to carry large stocks for immediate shipment, which is risky because of changes in the demand for colors. The stocking manufacturers have, however, an advantage which the dress manufacturer does not possess in that they can manufacture stockings in the undyed state, dyeing them as the season advances and the trend of demand is disclosed. Also stockings dyed the wrong color can be redyed, but naturally this is an expense to be avoided if possible.

The jewelry business is definitely seasonal, with a high peak in December. It has always been a business which depended very much on the introduction of novelties, and with the coming of costume jewelry this tendency has increased. The R. F. Simmons Company, in Attleboro, Mass., has as a chief product men's watch chains, which are probably as near a staple article as is being produced by the jewelry industry. Nevertheless, operating schedules in this company are cut from one-fourth to one-third from March 1 to September 1, since the risk of advance manufacture is held to be altogether too great.

The radio industry is at present passing through a stage where advance manufacturing is definitely dangerous. At least this is the opinion of many prominent radio manufacturers. The selling season is very definitely concentrated in the winter months, and the manufacturer who attempts to build up stock out of season is likely to find that mechanical improvements in the sets of his competitors will very probably make his own stock unsalable at profitable prices.

The Hoover Company, maker of vacuum cleaners, points out as a difficulty of advance manufacturing that when new models are brought out the accumulation of old models held in stock has to be worked off on unfavorable terms. The automobile and radio companies, which are frequently forced into bringing out new models because of competitive offerings, are loath to accumulate large stocks for similar reasons.

The possibility of price decline is another factor which militates against holding of inventories. This may be true in certain lines even when stock is held for only a couple of months.

Fear of Deterioration as a Deterrent

Manufacturers of food products, with the exception of canners, often prefer not to manufacture for stock because of fear of deterioration. Candy manufacturers, through refrigeration and other devices, have overcome this handicap for certain of their products, but other types of candy cannot be successfully stored. The Coca Cola Company does not manufacture greatly in advance of demand, feeling that freshly bottled syrup is essential to the right flavor for its beverage. It encourages dealers to keep on hand only a 30 to 60 days' supply of Coca Cola. Canada Dry Ginger Ale, Inc., has experimented with keeping production on an even plane, but found it unsatisfactory because of the difficulties of handling heavy cases in large quantities for storage, and the fact that the product goes out again with great variance in age.

Dangers of Overproduction

Advance manufacturing always carries with it the danger of overproduction. This may be true even when the merchandise is entirely in line with consumer demand. When a seasonal business finds itself as the season progresses with heavier stocks than it can hope easily to dispose of, it feels forced to unload these goods at unfavorable terms rather than to undergo the risk and expense of holding them until the next season. That is why the Stehli Silks Corporation, for instance, which is obliged to manufacture in advance of demand, believes that the ideal situation under present conditions of hand-to-mouth buying is to carry sufficient stock to make prompt deliveries, but never to have out-of-demand merchandise on the shelves or to build up inventories in the dull period to a point where dumping and price cutting are necessitated in the busy season. Experience has demonstrated to this company that maximum inventory should never be more than two and a half times the average month's sales.

The most serious result to be apprehended from manufacturing to stock is that the building up of inventories by a sufficient number of concerns may reach a point which constitutes overproduction for the entire industry. This inevitably hastens the coming of a serious depression. Such a state of affairs is particularly likely to happen if piling up inventories in anticipation of seasonal business corresponds with a situation of general overproduction throughout the country whose presence has not yet made itself distinctly felt. That is what occurred in the radio industry, which in the fall and winter of 1929 and 1930 definitely overproduced and was obliged to go through a period of selling merchandise at distress prices. Such an unwise building up of inventories is of course reflected back into the business of the suppliers of the manufacturers who are overproducing. It is for this reason that the R.C.A.-Victor Corporation insists that all its suppliers accept R.C.A. orders for material with the distinct understanding that it cannot build up inventories.

In any seasonal industry an inventory policy that is directed toward the anticipated sales of a normal season might well prove a boomerang if the whole business world abruptly enters on a period of declining volume before this season arrives. To say this is not to say that a policy of building up inventories out of season is unsound. It becomes unsound only if the concern pursuing this course fails to check its inventory expansion constantly in reference to the present and probable future trend of sales in its own and other industries.

In its effect on stability of production and employment, lack of control of inventories by prospect of profitable sales is in the long run disastrous. Temporary employment and fuller use of productive facilities are obtained at the price of long periods of idleness for both capital and workers when business failures or prolonged shutdowns take their inevitable toll. Situations of this sort, occurring either in a highly seasonal industry or one with a more stable sales curve, are of course completely removed from the legitimate production in advance of demand, which is based on careful calculation of profitable orders to be expected when the buying season is under way.

Figuring the Costs of Advance Manufacture

A decision to manufacture for stock may come from any of the motives listed, such as the necessity for prompt delivery when the busy season opens and the desire to keep the labor force intact. Probably not many companies work out an exact comparison of what would be the difference in cost between a policy of manufacturing for stock and holding inventories, including interest and insurance, and a policy of resorting to overtime at peak periods or expanding plant space and adding to equipment in order to take care of the seasonal load.

Unforeseeable changes in price, money rates, or wage rates might well affect the validity of judgments as to the cost of manufacturing to stock or not manufacturing to stock. Nevertheless an attempt to figure these costs is warranted in view of the fact that stock is ordinarily held for a short time only, and therefore changes in price, etc., can often be anticipated.

In certain companies reporting in the study which the author has been conducting a policy of manufacturing to stock has been adopted only after careful consideration of its effect on costs.

The Ritter Dental Manufacturing Company made a study which indicated the definite seasonal character of its business. It then determined the additional value in inventory required to manufacture on a level basis. The result of the calculations showed that the cost of carrying additional inventory, both in storage expense and interest on investment, was more than offset by the cost of the additional investment which would be required in machinery and buildings to manufacture according to seasonal trend.

The Gorham Manufacturing Company of Providence, R. I., maker of silverware, have definite sales peaks in April, May, and June and in October and November. These are the popular months for weddings, and 70 per cent of the business of this company is in wedding silver. Its merchandise is fairly staple; it is meant to last for generations and therefore changes in style are made conservatively. This policy is also dictated by the fact that reorders for silver in patterns originally bought are an important part of its business. Before a change in management in this company took place some years ago production was carried on very much in accord with the sales curve. The new management found upon investigation that interest on merchandise held in stock was less than the overtime paid in old days when manufacturing was according to orders, to say nothing of the factor of impaired efficiency of workers who were subject to overtime schedules. The Gorham Company therefore deliberately divided planned annual production by 50 weeks, and has proceeded to manufacture on practically a level basis since then.

The experience of the Gorham Manufacturing Company at the beginning of the recent depression is an interesting illustration of the fact that a soundly financed company which has adopted a level production program need not necessarily fear that an accumulation of inventory will prove seriously embarrassing. The company, at the end of its fiscal year, Jan. 31, 1930, found itself with a larger inventory on hand than was desirable in view of the business depression. It proceeded to scale down anticipated sales for the year. As the product is not subject to rapid style obsolescence, the only embarrassment in connection with the inventory was the amount of capital invested. In laying out schedules for 1930, it took into consideration the inventory overage and computed the schedule of weekly production, having in mind both the additional inventory and the anticipated sales.

Results of Manufacturing to Stock

A policy of manufacturing for stock does not necessarily imply that level production is thereby achieved. The results of manufacturing to stock may range all the way from practically uniform production month after month to merely a small degree of anticipation of peak-season orders as a result of getting into production a few weeks or a few days earlier. However, there are probably numerous companies besides the Gorham Company which attempt, if they do not achieve, actual division of the production year into equal monthly or weekly quotas. The Procter & Gamble Company estimates the business it is going to do over a 12 months' period and divides it by 49 weeks. This company guarantees employment of 48 weeks to any worker who has been employed six months or more. The production schedule gives one week's leeway. The soap end of the business is not highly seasonal; the product is reasonably standard and can be stored without deterioration. Naturally these factors help to make a level production policy practical.

The White Sewing Machine Corporation points to the favorable result on employment of its manufacturing to stock policy as seen in the year 1929, when the month of smallest sales was 56 per cent of the peak, the month of smallest production 83 per cent of the largest, and the number of employees in the minimum month 92 per cent of the maximum month.

Even more significant testimony as to the spreading of production by means of manufacture for stock comes from the Wickwire-Spencer Steel Company. It attempted to manufacture window screens and poultry-yard screens in nearly equal monthly quantities, with the result that in this line of the business the ratio of production in the smallest month to the largest would be approximately 80 to 100, whereas in sales it would be only 18 to 100.

Summary

Some production in advance of the seasonal rush has become almost necessary in the present era of close to the season buying; otherwise deliveries cannot be made on time, or business will be lost because the capacity of the plant is not equal to the extreme demand that is made on it.

A reason frequently given for advance manufacturing is the desire to prevent skilled workers drifting away when the work is slack. In a seasonal industry a plant must either be large enough to take care of a heavy peak load or the load must be reduced by advance manufacture, thus making possible a fuller utilization of the capacities of a plant that is also smaller. These are two powerful arguments in favor of the economy of manufacturing in advance of orders.

There are many considerations that operate against manufacturing in advance of the season. When the product is bulky and costly, the carrying charges of holding inventories may be greater than the losses due to irregularity of production. A risk to be greatly feared is that a change in style or a mechanical improvement introduced by a competitor may render one's own product obsolete and salable only at a loss of profit.

Many products cannot be held in stock because they deteriorate. Certain kinds of food product are an example.

A concern in a staple industry runs the least danger when it undertakes advance manufacture. Sometimes a company making a style product is in a position to carry advance manufacture to a certain stage, leaving the finishing touches to be added when the season's style trend has made itself known. In most lines there are some items of staple merchandise which can be made up in advance of the season. Simplifying the line so as to eliminate slow selling items reduces the risk of advance manufacturing. Converting special orders into stock items is one device frequently employed that makes manufacturing to stock more practical.

Many concerns now maintain in-stock departments so that they may promptly fill customers' reorders. The additional storage space afforded by these in-stock departments makes possible advance manufacturing. Furthermore, the need to have an ample supply of goods to meet the retailers' current demands, even in the dull season, makes necessary a program of building up of stock inventories.

In a seasonal business inventories cannot be carried over until the next year without incurring heavy expense. Inventories which have been piled up in excess of the seasonal demand which finally makes itself felt frequently have to be "dumped" at a loss when the season is ending. When many concerns in the industry pile up inventories in a dull season, and general business takes a turn for the worse, the whole industry can easily find itself in a state of serious overproduction. On the other hand, if a factory produces a commodity that is not subject to deterioration or to

style obsolescence and that enjoys a unique reputation by which it is virtually insured of ultimate demand, it stands in little danger from overproducing beyond current demand. The only penalty it must pay if there is a serious business slump is the expense of holding inventories unduly long.

In evaluating the policy of manufacturing in advance of orders as a means of regularizing seasonal production, it must never be forgotten that the term is entirely relative. In the instance given a little piling up of inventory may be a highly desirable move, but an excessive piling up of inventory may be disastrous.

Unemployment—Causes, Palliatives, and Preventives

By DR. ROYAL MEEKER,[6] NEW HAVEN, CONN.

The causes of unemployment are manifold, but come under the single head of maladjustment between production and consumption. The chief contributing causes are: Lack of planning of production to meet and not to exceed consumption requirements. Faulty distribution and utilization of the national income, causing alternating periods of over-saving and over-investment in production or intermediate goods, accompanied by under-spending and under-consumption of immediate or final consumption goods, followed by over-spending and under-saving. Wide variations in the general price level, which mean changes in the purchasing power of money. Unforeseen calamities, including marked overproduction of some commodities, as well as destruction of lives and goods by earthquakes, floods, storms, droughts, famines, etc. Technological improvements due to new machines, methods, management, and organization.

Until recently little was known about the underlying causes of unemployment, and that little was known by very few. Hence the methods of treating this social-industrial disease, or accident, are for the most part merely palliatives, not cures or preventives. The palliatives which are most invoked to help us out of depression are: Soup and bread lines; odd jobs; "buy it now" slogan; wage maintenance; public works; regularization of individual plants; unemployment reserves in individual plants. The only cure for unemployment is employment. There are, however, three major policies which may be inaugurated at once for the amelioration and prevention of unemployment: Planning a consumption-production program in all industries for several years in the future. Stabilizing the general price level, through credit and monetary control. Providing carefully planned, all-inclusive unemployment insurance. To make a start on any of the measures proposed requires a degree of knowledge, of cooperative spirit, and of authority which are at present largely lacking. But the evil of continually recurring unemployment demands bold and drastic action.

IT IS true that but little progress has been made in the direction of creating systematic means for either preventing unemployment or caring for the unemployed. Our backwardness in this, the most important aspect of present-day life, comes from our overemphasized individualism and our failure to understand the nature and the causes of most unemployment.

Unemployment, seasonal, cyclical, or continuing, is the most serious defect in the present-day economic organization. The job comes first. Before the intricate and innumerable problems of wage rates, hours, division of labor, rate of production, conditions of labor, etc., can come into existence, there must be jobs and workers doing these jobs.

Unemployment means involuntary idleness from whatever cause, and includes besides the jobless all those who for any reason are employed less than full time. The unemployed includes, then, all jobless workers, all part-time and irregular workers, and all workers who for any reason are not fully employed while actually on the job—those unemployed on the job, so to speak. Unemployment concerns only those able and willing to work. It has nothing to do with the chronically unemployable and those who do not or will not work.

Causes of Unemployment

A large number of names have been used to explain unemployment, the most hackneyed being, "overproduction," "underconsumption," "underproduction," "changes in the price level," and "technological improvements." All these causes can be included under one head—namely, a lack of balance between production and consumption. Five causes contribute to this condition:

1 Lack of planning of production to meet and not to exceed consumption requirements
2 Faulty distribution and utilization of the national income, causing alternating periods of over-saving and over-investment in production or intermediate goods, accompanied by under-spending and under-consumption of immediate or final consumption goods, followed by over-spending and under-saving
3 Wide variations in the general price level, which means changes in the purchasing power of money
4 Unforeseen calamities, including marked overproduction of some commodities, as well as destruction of lives and

[6] Associate in Economic Research of Prof. Irving Fisher. Dr. Meeker was United States Commissioner of Labor Statistics from 1913 to 1920, when he took charge of the Division of Research of the International Labor Office. In 1923 he headed the Department of Labor and Industry of Pennsylvania. In 1924 he went to China on the Commission of Social Research in China. Since his return in 1925, he has taught for a year and a half at Carleton College, Northfield, Minn., and is now engaged in economic research.

Contributed by the Management Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

goods by earthquakes, floods, storms, droughts, famines, etc.

5 Technological improvements due to new machines, methods, management, and organization.

Until recently very little was known about the underlying causes of unemployment, and that little was known by very few. Hence the methods of treating this social-industrial disease, or accident, are for the most part merely palliatives, not cures or preventives.

Palliatives for Unemployment

The first and favorite palliative is the time-honored and dishonorable institution of the soup kitchen and the bread line. Every thoughtful American must feel angry and ashamed that, after passing through ten major business crises and some two dozen less severe business slumps during the past century and a half, we have no effective plan to deal with these emergencies, but still rely mainly upon voluntary haphazard doles to keep the victims of calamity from starving and freezing. The one unanswerable argument in favor of these doles is the fact that even the hard bitter bread and thin soup of charity do keep men from dying of starvation. The Roman emperors did as much. They dispensed free bread and circuses to the jobless proletariat of ancient Rome; we dispense free bread and soup. Whether or not we have improved upon the Roman method seems to depend chiefly upon the relative merits of the free Roman circus and free American soup!

So long as present conditions prevail in America, both the unemployed employee and the unemploying employer will continue to be the unwilling or the unwitting recipients of our doles of bread and soup. The employer whose discharged workers are kept by public or private charity during depressions receives alms indirectly but just as truly as do those who eat free bread, drink free soup, and sleep in free "flop houses." The great American dole is a very poor palliative, indeed, and it is our shame that it still remains our main reliance in every industrial depression.

Next, there is the odd-job palliative. This has less demerit than free bread and soup, but it has not very much merit. Reconditioning and rebuilding plants are not odd jobs. The odd job is merely one means of keeping workers from starvation or complete pauperization during an industrial depression. Again, so long as industrial and political leaders do not tackle the causes of unemployment, the odd job with all its inadequacies will continue as a method of dealing with unemployment.

Within the past few months the "buy it now" palliative has emerged. Merchants have discovered the power of "buying power" and are urging consumers to buy now whether or not they really need things or can pay for them. This palliative at least recognizes the need for buyers to absorb surplus goods, but its merit is slight indeed. The great danger is that large numbers of people may be persuaded to acquire things more or less useless by promising payments which they can never meet.

Wage maintenance is another palliative. This palliative has decidedly limited applicability. The employer cannot pay wages unless he has money with which to pay. Employers who maintain rates and employment out of surpluses accumulated during prosperous years are really practicing a form of unemployment insurance. This form of spreading industrial earnings in fat years over the lean years may be good or bad according to the plan used and the conditions. Most employers have no surpluses left when the lean years come and can maintain wage rates in the face of dwindling income only by cutting down labor costs through substituting machines and methods for men and by discharging, laying off, or giving partial employment to workers. An employer may maintain the rates, but he cannot maintain the earnings of his employees. Buying power of workers is cut down as much or more than it would be by allowing wage rates to decline along with other prices. The sole merit of this practice, if it be a merit, is in keeping a high wage standard against the time of business revival.

The palliative of public works has certain merits, but its function has been greatly exaggerated, not to say misrepresented. Its applicability is very limited. Money or credit taken by taxation or by loans from the people for public works leaves just that much less for private industry. It is not asserted that public works are no good. It is possible to work out a program of public construction extending over 10 or 20 years, although no such program has ever been planned anywhere. Such a program might be speeded up to tide over a severe emergency situation, but at best such work can be helpful for only a very short time. Private initiative and enterprise has in times past been less tardy and more effective than public efforts to aid recovery. However, hearty support should be given to the present public-works program as an emergency measure.

Regularization of Industry

The regularization plans of private corporations and plants deserve hearty commendation and their extension to other plants and industries should be encouraged. They are most interesting and valuable, but they are as yet too limited in scope to help much toward the solution of the major problems of unemployment. No single employer, or group of employers in one industry, or even in several related industries, can effectively regularize production and employment in the one plant or the group of plants included under the plan. To be effective regularization must include all industries in the country and eventually in the world. Encouraging beginnings have been made on a national scale in this country by employer associations similar to the Citrus Fruit Growers Association. International cooperation has been attempted by the several international cartels like the Steel Cartel, the Copper Syndicate, and the proposed Sugar Agreement. The limitations and weaknesses of the existing employer agreements and associations, international as well as national, are too patent to require comment. Before any considerable progress can be made in this country toward extending and making effective an industrial regularization plan, the anti-trust laws must be drastically reconstructed.

The limitations which apply to the individual plant and to groups of plants apply to cities and communities. All these plans by individual employers, by employer groups, by industries, and by communities to regularize production are all to the good. They show that public-spirited employers and public officials are on the right track in tackling the chief cause of unemployment. These plans call for hearty commendation and support from all who desire to attack the causes of unemployment. These efforts are classed at present among the palliatives because of their limited scope and extent.

Unemployment funds set up by individual employers and groups of employers to tide over periods of depressions are described in the report of the Industrial Relations Counselors. But these unemployment funds are quite inadequate to solve unemployment. They are likely to go on the rocks during serious depression because the regularizing of employment is too big for a few individual employers or even groups of employers to accomplish. The plan deserves the heartiest commendation, however, as a first application on a limited scale of the insurance principle to the care, cure, and prevention of unemployment.

Preventives of Unemployment

The only cure for unemployment is employment. This sounds like saying the cure for illness is health; the way to be good is to avoid doing wrong.

There are, however, three major policies which may be inaugurated at once for the amelioration and prevention of unemployment:

1 Planning a consumption-production program in all industries for several years in the future so as to minimize or eliminate overproduction or underconsumption. This should include a more equitable distribution of the products of industry; that is, the national money income
2 Stabilizing the general price level; that is, the buying power of money
3 Providing carefully planned, all-inclusive unemployment insurance to prevent unemployment, as well as to care for the unemployed.

To make a start on any of the measures proposed requires a degree of knowledge, of cooperative spirit, and of authority which are at present largely lacking. If the main framework of the existing economic order is to be preserved, these deficiences must be supplied. If it is difficult to make a beginning, it is even more difficult to carry through to a successful finish. But the evil of continually recurring unemployment demands bold and drastic action. Nothing worth doing is easy.

A Planned Consumption-Production Program

The measure of first importance is naturally the most difficult of all. All that can be attempted here is to emphasize the necessity of directing and controlling our work toward the end of gratifying our wants most effectively. The working out of the complete program will require years of careful research and experimentation.

A beginning, however small, has been made in the planning of production for consumption. These small beginnings should be supported and expanded as rapidly as possible to include all industries throughout the nation, and eventually the nations trading with us. This program is of tremendous difficulty and magnitude, but it is indispensable to the realization of regularization in business and employment. It calls for all the genius of organization and statesmanship among our industrial, financial, and political leaders. It demands the fullest understanding between employers' and employees' organizations, the public, and their representatives. The very difficulties in the way should stimulate our leaders to overcome them.

Stabilization of Money

The reasonable stabilization of the price level is indispensable to the reasonable stabilization of business. Here again there must be clear understanding and complete cooperation between private financial organizations and the public agencies. More progress has been made in theory and in practical application in this field than in either of the others. Numerous books and numberless articles have been written on the subject of stabilizing the price level. While it cannot be proved statistically that price fluctuations precede and cause business fluctuations, nevertheless price changes do affect business profoundly. A consumption-production program would be very difficult or impossible to carry into effect without reasonably stable prices. A good beginning in monetary control has been made by the Federal Reserve System, but it is only a beginning, and there is serious danger that the feeble efforts already made may be repudiated by Congress unless business men act vigorously to support and strengthen the system. The completion of the task will be difficult, but it can and must be done, if socialistic and communistic experiments are to be avoided.

Unemployment Insurance, Preventive and Cure

Unemployment insurance should be enacted primarily to prevent unemployment and to take care of the unemployed. To avoid confusion fundamental terms must be made clear.

First, unemployment insurance is a means of distributing the losses due to business crises and depressions among all buyers of commodities, instead of letting them fall with smashing force upon unlucky individual employers and employees as at present.

Second, the fundamental principle of insurance is the distribution of losses among a large body of risk takers. Losses so distributed would cost the employers and insurance carriers nothing, and the consumers who ultimately would pay the bill could detect no difference in prices, so slight would be the portion of expense falling on any one commodity.

Third, the notion that the whole cost of unemployment insurance will be an *added cost* upon production is absurd. This theory, which seems to be accepted by most employers without question, assumes that unemployment now costs nothing because no system exists to care for the unemployed. A national employers' organization has published a statement declaring that unemployment insurance would cost each year from $195,000,000 to $334,000,000. Possibly the costs would total those sums if *unemployment were not reduced.* The total cost of insurance is of no importance. The difference between the cost of unemployment under the insurance plan and under no plan at all is of vital importance. We must quit deluding ourselves with the foolish fancy that unemployment does not cost us anything so long as we do not recognize it. Make no mistake! We are paying for unemployment and paying through the nose—in taxes for maintaining our extravagant poorhouses, our ever-present street beggars, our sporadic and prodigal outbursts of charity to keep the unemployed from starving and freezing, the colossal costs of inefficiency and ca'canny directly ascribable to unemployment and the fear of losing the job. Even if unemployment insurance should add $500,000,000 to the price of all commodities, the whole nation, including employers, would be the gainers. The total drain upon the national income would be substantially less than at present, but the greatest gain would be in the improved morale of workers freed from the fear of being thrown out of a job.

Fourth, the unemployed eligible for insurance would consist of those alone who lose their jobs through business hazards and not through the workers' own fault. Unemployment is really an industrial accident or disease. In fact, it is the only truly industrial disability. One can fall ill or fall and break a leg at home as easily as at work. One is more likely to get killed or maimed by an auto on the highway than by an operation on the job. But the only way one can become unemployed is to lose his job. Unemployment is just as insurable as illness or traumatic disability.

Unemployment insurance is not intended to pay workers for not working. Its most important effect is prevention. Unemployment insurance will unquestionably act like workmen's accident compensation insurance to prevent the disability insured against.

The Complete Program of Prevention

It is no cause for regret that this country has not thus far resorted to unemployment insurance as a cure-all for unemployment. Insurance should be utilized only as one part of a completely rounded program to combat booms accompanied by hectic over-employment followed by depressions with under-employment and unemployment.

Employment assurance is being advocated by many American employers as an alternative superior to *unemployment insurance* which has been so widely adopted in Europe. It may be said that *job assurance* is but another name for *job insurance.* There is much truth in this statement, but regularization of employment may be practiced without unemployment insurance, and

unemployment insurance may be applied so as to have little or no effect in smoothing out booms and depressions.

Employment assurance and unemployment insurance are the two halves of a complete employment program. Either one alone will fail to take care of fluctuations in business and employment. Both together will achieve a much desired stabilization—economic, social, and political.

When consumption-production is equalized, the value of money stabilized, and unemployment minimized by putting the costs where they belong, we may reasonably expect unemployment to be so reduced that the actual payments by insurance carriers to the unemployed will be much less than the lowest estimate of costs made by the National Association of Manufacturers.

When employers and the rest of the public realize fully that unemployment is what President Hoover has described it to be, the greatest waste in industry, and that every partly or wholly unemployed person must be supported out of the national income produced by those employed in industry, they will take the necessary preventive measures against this greatest evil in the modern economic system.

Employment Stabilization in the Procter & Gamble Company

By HARVEY C. KNOWLES,[7] IVORYDALE, OHIO

The history of one company in stabilizing employment is detailed. Having but three presidents over a period of 93 years, a definite family policy has been carried out, resulting in a plan that virtually guarantees an employee that he shall have work for 48 weeks during the year; that he shall have a share in the profits; that in case of sickness he shall receive two-thirds of his pay for a year and a pension in case of disability; that he may participate in a home-buying plan.

EMPLOYMENT STABILIZATION is a subject much discussed, and there have been thousands of pages written about it. Methods and systems without number have been studied and tried out. It has received the sincere and detailed thought of educated men. Plans, most of which are quite complicated, have been worked out by the score. The method used by us, we believe, is complete and yet very simple. This is all it is: "We guarantee you full pay for full-time work for not less than 48 weeks."

This guarantee of regular employment reads as follows:

To employees located at Ivorydale, Ohio, factory, and participating in the Procter & Gamble profit sharing plan, the company hereby guarantees full pay for full-time work for not less than 48 weeks in each calendar year, less only time lost by reason of the customary holiday closings, or through fire, flood, strike, or other extreme emergency, and subject to the following provisions:

(*a*) When an employee enters the profit sharing plan after January 1 of any calendar year, the company guarantees to him under the terms and provisions outlined herein, full pay for such of the 48 weeks as may remain in that calendar year.

(*b*) The company reserves the right under this guarantee to transfer any employee to work other than that at which he is regularly employed, and to compensate him for same in accordance with the following terms and provisions:

(1) If the transfer is the result of a full or partial shutdown, or a temporary cessation of work in the department in which he is regularly employed, or as emergency in the department to which he is transferred, it shall be regarded as temporary, and the employee shall receive, during such period of transfer, his regular wage rate per hour.

(2) If the cause of the transfer is the permanent termination of the particular department or the particular job where he has been employed, or where the transfer is in the nature of a demotion for any cause, it shall be officially recorded as a permanent transfer, and the employee shall receive the wage rate which prevails for the work to which he has been transferred.

(*c*) The company reserves the right to discharge any employee at any time and for any cause, and further reserves the right to terminate or modify this guarantee in whole or in part at any time after serving six months' notice to that effect.

By this guarantee the employee is assured of regular work and regular pay, and he will also receive quarterly profit sharing dividends on his wages. In addition, he is accumulating with but little effort on his part an investment from which he will receive the regular dividends paid on the Procter & Gamble Company's common stock.

The whole plan of the Procter & Gamble Company has a background covering a couple of generations. From 1837 to 1930 the company has had only three presidents—grandfather, father, and son in 93 years.

Mr. William Cooper Procter four years out of college, when he was 25, went to his father with the thought that it was right for the factory workers to gain some part of the company's earnings. Such a plan was put into effect, and in 1887 Procter & Gamble employees drew profit-sharing dividends, paid in cash. As near as can be told today, the ideas and ideals forming in the young Mr. Procter's mind at that time were something like this:

You have a better and stronger company; you have better citizens in your factory if ways and means are worked out to make it possible for your employees to become part of the company and to become independent and free from worry. There is no thought of dealing in the home affairs; there is no thought of charity; there is, though, a way to meet the conditions feared by labor.

Someone has summed up the needs of the working man as follows:

He must be taken care of when an accident happens.

He must be taken care of when sick from any reason.

He must have a way of purchasing a home—a way of being sure to be able to pay for that home.

He must have a method to build up an independence so that there is no fear of old age.

Mr. Procter has worked out plans in the last forty-odd years that take care of all of these points. These points can be covered as follows:

Accidents. This has been recognized by the states and is well taken care of.

Sickness. Our present plan takes care of a man who is sick from other than industrial accidents, by paying him 66⅔ per cent of his salary over 12 months, and then providing a dis-

---

[7] General Superintendent, Procter & Gamble Company.

Contributed by the Management Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

ability pension in case of continued sickness. The fund for this plan is built up by the employee contributing $10 per $1000 of payroll, and the Procter & Gamble Company contributing $20 per $1000 of payroll. This plan has been in effect long enough now so that several actuary studies have been made and we feel it is quite sound.

Home Buying. The company has a home-building plan by which money is advanced on homes selected by employees. The assurance of paying for the home comes through good wages and steady work, together with profit sharing. Our profit-sharing plan is as follows:

An employee must be in the service of the company for six months before being eligible for profit sharing. There are numerous reasons for this. First, there are a certain number who are always moving from one job to another. Second, we want the employee to be sure that he likes us, and we want to be sure that we like him. After this six-month period, and upon recommendation of his foreman, a number of shares of the Procter & Gamble Company stock is bought in the open market, the value of which comes closest to the employee's yearly salary. We guarantee this price against decline on the first block of stock. The employee must pay in 5 per cent toward the redemption of the stock, and the company pays in 12½ per cent. Interest of 4 per cent is charged on the unpaid balance, and dividends are credited. This means that the employee will get his stock in about six years. If he leaves prior to a five-year period he gets back the money he paid in, plus 6 per cent interest. Under this plan there are several men who today have over $100,000 worth of Procter & Gamble stock and who have never earned over $35 per week. As a part of profit sharing, the employee comes under the guarantee of employment plan.

It is our belief that what people want is to become independent through their own efforts and not have to rely on charity in old age. Under the plan a man may retire from 50 years of age on, with an income somewhat larger than his wages. He builds his own security.

The method used in establishing the guarantee of a full-time job for 48 weeks was worked out about as follows:

Shortly after the war, Mr. Procter said: "Are we not missing the biggest point in our manufacturing program when production is not controlled to flow evenly? The consumer uses our products at an even rate; the grocery store sells to them at an even rate; why should not our business be operated at an even rate?" The first work was for the sales department to estimate the business of the next five years.

Each salesman covers a population of approximately 200,000, or 50,000 families. He knows what he has sold in that section, and he estimates what he can do in the way of bettering his condition in the future. We have then a real estimate made up by the men who sell the goods. This five-year estimate was then gone over with the manufacturing department. The salesman takes the first responsibility, and before the schedule is put into effect every person in the company takes his responsibility in seeing that this guarantee of employment is carried out. The plant went into effect August, 1923. As a matter of interest, the 1926 production schedule was within 0.12 per cent of that schedule made out in 1922 for 1926, and the only variation was in slightly larger sales than estimated.

Our present method is about the same. Each fall of the year the sales department estimates the coming year. If our equipment is close on certain brands, we will ask them for a three- or five-year estimate, but the year schedule stands for the next calendar year. We guarantee 48 weeks; we divide the schedule by 49 weeks, and our experience has been that we give about 50 weeks. There are seven recognized holidays throughout the year, so that a little over a week is lost that way. The month is worked out by dividing the week by 5½ and multiplying by the number of working days in the month. When this yearly schedule has been agreed to by the sales manager and the manufacturing department, it is approved by the administrative committee of the company and becomes fixed.

It is agreed that on major brands the sales department will hold their monthly schedule within 10 per cent; this is only by brands, and the total of the factory must remain the same. Here are a few figures on how this has worked out:

On one brand of goods the schedule was 1,463,000; the goods produced was 1,447,000. On another brand the schedule was 389,000; we produced 399,000. On another brand the schedule was 349,000; we produced 343,000. On a whole group of products there was a schedule of 12,096,000 for the year, and we produced 12,173,000, or six-tenths of 1 per cent over the schedule. Now this is not at all easy to do. It means that every one in the organization must accept his responsibility.

When conditions such as for the last twelve months have called for increases in some particular brands and decreases in others, the matter is talked over between the sales and the manufacturing departments, carrying it right down to the distribution of labor.

This guarantee of employment has cost us a little money at times. Due to the lack of buying last spring, it was necessary to stock quite a lot of soap. On the other hand, it has had many fine sides in addition to providing steady employment. For instance, when we built our Baltimore plant, which started operations in July, and when we planned the Long Beach, Calif., plant, which will start next July, we not only have a well-thought-out schedule for these plants, but we plan our manufacturing on the basis of the plants producing 50 weeks a year, and the equipment is not much more than that. Prior to starting up the Baltimore plant, for a period of three years it is doubtful if we had a capacity of 10 per cent in excess of our average production. It has meant that all the equipment is going and all building space is used; both are naturally maintained because of the necessity of keeping them in operation, and everyone knows that the way to have good equipment is to run it.

We have not yet applied guarantee employment to the cottonseed mills, but we have been able to double the time of operation of the mills and make it perfectly possible to run the year around, if the seed could be bought. Mr. Procter took active charge of the mills four years ago, and it was not very long before tanks had been erected for storage and ways and means for preserving cotton seed had been perfected.

We do not know how this guarantee would apply to other businesses, but we do know that there are many ends of our business that ten years ago we would have said were impossible to stabilize. Today it seems quite logical. There are many instances where certain by-products are seasonable, and we have trouble disposing of them evenly, but we have made arrangements to fill in with another product during that time.

The number of people affected by the profit-sharing plan will be of interest. To be eligible one must have been with the company six months and be either an hourly worker or a weekly one getting $2000 a year or less. Of those eligible over 95 per cent are profit sharers and come under the guarantee.

The main thought is this: Ten years ago our business was quite seasonable. There were periods when we would do almost twice as much business as other periods. Every one naturally fell into his or her responsibility and has helped work out a definite plan.

If we have not obtained the best type of plan, we have at least found out that when one has set a goal ahead that is right and is determined and willing to put in the time and effort, it just becomes natural to find a solution.

Discussion on Stabilization of Employment

REPORTED BY CHARLES W. LYTLE,[8] NEW YORK, N. Y.

IN inaugurating the discussion, Chairman Ralph E. Flanders presented Dr. Lillian M. Gilbreth,[9] who spoke from her experience as the women's representative on Mr. Hoover's Unemployment Committee. Mrs. Gilbreth first told something of the personnel of the committee. She then pointed out that those present represented types who still held positions and that there was much they could do to support the work of the regular committee. While she praised the measures for long-run stabilization, she also stressed the importance of immediate relief measures. One of Col. Arthur M. Woods' first slogans, she said, was "Spruce up your home." Finally, she stressed the importance of maintaining American standards of living and present wage rates, saying: "I think, from the President all the way down to the committee, we all feel that reduction in wages and in living standards would be absolutely fatal."

The general discussion centered largely around Mr. Knowles' paper and continued at a high pitch of interest. The Chairman commented on the great responsibility of a management which undertook such a program and asked Mr. Knowles if the salesmen did not feel like quitting when they received their quotas. Mr. Knowles answered that, on the contrary, every one knew just what he was expected to do and that there was a clearer understanding of responsibility all the way through. In answering questions regarding the security of markets for the soap business, Mr. Knowles pointed out that the general run of soap constituted only half of the business and that the making of fats such as Crisco constituted the other half. The first half had recently fallen off 10 to 15 per cent, but as a whole the business for the present year would be a little larger than for the last year. A factor which had facilitated sales was replacement of a certain amount of the jobbing business by chain stores.

To illustrate that the soap business has not always been as stable as at present, Mr. Knowles cited the cotton-seed end of the business. Ten years ago 90 per cent of the cotton seed used was crushed by the middle of January. Mills then shut down for the rest of the year. The reason for this was that they did not know how to carry cotton seed over without its deterioration. At present Procter & Gamble has three cotton-seed mills operating all the year round, crushing a total of 157,000 tons per year, their other mills operating up to April 15. Even here it is expected to bring about greater stabilization. Referring to the fact that some money had been lost in order to work out the ideal, Mr. Knowles asked, "Is it not logical and necessary to lose money in advertising and experimenting when you have faith that your plans are right?" M. F. Simmons[10] pointed out that there was naturally a seasonal factor in the marketing of the electric lamp globes, but that his company had already announced a guaranteed employment plan in all lamp divisions and hoped to do the same in the turbine divisions.

The Chairman called attention to the fact that the Procter & Gamble Company and the General Electric Company represent two radically different types of management and yet they seemed to be in substantial agreement. "This ought to give some of the rest of us a little more courage."

Mr. Knowles continued by saying that their sales department was bound not to increase the yearly schedule until such schedule could be maintained. This puts a premium on making the sales continuous and does not allow coverage of poor sales by a temporary excess. Some one asking if Procter & Gamble made their purchases of equipment steadily, the answer was that purchases were reasonably steady. In fact, this is facilitated by the knowledge of how much volume is to be produced in advance.

Another machine-shop man recalled that years ago F. W. Taylor demonstrated certain principles in the machine industry which were reluctantly received by other industries on the grounds that they were not applicable. He then raised the question if such other industries were not now demonstrating new principles which in turn were being reluctantly received by the machine industries for identically the same reason.

The Chairman then called upon L. W. Wallace.[11] The latter expressed the opinion that Dr. Meeker had painted a picture which was entirely too pessimistic. "I think that individual industries and leaders of industry must begin to educate themselves and try out various methods in their own plants." He quoted the motto applied to the engineer at the 50th Anniversary of the A.S.M.E., "What is not yet, may be." "What leaders of industry need today is to carry that sentiment in their minds constantly." To the machine-tool people, he said, "May I suggest that you go back to your respective plants and keep engraved upon your hearts that statement. If you will start to work with that as a foundation, I believe you will be much surprised to find how quickly you can do something equivalent to what has been done by Procter & Gamble, Dennison Manufacturing Company, and others. Until we adopt that kind of a program we will confront crises similar to this 10, 15, and even 30 years hence. The encouraging thing is that people are beginning to think and act in a new direction. They are building a program which will eventually even out the peaks and valleys of the business cycle."

Referring to what the engineering profession could do in the present crisis, Mr. Wallace continued: "I realize that there is no group in the United States that can be more depended upon in the call to solve national emergency problems than the engineering profession." Here he explained the plan of Colonel Woods to make a canvass throughout local sections for the purpose of ascertaining what industry has done to stabilize employment.

Again taking issue with Dr. Meeker, he objected to the inclusion of public works as a panacea. He said: "Intermittently since 1882, the engineering profession, in cooperation with other organizations, has advocated a Federal Department of Public Works. Such a step would bring about great economy of operation." In the several bills before Congress, a planning board is proposed, the function of which would be to lay out long-time plans for public-works activities for the Federal Government. These plans would be projected forward from 10 to 15 years and comprise designs and estimates making possible scientifically accurate appropriations.

"It is proposed to set aside reserves so that funds will be available and plans ready when needed for execution. Had there been such a set-up in the Federal Government, it would have been unnecessary for the President to form a Cabinet committee or for Colonel Woods to organize the public-works section of his committee."

[8] Director of Industrial Cooperation, New York University; Chairman, Management Division, A.S.M.E., 1930.

[9] President, Gilbreth, Inc., Montclair, N. J. Mem. A.S.M.E.

[10] Assistant to Vice-President in Charge of Manufacturing, General Electric Company, Schenectady, N. Y.

[11] Executive Secretary, American Engineering Council, Washington, D. C. Mem. A.S.M.E.

In conclusion he expressed the belief that the findings of Colonel Wood's committee would inspire many activities which would tend to lessen the frequency and severity of depressions.

The Chairman then asked R. E. W. Harrison[12] to describe the activities which Cincinnati has had under way for several years. Mr. Harrison paid tribute to the public-spirited employers in that city and to the City Manager, and also gave credit to Fred K. Hoehler, Director of Public Welfare. Mr. Harrison pointed out that the Cincinnati movement was started at a time of maximum prosperity, purely on the grounds of preparedness for recessions which were fully expected. He said that although Cincinnati was best known for its machine-tool industry, there was a wide diversity of industry. He agreed that it was good business for the employer to retain the skilled working force and to keep labor turnover at a minimum. Mr. Harrison then read a discussion by Mr. Fred K. Hoehler.[13]

The Chairman then introduced William J. Pedrick,[14] representing former Governor Alfred Smith, of the Committee on Employment Relief for New York City. Mr. Pedrick assured of the committee's cooperation.

The Chairman brought up the subject of maintaining wage rates. He had no hesitancy in saying that he was one of a large number of manufacturers who felt that it was advisable to maintain rates simply from the standpoint of maintaining the rate of consumption. He then asked Dr. Meeker to lead the discussion. Dr. Meeker outlined the economist's point of view on this question somewhat as follows:

In particular establishments, reserves may have been accumulated which can be used to contributed toward wage-rate maintenance, regardless of bad times. Throughout industry as a whole, he did not think such maintenance desirable. According to him it would mean that those employers who lacked sufficient reserves would, when faced with price reduction, be obliged to cut the labor cost of production. An attempt to maintain wage rates would intensify the substitution of machines for men and perhaps necessitate shutting down of the plant part or all of the time. In fact, he gave the opinion that the purchasing power of workers throughout the country is lower today than if wage rates had taken the older course downward. He believed that if this had happened there would now be much more employment and that the purchasing power of employees would be higher.

W. W. Nichols[15] dissented from this point of view and told about the findings of Whiting Williams,[16] who had repeatedly worked as an unskilled laborer in order to ascertain just what was on the worker's mind and had now completed another of these adventures at the request of Mr. Hoover. One of Mr. Williams' observations was that he could not at first understand the apparent serenity that prevailed throughout the country and that the laboring class suffered so much unemployment calmly. For instance, there were not the crowds at factory gates as in former depressions. In answer to this he found that there was an appreciation on the part of the employee of the necessity of the layoff and a belief that it would be only temporary. The employee seemed to retain his interest in his former job, with full expectation of getting it back shortly. He had, as Mr. Williams expressed it, "a vested interest in his job" that he had never felt before. Mr. Nichols said that it is this state of mind that has prevented for the first time the appearance of strikes and great unrest. In other words, there is an acceptance by the laboring element of the conditions that prevail and a feeling that both employer and employee are in the same boat. "When the change does come, both will benefit thereby."

Referring to Dr. Meeker's interpretation, Mr. Nichols said: "I want to refute the remarks that Dr. Meeker makes about reduction of wage rates. Recently I attended a meeting of a most representative group of industrial employers, at which this subject was thrashed out. While there was some difference of opinion at the beginning of the discussion, at the close there was no question as to the concensus of opinion in favor of maintaining wage rates. These men had all had long experience in dealing with such difficulties. They felt generally that they must recognize the psychological attitude of labor which interprets a reduction in rates as a thing to be resented. This naturally involves a loss of confidence. The worker today is willing to accept a reduction of hours, but the moment that a reduction of rates is proposed the opposition is strong."

Dr. Meeker insisted that economically the destination was the same. He maintained that it mattered little to the employee what the rate was if the total earning per week was less. In fact, it was his experience that workers were quite willing to accept cuts in rates on the ground that "a half loaf is better than no loaf."

Walter F. Dixon, General Manager of the Singer Manufacturing Company, said that his company had a 75-year record for keeping a reasonably steady gait in good times and bad. "We had reason to realize early that the backbone of the company, the manufacturing plants, was made up of the skilled, loyal, faithful employees. Our product is one requiring a high grade of material, but even more than that, extremely accurate, minutely measured work. One could get some idea of what the work amounts to in our product if it is told that the cost of material in one of our sewing machines is about 13 per cent of the entire cost; that is to say, between 13 per cent and 100 per cent is laid out for labor and overhead charges. As to the plan of the Procter & Gamble Co., with the legal promise they have made to all of their employees within the last two years guaranteeing them constant employment for fifty weeks of the year, we have never considered any legal agreement, probably because we thought the time might come and possibly would come when we would be unable to fulfil our legal promise to the employees; but we have had a feeling which has grown up in the course of years that a man who wants to work with us and has been an employee for any length of time is guaranteed employment, although full or part time we do not say. During the last year of depression we felt the effect of it a little later than every one else, and similarly we pick up a little after every one else. Fortunately for us, whether things are good or not, people have to wear clothing and most of them shirts and things of that sort, so that commencing a little sooner, when we began to feel the full effects of depression, we put our shop, instead of on a five-day nine-hour per day basis, on a four-day eight-hour basis; that is to say, we started working thirty-two hours a week instead of forty-five hours. We did not, however, decrease the piecework rates or the time rates paid to those that are not on piecework. We believe the psychical effect of not reducing rates but reducing time is a very important one; that the employee will look forward to the time when things will be better with no occasion of having to fight the rates back to what they were. They know that when things improve and the factory goes on longer hours their incomes will be greater because their daily rates have not been decreased. Of course every office manager having several thousand people under his jurisdiction hears all sorts of reports as to how things are being taken in the shops. We do not have detectives to report on what the men are saying, but many of the men voluntarily report either to me or the superintendent what the general feeling about business is. I can say

[12] Engineering Director, Cincinnati Grinders, Inc., Cincinnati, Ohio; representing Cincinnati Committee on Stabilization. Mem. A.S.M.E.

[13] This is presented in full at the end of the discussion.

[14] New York, N. Y.

[15] Assistant to President, Allis Chalmers Mfg. Co., New York, N. Y. Mem. A.S.M.E.

[16] Author and consultant, Cleveland, Ohio.

that the feeling is distinctly good at this time. We employ about 14 per cent of our total employees in mills, and the company is probably doing the best it can for them.

"We make all forms of machines. We make the ordinary family machine and we manufacture the so-called factory machine, which is very much more complicated and costs a great deal more to build. We call our simplest type of family machine, 'Unity.' One complicated machine may be equal to six of the Unity machines, so in watching our stock very closely, if we find that the Unity machines are piling up a little too fast, we switch the men over to the complicated machines. Of the various terms to represent this present condition, I do not know whether it is overproduction or underconsumption, but I do think that those of us who have done export business agree that in the conditions in China and the conditions in Russia, it has been underconsumption in the one case, due to chaotic conditions existing in the country, and in the other case conditions of a distinctly different kind. Of course it is a question which each manufacturer must settle for himself as to what extent we are justified in sending over American engineers and American experts of every description, including our agriculturists, in order to assist them in bringing about ultimate competition with ourselves and every other civilized country."

The Chairman introduced John A. Garvey,[13] who outlined the policy of Dennison Manufacturing Company on employment.

At this point the Chairman called on Harrington Emerson[17] to express his views. Mr. Emerson told something of his experiences abroad during the last four years, in which he visited many foreign countries, including Austria, Russia, China, and Japan.

John M. Carmody[18] pointed out that industry as a whole possessed too much obsolete machinery. "In some cases our inventories were heavier last June than they were a year ago in February or October. During the past two years I have visited perhaps 800 plants. I have said to them that we ought to take up this question of obsolescence. I think that question affects not only the great machine industry, but every great industrial manufacturer in this country, and there ought to be a concerted drive toward the elimination of obsolete plants."

Dr. Max Toltz[19] asked Dr. Meeker to explain how the various groups, such as economists, engineers, and politicians, could cooperate toward the relief of unemployment. Dr. Meeker replied that we are all politicians, and furthermore we are all either employees or employers. "It is up to us to solve that extremely difficult problem of keeping a balance between production and consumption. It is not confined to national wants. Unemployment is an industrial accident or disease. In fact, it is the only industrial accident or disease. You can fall and be injured; that is an individual affair. You can have any kind of disease or you can suffer any kind of accident at home as well as at work, but you cannot be unemployed without losing your job; that is an industrial affair. It is the only true industrial accident there is, and it is the only form of industrial accident that we do not insure."

James Winston[20] expressed the opinion that there were as many cures for unemployment as speakers and that the cures conflicted in many respects. He said that an unemployed person is primarily interested in money to supply his needs, and secondarily in a job. He believes that this more fundamental concept might throw light on the varying points of view. He said:

"Some time ago it was reported in the press that the governor of a state ordered certain road work to be done with pick and shovel to the exclusion of machinery. It was claimed that by doing the job by hand 50 men would be employed 60 days as against 15 days if the job should be done with the aid of machinery, making a payroll of $15,000 as against $3750 otherwise.

"If the state wanted to spend $15,000 for that job, why did it not use its road machinery, complete the work in 15 days, and pay the 50 men $300 apiece and let these men do whatever they wished (or do nothing if they preferred) for the remaining 45 days? I cannot see that either the state, the men, or any one else benefited by keeping 50 men 60 days on a 15-day job. On the other hand, I do see some very harmful results from this practice.

"My reasoning is in a mathematical or technical line. On the other hand, I have heard prominent business men, bankers, and statesmen, using the common method of reasoning, prescribe for the same ill a high tariff, a low tariff, high wages, low wages, spending, saving, recognition of Russia, non-recognition of Russia, mergers, non-mergers, etc."

Mason A. Stone[21] raised the question of what makes business grow better eventually. He said that there is a belief that the stock market anticipates the condition of industry, but that he had not found it so. "Last winter I canvassed the town to see what was being done in the building industry and was met with the single reply, 'We can't get any money.' That brings up the subject of credit. The reason that they could not get money was that the bankers did not think it time to build. I think that with the price of materials now decreasing we will probably see the bankers loosen up and construction increase. Thus we will find a demand for raw materials, and without exactly knowing how it all happens, we will find ourselves looking for the swing upward. The stock market will then take note of the situation, and we will be ready for another crash in about seven years."

John A. Garvey.[22] The contribution of the Dennison Manufacturing Company to the subject under discussion will be a description of a practical experiment in unemployment prevention and relief. It is more than 20 years since the company began work on a program to reduce unemployment to its lowest possible terms. Preventing unemployment, even in one small company, is a man-sized job. It meant breaking down buyers' habits and selling prejudices. It meant instilling factory management with a new point of view as to the responsibility for keeping employees busy. Above all things we have found that it is not a job that stays done.

Sometimes in discussion of our unemployment plan we have been greeted with a remark: "Oh, Dennison business is different. You can do things with your line that we cannot do with ours." This is to some extent true, but the reason that Dennison business is different is because it has been made so. Although the product is a varied one, some of the main lines were highly seasonal in character, and in the early 1900's there were few businesses that had more seasonal unemployment than the Dennison box business. Most of the boxes and paper cases were supplied to the jewelry trade to be used with Christmas gifts. It was the custom for salesmen to go out in the early summer and sell the box lines for delivery in November and December. As a result the box shop was extremely busy from July until December, and then business fell off sharply.

When the plan of selling the box line in January rather than in June was first proposed, it was met by a stubborn opposition. Nevertheless, the experiment was tried out in one of the sales

[17] Counselor, New York, N. Y. Mem. A.S.M.E.

[18] Editor of *Factory and Industrial Management*, Chicago, Ill. Assoc. A.S.M.E.

[19] St. Paul, Minn. Mem. A.S.M.E.

[20] Chief Engineer, Elizabeth Iron Works, Elizabeth, N. J. Assoc. Mem. A.S.M.E.

[21] Engineer, Hine, Goldthwaite & Mylott, New York, N. Y. Mem. A.S.M.E.

[22] Personnel Manager, Dennison Mfg. Co., Framingham, Mass.

districts, and much to the surprise of most of our box salesmen it succeeded very well.

Within a few years our selling had been swung ahead from June until the early months of the year. The advantage to the factory in regularizing employment is obvious, for schedules could be laid out at the beginning of the year to produce with a stabilized force ten out of twelve months of the year. During the other two months stock could be manufactured to fill in the valley.

To cite an example from one of our other lines, consider our Christmas novelty merchandise. This line of seals, cards, tags, etc., requires expensive machinery and high-class workmen. We found it impossible to keep men and machines busy during the year on Christmas goods, and therefore evolved the plan of popularizing goods of the same type for other holidays in the year. This has been done for St. Patrick's Day, Easter, Hallowe'en, etc. And so to a far greater extent than before we have been able to regularize our production on this type of merchandise.

Our business is not vitally different from any other, and it requires long-range planning as well as considerable energy to get away from seasonal depressions. The program instituted for keeping cyclical unemployment at a minimum differs but little from what now seems to be more or less standard practice.

We guard against overexpansion of facilities and building during peak times. We do not launch new items when business is at the top, but we wait until it begins to fall off, and then use these items as a stimulus for additional business. We keep our advertising expenditure low during rush periods, and increase expenditures when the depression seems to be coming on. We try to keep inventories small during the expansion and build them up during depressions. We centralize the control of hiring to be sure that extra people are not taken on at a time when business is about to fall off. We try to train our employees to be versatile so that they may be shifted from one job to another. All these principles have been worked out little by little over a course of years and have been accepted by our management as being sound.

So much for prevention. But our directors went one step further. They knew that in spite of all these provisions some unemployment would probably come at intervals, and therefore they put aside a fund which would be used for the relief of unemployment. This fund of $150,000 was accumulated during the boom days of 1916–1920, and it was in 1920, after more than ten years of experience in building up a prevention program, that the relief plan was started.

In conference with the employee representatives a plan was devised which provides that all employees of more than six months' service who are temporarily laid off for lack of work will receive 50 per cent of their wages if they have no dependents and 70 per cent if they have dependents. The plan provides that, as soon as there is lack of work, employees who are below the standard requirements will be dropped first, unless there is special reason for keeping them on the payroll; that employees with less than six months' service shall go next; and then as far as possible transfers shall be made from those departments where layoffs are necessary to departments where business may be better. The difference between what employees are actually worth in the department to which they are transferred and the wages which they have been receiving is made up out of the fund. In other words, layoffs are the last step to take after all other steps fail.

Our first test of this plan came in 1921, and we were able to keep practically the entire force at work, and the total expenditure from the fund was only slightly over $20,000. We had a number of new items to throw on the market, and these happened to take very well, backed up by an advertising expenditure which was very much greater than we had used in previous years.

It would be well if it could be said that the same situation exists this year, but unfortunately it is not so. The reasons for this are varied. During the period since 1925 we have had, in common with other manufacturers, a hard fight to retain our markets. During this five-year period, which has been decidedly a buyers' market, we have not been in a position to hold back any new items for the day of severe depression; we have had to launch them as soon as they were ready in order to keep up our usual sales. We were unable to set aside large amounts from our advertising budget. On the contrary we have had to increase our advertising considerably year after year to give our sales folk the backing which they needed. Although inventories have been kept down, this year is no time to increase them, because the style element, even in our varied merchandise, has become important, and paper boxes and novelties of all sorts change styles much more rapidly than was the case 10 or 15 years ago. To be left with a large amount of novelties or boxes on our hands that are out of style would mean a very large loss. Moreover, continual pressure for cost reduction has meant continual work on better methods in order to reduce direct labor and overhead. Installation of machines and facilities for getting out production without the necessity of hand labor has been, as the world well knows, an added force which makes the recent depression a more difficult one than perhaps any other in this generation. The Dennison company has been no different from other companies in feeling the force of this development.

Yet in spite of these factors our experience has not been unsatisfactory taken all in all.

The total number dropped from our employee group since last January is less than 15 per cent of our total force, and this includes the six-month employees who do not fall under our unemployment benefit fund. We have kept most of our factory working on the 44-hour week, although certain departments have been shut down Saturday mornings occasionally.

Expenditures from the unemployment fund for layoffs and transfers have been very much higher than in any other year and will total before the first of 1931 more than $50,000. While this sum is a fairly large one taken by itself, considered in the light of our total yearly payroll it amounts to only 2 per cent, which is much less than many companies spend for projects far less worth while, both to the company and to their employees, than our unemployment fund could possibly be considered to be.

That point should be emphasized to business management—that while an unemployment fund may be a benefit to the employees, it is often of more benefit to management. This statement may not be taken seriously by many managers, and yet our experience of ten years with the fund has taught us that it is true.

It may be thought that the importance of relief rather than of prevention is being unduly stressed, particularly on a program which is devoted to the matter of prevention. Yet the relief side bears the same relation to the prevention of unemployment as the expenditure for fire-insurance premiums has to fire prevention. Years ago business men looked upon accidents as being very sad, and they extended their sympathy to cases where employees were injured or to the families where employees were killed. Yet they did very little about it until accident insurance was made necessary and they were obliged to pay either prevention expense or increased insurance premiums.

So it may be said that the spending of real money for unemployment relief has made the Dennison Manufacturing Company less theoretical about unemployment and more practical. There have been unquestionably fewer platitudes and more action since our fund began to operate. It is a spur to greater effort on the part of our salesmen, and we hear much less of that old story that the factory exists to produce what the sales organiza-

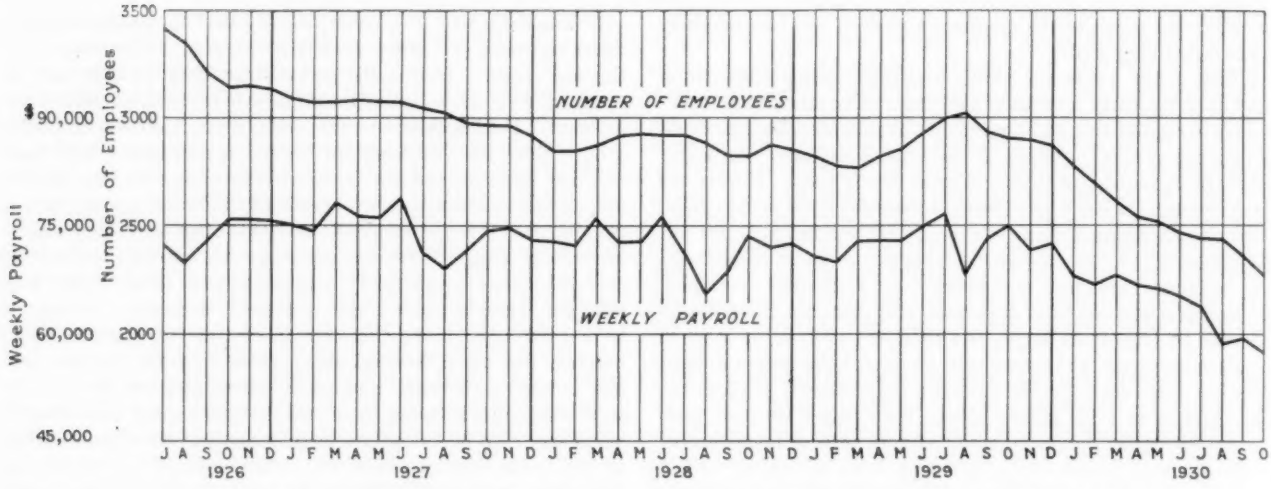


FIG. 1 CHART OF NUMBER OF EMPLOYEES AND AMOUNT OF PAYROLL IN DENNISON MFG. CO.

tion sells and not to give the sales organization orders as to what it shall sell.

At the same time the factory management is forced to take its share of responsibility, and the expenditures from the unemployment fund of each department are given plenty of publicity. As a result the factory management has become more farsighted; they are careful in regard to additions to their force in peak times; they are more apt to spend time training employees to become versatile on various jobs; they are more alive to changing conditions rather than keeping their nose well down to the grindstone and letting the rest of the world go by.

No longer does the sales management say about unemployment, "That is too bad, but really it is not our lookout." No longer does the factory management say, "Well, we are real sorry, but we cannot help it if there are no sales." Today both factory and the sales organization are drawn into close cooperation to prevent layoffs which can be humanly avoided.

And what about our employee group? It is safe to say that they have been more interested in our unemployment scheme in its various developments than in any other project that has been undertaken in our personnel program. In fact, one of our older employees who for many years served as chairman of our Employee Representative Committee said one day: "We can give up our lunch room and the clinic and the suggestion system and even the profit-sharing plan; all of them together are not as important as the unemployment plan. For if a man has a steady income, he can get along without all the other things, but if he has not a steady income, all the other things together cannot help him very much."

Our employees generally recognize the fact that our management is actively interested in this program of unemployment relief and prevention. They recognize the fact that here is one important subject where cooperation of employees and management is something more than a mere catchword. Through their Employees Committee we have brought them into active contact with the problem we have been facing to meet competition to provide steady work. Just as our management has learned a great deal from its experience with this subject, so the employees are receiving an education that it would be hard to give them in any other way. It would be too much to expect that they should forget their personal interests to safeguard our unemployment fund, and yet during the past year they have agreed to, and often have initiated, changes in the plan which we hardly would have expected only a few years ago. And so we face the continuance of our unemployment experiment with confidence that as the years go on we shall work out continually better practices. Although the difficult conditions of 1930 have made it impossible for us to achieve the same results that we were able to achieve in 1921, we have done better, we know, than we could otherwise have done if our plan for prevention and relief of unemployment was not in operation; and we feel even more certain as the result of our experiences this year that fundamentally the program that we have undertaken is both sound and worth while. The details of it will have to be changed, developments will have to be made, but the fundamentals, we are sure, are sound and workable.

E. F. DUBRUL.[23] The machine-tool industry, through the National Machine Tool Builders' Association, is continually taking stock of various conditions. In doing so we at times discover some things that may be of benefit to all other groups of machinery and equipment builders. Some of the latest investigations fail to confirm ideas that were widely prevalent during the last "new era" of stock booming, and also disprove other ideas concerning the widespread prevalence of "technological" unemployment, which is worrying so many people right now.

All can remember how the boomers used to say that last year's stock prices reflected the increasing prosperity of the country due to things such as these:

"Alacrity with which American business management adopted countless new improvements in existing implements of production."

"Strides in building new and more efficient machinery, demands for perfected types of productive instruments, a scientifically controlled industrial and commercial régime brought increasing profits to industry."

They explained the high prices of stocks as being a proportionate reflection of the widespread adoption of the newer productive instruments and methods. There is reason to think this to be a greatly exaggerated picture and one which gave stockholders, bankers, and directors a false sense of security.

That glowing picture seems greatly overpainted as concerns the metal-working industries as a whole.

The curve of stock values, as calculated by the Standard Statistics Company, tended upward enormously from early 1924 to the peak of August or September, 1929. The rise in values was out of all proportion to the increase in earnings of the companies. A glance at the curve of machine-tool orders shows it to have

[23] General Manager, National Machine Tool Builders Association, Cincinnati, Ohio. Assoc. A.S.M.E.

fluctuated at a low level for seven years from 1921 through 1927, and to begin its ballooning only in 1928.

Certainly this curve shows no great "alacrity" to adopt the new and more productive machines which had been designed during that time. Quite in line with these statistical facts were the complaints of the sales forces concerning the difficulty of getting users even to investigate the profit-making possibilities of these machines. So one may wonder why, during those seven years, management displayed so little "alacrity to adopt new methods."

The curve of machine-tool orders seems to indicate such alacrity only in 1928 and 1929. With the drop in the curve of orders to levels below 1927, we wonder where all this "alacrity" has now been stowed away, and how long it will remain under cover. Then, too, some other things make us doubt this "alacrity theory."

In 1926 the *American Machinist* made a survey of the ages of machine tools in service on January 1 of that year. This survey showed about 44 per cent to be over ten years of age.

This year the same paper made a similar survey which indicates that about 48 per cent or more of the machine tools in service on January 1 of 1930 are over ten years old. Apparently the "alacrity to adopt new methods" had been running so much "in reverse" in 1926 and 1927 that our boom orders of 1928 and 1929 were not enough to keep equipment as youthful as it was in 1926.

While some concerns did make a considerable advance over the manufacturing technique prevalent before the Great War it seems probable that the total advance was not as much as the enthusiastic stock-market boomers led the public to believe in the machine-shop field.

It is hard to believe that the managers of the 15,000 important machine-tool-using plants are greatly different, as a class, from the managers of plants that use other kinds of machinery.

So we suspect that there is not much basis for worry about the effects of "technological" unemployment. We suspect that large numbers of manufacturers do not realize how much their plants need modernizing to maintain and advance our present American standard of living.

We suspect that this condition may be due to lack of understanding of the problem of productivity on the part of bankers and financial men who control the expenditures of corporations of all kinds.

These surmises seem clear enough in the case of machine tools, but we have no similar surveys of other machine industries with which to compare our surmises. If these surmises are correct, it appears that there is a job ahead for the whole machinery industry in educating financial America still more concerning the financial benefits to be gained by installing more modern machinery of all kinds.

If the financial clamps were taken off, the machinery industries could do much to give more employment to a well-paid class of workmen, and thereby greatly increase purchasing power for consumers' goods.

GEORGE G. BERGER.[24] Much improvement can be effected and the severity of a depression considerably lessened by the application of the fundamental principles of scientific management. One of these principles is the principle of evaluating results accomplished and using the rating as a basis for wage payment for the purpose of stimulating improvement in these results. The principle of evaluation in terms of adequate standards has many proofs of its force as a stimulating agent. The scientific notation of physical properties of a product may be considered an evaluation on the basis of a standard. By its clarity, the evaluation stimulates the proper design of means to effect the end, the product produced. These evaluations in turn become the standards by which conditions are set for the worker's task. Payment on the basis of the worker's task stimulates productivity. Here we make important discoveries which have been proved by considerable time-study technique since Taylor's time. They are:

1 The human being is capable of doing much more work than he thinks he can do, and which becomes apparent only on stimulation.

2 There are fairly defined limits to the higher production rates which tend to make well-trained workers equal.

3 Wage payment based on productivity is the most effective means of stimulating improvement in productivity.

Now what I have really in mind is the evaluation of the results of the industrial administration, an evaluation in terms not of predicted standards with broad tolerances, but of natural standards set with the idea of maximum turnover of capital in mind. Such standards with one exception are to be secured only when the results to be evaluated are in. This idea of evaluating managerial results has come up before this society under the familiar heading of "the measurement of management." Consider what an effective evaluation or rating of the results of management leads to. First, it can be used to stimulate the personnel to secure improvement in results—that is, to increase competitive power, and work for the survival of the fittest in our economic structure. Second, as a basis for bonus payment, it serves to stimulate the administration executive to produce results which he may not think possible today. For although responsibility accompanies authority, responsibility is psychologically effected by stimulation, by gains accruing to the individual. Third, the balance sheet may be adjusted by the managerial constant to be derived from the ratings to give a conservative statement of what a concern is really worth, making for stability of investment values. These are all important factors in industrial and employment stability.

FRED K. HOEHLER.[25] During the depression of the early months of 1930, there were signs of an awakened public interest beyond anything experienced in former years, in the regularization of employment and the causes of unemployment. This same interest was created in Cincinnati as early as January, 1929, when that city, like many others, was enjoying a considerable amount of prosperity. The Permanent Committee on the Stabilization of Employment, consisting of twenty members, was appointed by the City Manager. The committee elected the City Manager, C. O. Sherrill, as its first chairman and his Director of Public Welfare was elected secretary, thus giving it official standing. The committee membership, representing banking, industry, education, social service, labor, and government, determined the purposes of the committee to be as follows:

(*a*) To study the problems of stabilizing employment

(*b*) To create machinery to handle an unemployment emergency should one arise.

The permanent committee began working through ten subcommittees, as follows: (1) State City Employment Service, (2) Continuous Employment, (3) Temporary Employment, (4) Public Works, (5) Cooperation With Private Social Agencies, (6) Budget and Finance, (7) State and National Cooperation, (8) Transients, (9) Fact Finding, (10) Publicity and Education.

The one great fact facing the committee was the absolute lack of reliable information on employment and unemployment. The very first step, therefore, was to secure and publish accurate and adequate information. There was in existence no one source from which such information could be obtained.

[24] Consulting Industrial Engineer, New York, N. Y. Jun. A.S.M.E.

[25] Director of Public Welfare, Cincinnati, Ohio.

The Subcommittee on Fact Finding utilized the best fact-finding agencies in the community and began work at once. In May, 1929, Cincinnati had its first complete census of unemployment. This was accomplished by using the machinery created by the Board of Education. This body is required by state law to secure an annual school attendance census by means of a house-to-house canvas. The board willingly gave its cooperation to the Fact Finding Committee, and enumerators were requested to ask a few extra questions. The results were later tabulated in the office of the Department of Public Welfare. The Fact Finding Committee then, with a fairly reliable census as a basis, developed current monthly figures on employment. It was this committee which saw the tendency toward an increase in unemployment in the summer of 1929, and so informed other subcommittees, which began to plan intelligently to meet a situation which was imminent.

One of the first tasks of the Committee on Stabilization of Employment was to strengthen the State City Employment Bureau. A subcommittee made a series of recommendations for improving records and methods of functioning. This committee inaugurated the monthly bulletin in Cincinnati and has encouraged employers to use the bureau continuously in supplying their labor needs. Regular and constant visits to industry by members of the staff were urged and insisted upon. The committee plans in more normal times to develop the bureau as a center for employment clearance and employment information.

The need for a strong Federal employment service, cooperating with state and local governments, similar to the Federal Provincial System of Canada, was stressed in the finding of the committee early in April of 1929.

The Subcommittee on Continuous Employment, which was headed by an economist and a former president of the University of Cincinnati, made a study of the concerns in Cincinnati to determine how many of the industries in that city made a conscious effort to stabilize employment. This committee realized that it must have available an adequate statistical and information service and a well-organized and efficient employment exchange to serve industry in its various employment problems. The work had barely begun when unemployment began to grow. The first essential, then, for this committee in dealing with an unemployment emergency was to spread whatever work might be available to as many employees as possible. The committee encouraged the introduction of methods such as the staggering of employment and using employees on a short-hour basis when production had to be decreased. Cooperation of the employers was very heartening and indicates the type of interest which will be given to this committee and its work as it continues throughout the years. Over 50 per cent of the employers reduced hours in order to provide work for the largest possible number of their men.

Largely through the activities of the permanent committee, the city, county, and state governmental units were active in speeding up public works. Public improvements which normally would have been deferred to another time were started during the last winter while labor was plentiful. Plans were formulated to start other improvements early in the spring. The city of Cincinnati and the various city, village, and county units surrounding that city have furnished more work for the unemployed during the past winter and spring than they have ever furnished before under similar conditions. This expenditure of public funds was not only well timed, but was according to a program well thought out in advance. Cincinnati and Hamilton County, Ohio, have for the last few years operated all public improvements on a five-year public-improvement program. Bonds are authorized after the Improvement Program Committee studies the projects, and the same committee then recommends and authorizes the expenditures of such bonds. This has virtually created a public-works reserve.

The Committee on Temporary Employment made a survey of possible jobs should an emergency arise, and set up a large committee of over 100 members geographically distributed. This committee functioned in locating jobs of a temporary nature. These jobs were sought in and about homes or in factories or business houses. In addition to the temporary jobs actually secured by the committee, its efforts helped to create a desire on the part of the public to be helpful, and many temporary jobs, not cleared through the committee, were given out direct to the unemployed.

After the various committees had secured all the work possible the residue of the unemployed were cared for through relief-giving methods. The Subcommittee on the Cooperation of Social Agencies was in charge of this phase of the work. Even here the policy of furnishing work, rather than relief, was followed out so far as possible.

A joint committee sponsored by the Public Welfare Department of the city and the Community Chest operated an industrial relief program, in conjunction with the State City Employment Service, more to furnish work than to provide straight relief.

The reason for securing this work at public and semi-public institutions was that the money which was used to pay for the work came out of tax funds and Community Chest funds. In the city and county we had work only in such institutions as the Hospital, County Home, and in the parks. Other public works were considered a legitimate charge on the regular funds and not as relief work.

The types of work provided by these institutions were cleaning, odd jobs of repair work, digging weeds, repairing camp projects, etc. The committee endeavored always to avoid conflict with regular labor organizations and skilled workers.

The wage paid for work supplied in the industrial plan program is only 40 cents an hour. This is considerably under the minimum wage paid by the city, which is 47 cents an hour, and by the Association of General Contractors, which is 50 cents an hour.

It is important, in this regard, to have the weight of the influence of the best employers of Cincinnati behind our effort so as to prevent a reduction of the hourly wage.

In referring the men to the jobs, preference was given to heads of families who are residents of this community.

In the early days of this experience, 44 hours a week were allowed each man. Because of the number of calls for this kind of assistance and the limited funds available, this was arbitrarily reduced to 24 hours a week as a minimum, with some cases allowed additional time because of special need. Where a man was on a 24-hour schedule, he worked for three consecutive days; i.e., Monday, Tuesday, and Wednesday, with Thursday, Friday, and Saturday free, during which time he could seek odd jobs to augment his earnings. There were two crews of men working; the first crew was paid off Thursday morning, and the second crew was paid off Saturday evening or Monday morning.

A card was used at the industrial relief headquarters for introducing a man for work to his employer. This card was later signed by the employer, who recorded the number of hours for which the man should be paid. At the end of a regular period the supervisor or the superintendent of the institution reports in letter form on the character of the work done and gives his opinion of each individual.

Men must not be allowed to use this as a solution of their unemployment problem. A great deal of vigilance is necessary in order to be assured that the man is not content with this work and will try to secure a permanent place. Where a man refused to take a regular job offered him, he was perhaps penalized by not being allowed industrial relief for a certain period.

The personnel required for this program is two capable social workers and one employment man or a man with employment experience who can follow up these workers and act as paymaster. The two social workers are used for interviewing the men and contacting social agencies.

This method was found to be not only more effective than relief-giving, but more economical. Over 3000 families were taken care of during the last winter by this committee.

The transient problem was handled directly by the Department of Public Welfare. Some of these men were found temporary employment in odd jobs; most of them were not looking for steady employment.

During the past winter nearly 10,000 homeless and transient men were provided with lodging, and nearly 200,000 meals were served. Each man who applied for assistance was given a work card, and by putting in a certain number of hours in the woodyard he could secure for himself lodging, meals, and some odd pieces of clothing. Through this work program the department was able to provide 40,000 bundles of wood to be distributed to families of the unemployed through social agencies. The deficit for the Transient Service Restaurant was carried on by a young woman in Cincinnati who has been working during the past winter with all activities of the Stabilization Committee.

Where it was necessary to give relief to families when the breadwinners were out of work, it was done largely through the regular family welfare agencies, with scarcely any increase in their personnel. The Community Chest appropriated extra funds to be used by the private agencies on unemployment cases. Due to the fact that the case workers of the relief societies knew absolutely that relief funds would be furnished, they were able to handle the problem of relief intelligently. The condition of panic and uncertainty that so often exists among relief workers and their clients in an unemployment crisis was almost entirely lacking in the Cincinnati area during the last winter.

Over 8000 families in which unemployment was the major cause of distress, in addition to those given industrial relief, were taken care of by private agencies during the winter.

The total cost for relief in Cincinnati, as carried by private and public agencies from June 1, 1930, to June 1, 1931, is slightly in excess of $650,000. This is a comparatively small sum for a city the size of Cincinnati. Planning and effective organization has made possible a very good job with relatively low cost.

The question of legislation as a relief for unemployment, such as the matter of unemployment insurance and old-age pensions, has not been discussed by the Permanent Committee on Stabilizing Employment, but our program includes a thorough study of such types of legislation, with the hope that the committee might find something which it can unitedly support before the community.

In brief, the Cincinnati committee has accomplished cooperation between government, the social agencies, and Chamber of Commerce, the university, labor, and industry, for the study of the problems of unemployment.

Progress in Materials Handling

Contributed by the Materials Handling Division

Executive Committee: G. E. Hagemann, *Chairman*, M. W. Potts, *Secretary*, F. D. Campbell, C. D. Bray, E. D. Smith, and F. L. Eidmann

ALTHOUGH the year has been marked by no spectacular developments in the field of materials handling, it has been one of growth and progress. The quiet business conditions of 1930 have had at least one beneficial effect—stressing upon manufacturing plants the need for making increased use of facilities to lower the cost of production. As soon as business conditions again offer the opportunity to make further investments in plant equipment, materials-handling manufacturers are certain to profit from the situation. Materials-handling equipment offers one of the most definite means for obtaining continuously lower production costs, not only through the substitution of mechanical operation for manual labor, but also through the elimination of useless steps in the manufacturing cycle.

In the following paragraphs are listed comments on such of the important new developments in equipment and methods of application as could be collected from manufacturers for publication in this Report. The work of compilation was divided among fifteen men, and, in each section of the Report credit is given, by means of a footnote, to the individual who prepared the material in that particular field.

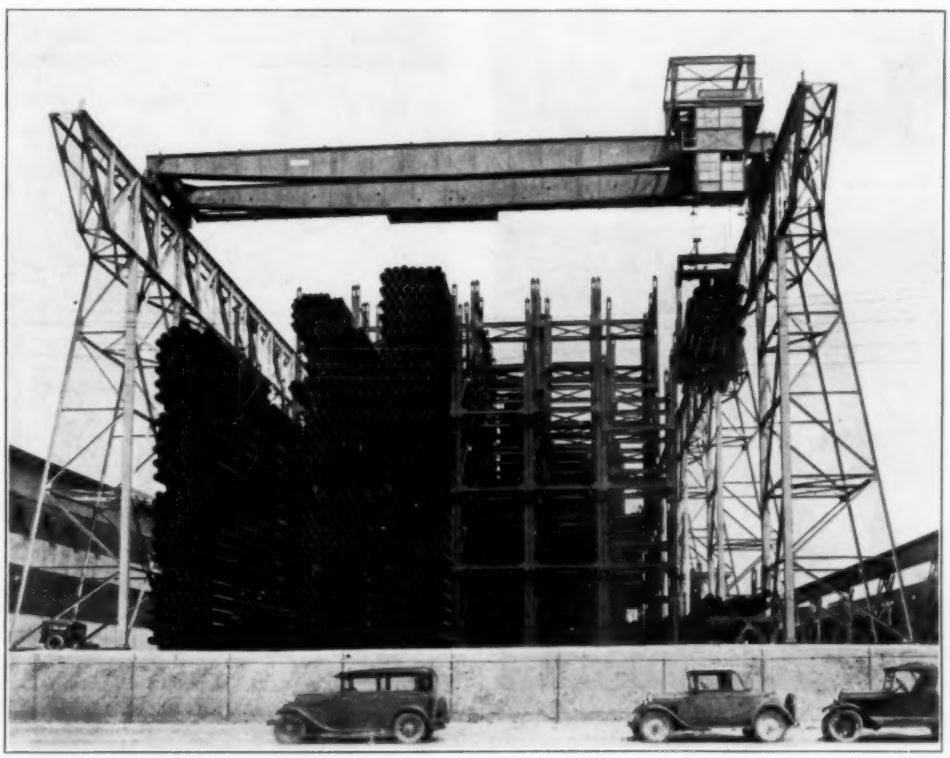


FIG. 1 HANDLING WELDED PIPE STORED IN UNIT FRAMES WITH OVERHEAD TRAVELING CRANE

OVERHEAD TRAVELING CRANES[1]

Engineers have collaborated with industry in designing cranes that provide a maximum of service under the hook, high speed, and low maintenance cost. Anti-friction bearings, improved lubrication, precision in manufacture, and flexibility of design have all contributed to this end.

[1] Section prepared by Henry W. Standart, President, Northern Engineering Works, Detroit, Mich.

Of signal importance is the application of the overhead traveling crane to special service requirements. In this effort all manufacturers have contributed generously, until today there is scarcely an industry which has not adopted the overhead traveling crane as a means toward the economical handling of material. A few interesting applications of the traveling crane are presented herewith.

The handling of carload shipments in containers has contributed to economy and efficiency in freight yards. The Harnischfeger Corporation recently has applied this principle in an interesting manner for the handling of carloads of welded pipes in units, as shown in Fig. 1. The pipe is stored in unit frames and, when shipped, the entire frame is lowered into the car.

Many new applications of the overhead traveling crane are found in handling paper, the Harnischfeger Corporation recently having developed an automatic grapple, shown in Fig. 2 for handling rolls of paper used in the manufacture of containers.

The rayon and dyeing industries, fields recently developed for the traveling crane, have brought about many new and novel applications. The Box Crane and Hoist Corporation has introduced the container principle into this development. Cranes are employed for handling containers, or racks, for dipping material into dyeing vats, as shown in Fig. 3.

Still another interesting use of the traveling crane is found in the field of sports. The Northern Engineering Works has applied the standard overhead traveling crane to the handling of speed boats in a motor-boat garage, as illustrated in Fig. 4. This crane is of the low-headroom drop-truck type, permitting applications within limited clearances. An adjustable head

beam is used for handling boats of various lengths. The boats, when out of service, are removed from the water and placed in storage cradles.

Electric Hoists and Cupola-Charging Machines[2]

Electric Hoists. Several new electric hoists, mainly in the small capacities, have been rather recently placed on the market. They are designed to supply the higher speeds now currently demanded. The single-speed reversing switch, which was adequate to handle the slower speeds previously prevalent, is no longer suitable to control these higher speeds, on account of the more severe resultant strains on the hoist mechanism. These



Fig. 2 Handling Rolls of Paper With an Automatic Grapple

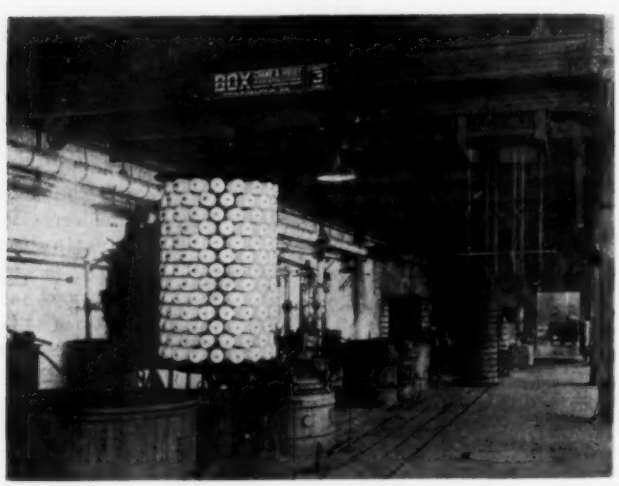


Fig. 3 Rack With Material to Be Dipped in Dyeing Vat

higher speeds also require more rugged brakes and greater braking area.

There is now a general use of heat-treated steel gears, of special-steel shafts, and higher-speed motors of special design for hoisting service only. The lack of lubrication being the greatest cause of trouble, there is a constant endeavor to cut down the number of places to be oiled or greased, so that no place will be overlooked.

A novel use has been developed for single-I-beam hoists made by the Northern Engineering Works. These hoists are now used for handling motor boats from the water into storage in a "boat garage" and then back into the water. Boats varying in lengths of from 17 to 28 ft., and in weight from 1500 to 4500 lb., are hauled out of the water, repaired, and launched again. Cranes are likewise used for this purpose as explained previously in the text and illustrated in Fig. 4.

Cage-Controlled Monorail Hoists. New cage-controlled monorail hoists, in small capacities have been developed to meet the

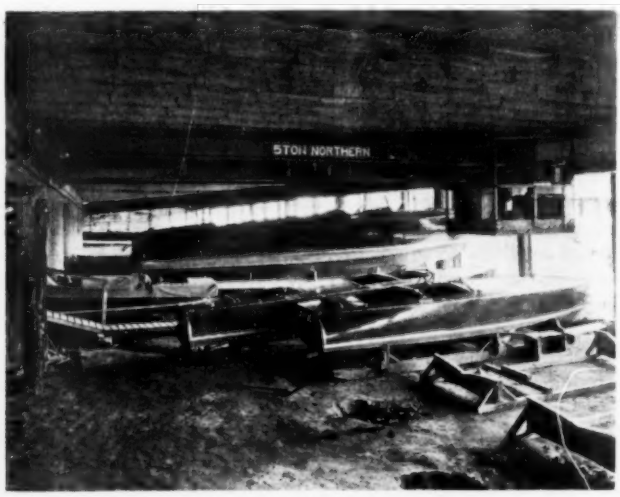


Fig. 4 Handling Speed Boats in a Motor-Boat Garage



Fig. 5 Light Cage-Controlled Monorail Hoist Rounding a Short-Radius Curve

overhead-transportation problem. These hoists can transport small loads at higher rates of speed around curves of small radius through narrow-throw switches, and operate on a system of transfer cranes. An installation view of such a light, cage-controlled monorail hoist, made by the Louden Machinery

[2] Section prepared by Robert T. Turner, Vice-Pres., Shepard Niles Crane & Hoist Corp., Montour Falls, N. Y. Assoc-Mem. A.S.M.E.

Company, is shown in Fig. 5 rounding a short-radius curve, and in connection with a transfer switch.

Heavy, rugged, cage-controlled monorail hoists have also been designed to perform work hitherto delegated to 5- or 10-ton-capacity overhead electric traveling cranes. These cage-controlled hoists are now designed to furnish the extremely long lifts that are required for grab-bucket and magnet service. The hoists are built to meet more restricted clearances than formerly, and are provided with larger cabs to contain the more complicated control apparatus required for handling grab buckets and magnets and for the three travel motions.

The cage-controlled hoist shown in Fig. 6, and made by the Shepard Niles Crane & Hoist Corporation, handles a $1^1/_4$-cu. yd. bucket. It has a 50-ft. lift and goes through a door that is 71 in. in height to the power house. The hoist unloads coal

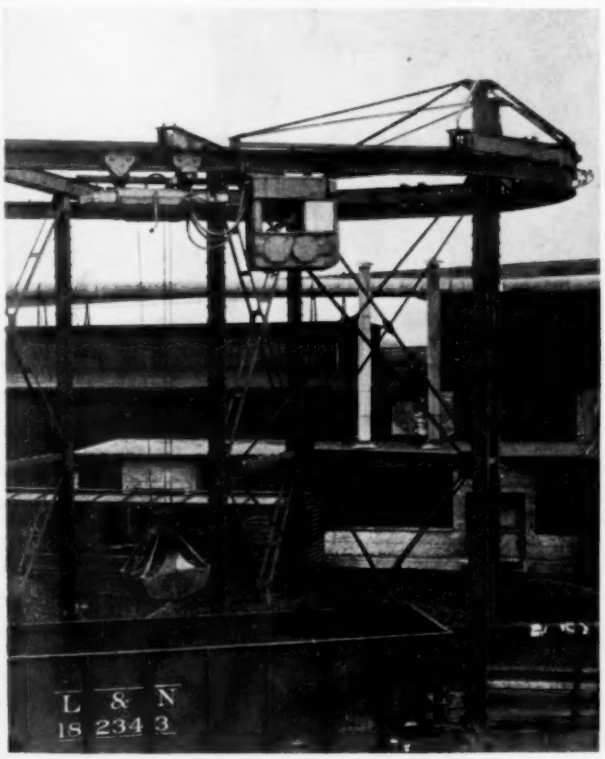


Fig. 6 Cage-Controlled Hoist With $1^1/_4$-Cu. Yd. Bucket for Unloading Coal and Carrying It to Hoppers

from cars to hoppers located on the third floor of the power house, takes the coal from cars to storage piles under a portion of the runway, and also reclaims the coal from storage piles to the hoppers.

Hoists and Cranes for Cupola Charging. Cage-controlled monorail hoists for cupola charging have been developed to handle the larger charges now required. They are designed to carry these larger charges in buckets either of the bottom-dump type or of the cone type and either to lower these buckets into the cupola or discharge them directly inside the cupola opening. These cupola-charging hoists are also used for grab-bucket and magnet service, and often in connection with transfer cranes. The transfer crane can be latched on to spur tracks leading into the cupolas, and the cupola-charging hoist run off on the spur track to deposit the charge in the cupola.

A simple manner in which a cage-controlled cupola-charging hoist of Shepard Niles Crane & Hoist Corporation design can be installed in a foundry having a very small melt is shown in Fig. 7. There is no covered portion either to the charging floor or crane runway. This charging floor was built over the roof of the foundry when the location of the cupola was changed. The materials are brought to a convenient location at the scale in the foundry yard by means of industrial cars, and from there put into a bucket by hand. This operation requires only one man. The charger is cage-operated, and thus two men are required to complete the cycle of charging. The man operating the charger takes care of hand charging the coke into the cupola,

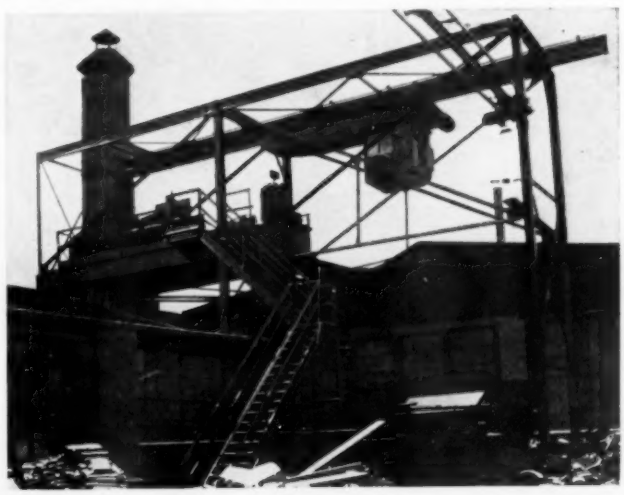


Fig. 7 Cage-Controlled Cupola-Charging Hoist

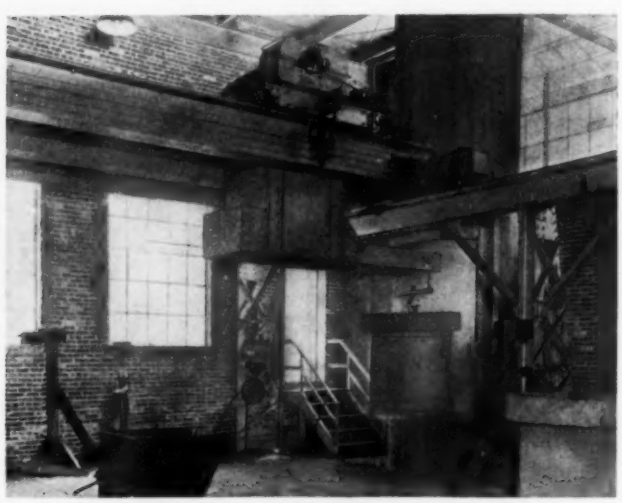


Fig. 8 Underslung Crane-Type Charging System Showing Cone-Bottom Bucket Entering Cupola

the coke being delivered by means of the charging bucket. This arrangement provides an inexpensive installation which operates at low cost.

Double-rail hoists for cupola charging have been developed by the Whiting Corporation. This horseshoe type of charger is useful for serving a single cupola. It is applicable especially to buildings where the headroom is low, because it can operate between the roof trusses.

A Whiting Corporation crane-type charging machine is shown in Fig. 8. This machine will serve a battery of six cupolas operating three one day and three the next, in long heats of from 9 to 18 hours. It can be installed on narrow charging floors with limited headroom.

Conveyors[3]

Coal-Handling Equipment. Coal-handling equipment of the full-automatic, trolley bucket-conveyor type has been designed and developed for the retail coal yard by the Gifford-Wood Co., designers and builders of complete coal-handling plants and ice-handling plants, so that coal can now be handled entirely automatically and with a minimum of breakage. The equipment is built in two sizes, one having a bucket of 65 cu. ft. capacity, and the other one of 100 cu. ft. capacity. It consists essentially of a hoisting unit for raising and lowering the bucket, and a traversing unit that controls the horizontal motion of the bucket over the various bins. It is necessary only to push the "start" button and the large bucket (see Fig. 9) lowers into the track pit, tripping the loading gate, which permits the bucket to fill with coal. The bucket remains in the pit for an interval of from seven to fifteen seconds, the length of this interval being adjustable and of a duration sufficient to enable the bucket to obtain a full load of coal. It then immediately ascends to the trolley (see Fig. 10), where it stops automatically; the trolley starts up and carries the bucket to a predetermined bin, where it automatically stops. The hoisting unit starts up in the reverse direction, lowering the bucket until the latch pads on the under side of the bucket touch the pile of coal in the bin, thus tripping the bucket and depositing the load of coal. At this point a specially designed bucket-return sheave, Fig. 11, immediately reverses the hoisting unit and carries the bucket up to the trolley, where it stops and is carried back by the trolley to a point over the track pit, where the bucket descends for another load. This full cycle of operations is entirely automatic, each separate operation being controlled by an electric control panel.

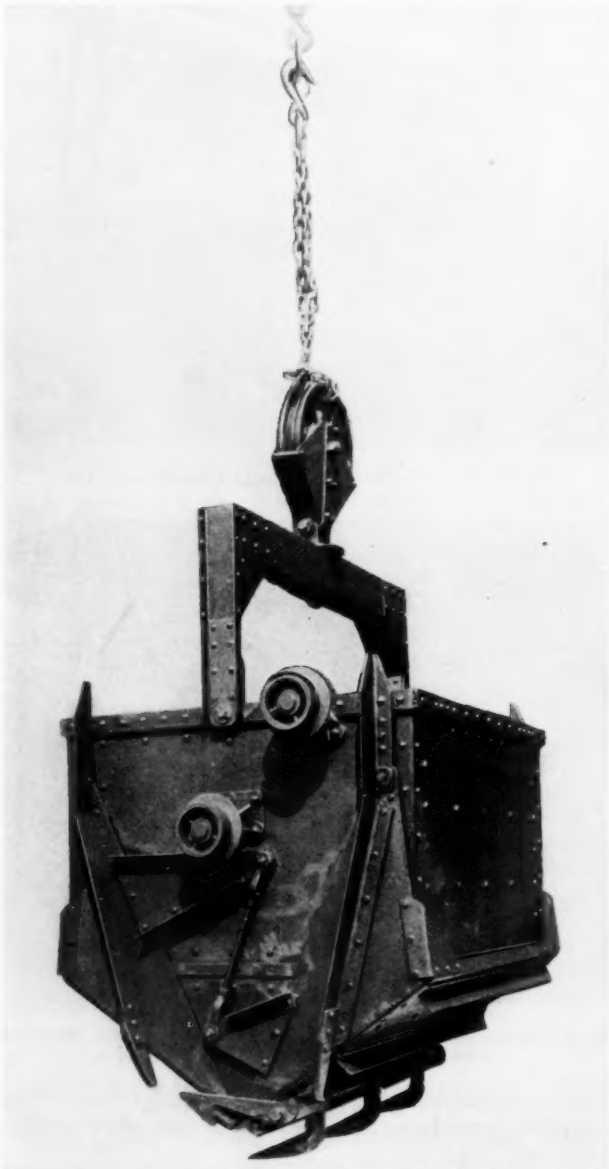


Fig. 9 125-Cu. Ft. Trolley Bucket Conveyor

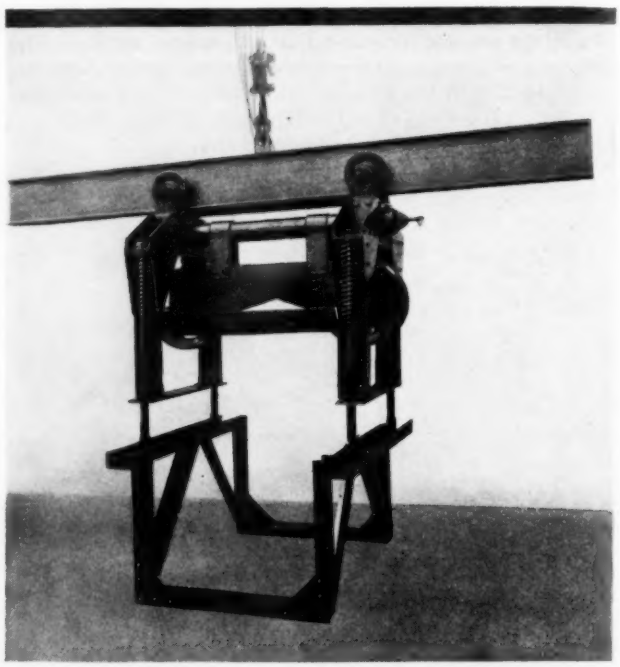


Fig. 10 Trolley With Bucket Buffer and Stabilizer

The equipment throughout is designed for ease of operation. Limit switches on both the hoisting and traversing units are readily adjustable, and it requires only a few seconds to set the equipment so that the coal will be delivered to any one of the various bins.

The trolley which conveys the bucket over the bin is provided with a spring buffer and stabilizer (see Fig. 10) which prevents the bucket from swinging while the trolley is in motion.

The bucket-return and cable-overload sheave (see Fig. 11) reverses the direction of the bucket after it has deposited its load of coal in the bin. It can be adjusted so finely for quick reverse that the weight of the bucket never rests on the coal already in the bin. This eliminates the crushing effect of the bucket coming in contact with the coal pile and, naturally, saves considerable breakage. This sheave also has another function in that it acts as a safety device. Should the progress of the bucket be impeded for some unforeseen reason, this sheave automatically shuts off the power, and consequently the equipment, long before sufficient strain is put on the cable to break it.

When the trolley returns to a point over the bucket guides which lead down to the track pit, an arrangement of limit switches slows it down gradually to a dead stop. If, for any reason, such

[3] Section prepared by Frank D. Campbell, Chief Engineer, Eastern Corporation, New York, N. Y. Mem. A.S.M.E.

as cable stretch, the trolley should go beyond the determined point over the pit guides, it immediately reverses and comes back to the right position. This provision always insures that the bucket will be exactly central over the guides.

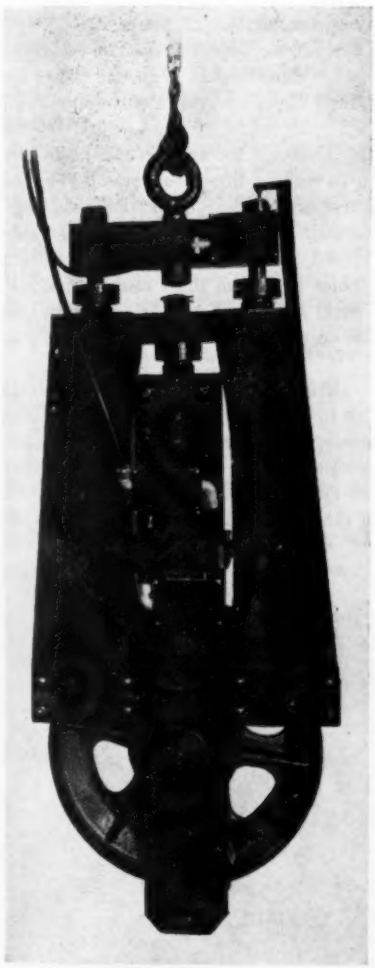


FIG. 11 BUCKET-RETURN AND CABLE-OVERLOAD SHEAVE

Fig. 12 shows a typical installation of the coal-handling equipment, utilized for storing and unloading coke.

Ice-Handling Equipment. Gifford-Wood Co. has recently put on the market several forms of ice-handling equipment to lower the cost of handling ice in manufactured-ice plants, by eliminating hand labor. Among these items are the multiple cake conveyors, spiral lowering chutes, and power transfer conveyors described in the following paragraphs.

Multiple Cake Conveyors. In a modern manufactured-ice plant, the practice of multiple cake pulls necessitated the development of equipment for handling these cakes from the dump to the storage room. The number of cakes pulled from the freezing tank at one time varies in different plants. There are a number of cases where as many as twenty-two cakes are pulled simultaneously. It was necessary to have a conveyor which would handle this full line of cakes at one time and discharge them to the storage room.

The tank-room conveyor, as developed by Gifford-Wood Co., is of the double-chain type, having a pusher attachment on each chain, which pushes the entire line of cakes as received from the dump. Although the two chains on the conveyor are always horizontal, the conveyor bed or ice slides are sloped, as will be noted in Fig. 13. This conveyor pushes the line of cakes at a speed of from 5 to 10 ft. per min., depending upon the conditions obtaining in the individual plant. A small doorway is cut through the wall to the storage room, and, as the line of cakes moves along, when each cake reaches this doorway, due to the incline of the conveyor bed, it slides off through the doorway to the storage room. The standard drive for this conveyor is from an electric motor direct connected to an enclosed spur-gear reducer which drives to the headshaft by means of a chain. This type of conveyor has eliminated all manual handling from the dump to the storage room. In cases where the tank room is located on the floor above the storage room, this conveyor delivers the ice to a spiral lowering chute.

Spiral Lowering Chutes. Chutes for lowering ice from one floor to the floor below were developed during the past year by Gifford-Wood Co. These chutes will handle the ice as fast as it can be fed to them. They also lower the ice without breakage, which is an important factor in the manufactured-ice plant.

The runway of these chutes is supported on a section of ladder-like steel frames distributed through a square steel chute, rigidly braced. The runway itself consists of an oak-slat bottom bolted to cross-pieces of the ladder-like frame. The sides of the runway are of steel plate with a 6-in.-wide rubber belt bolted to the plate.



FIG. 12 TYPICAL INSTALLATION OF COAL-HANDLING EQUIPMENT UTILIZED FOR STORING AND UNLOADING COKE

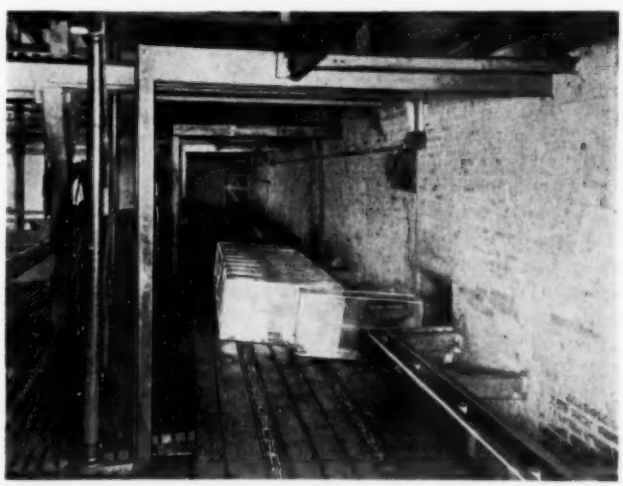


FIG. 13 CAKE CONVEYOR IN MANUFACTURED-ICE PLANT

Power Transfer Conveyors. A modern ice plant, in many

cases, requires numerous conveyors. Heretofore it has been necessary to transfer the ice by hand from one conveyor to the other, unless one conveyor was set higher than the other, in which case a gravity run was used. Gifford-Wood Co. has just developed a power transfer conveyor, Fig. 14, a unit for the mechanical transfer of the ice that can be installed between two conveyors, no matter at what angle the conveyors are set with respect to each other.

The power transfer makes it possible to synchronize two conveyors at angles to each other and eliminates the danger of ice jams and breakage at the transfer point which are liable to occur when the gravity run is depended upon to transfer the ice.

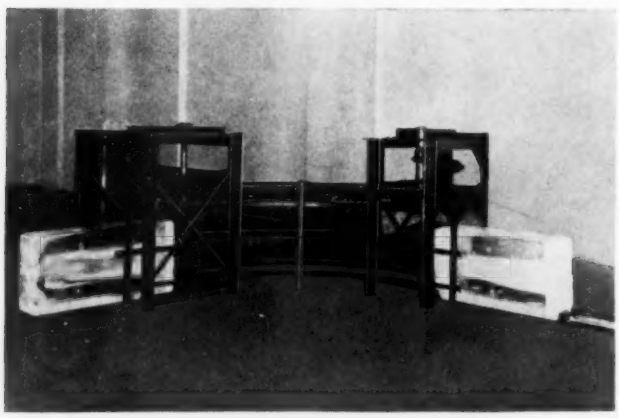


FIG. 14 POWER TRANSFER CONVEYOR FOR ICE PLANT



FIG. 15 ADJUSTABLE LEAD FOR LIVE-ROLLER CURVE

In using the power transfer, it is possible to convey the ice by means of a system of conveyors from any part of the plant to any other part without any labor beyond that of placing the ice on the first conveyor. The unit is very flexible in application, and it can be installed in any plant regardless of conditions.

The power transfer conveyor consists of a steel framework which supports a roller chain running in a horizontal plane, together with the necessary sprockets, driving mechanism, and ice guides. The chain is supported above the ice with pusher attachments extending downward, and is guided through a radial path by curved tracks. This unit is very compact and is driven from the other conveyors that comprise the installation. It is simple and inexpensive to install because it is placed entirely above the floor level.

Ship-Loading Conveyor. A recent installation of Lamson Company conveyors has been made to load bagged material into ships at a rate of over 100 tons per hour into each of two hatches, delivering the bags to any desired level in the hold and distributing them automatically over a sufficient area to enable the stevedores to stow them at this rate of speed.

The equipment is adjustable to varying sizes of ships, tidal conditions, hatch sizes, and spacings, and is also designed to convey bulk material. It consists of two automatic crane towers which receive the cargo from long belt conveyors at any point along the dock and deliver it through stainless-steel spiral chutes into the holds. The chutes are suspended from the tower booms. The equipment has effected great savings in labor and reduced the loading time of ships by approximately 50 per cent.

Developments made by the Logan Company in the conveyor field during the past year were described in a recent paper by H. R. Gotthardt,[4] from which the following items have been extracted.

Adjustable Lead for Roller Curves. To reduce the high cost of making roller curves and to make adjustments possible in the field, the Logan Company has designed a curve with longitudinally slotted holes in the outside rail for supporting the outside end of the roller shaft. Fitted on the outside of this rail is a secondary steel strip with holes to fit the shaft and hold it stationary. This secondary strip is bolted to the angle rail in such a way that it can be adjusted for loads in either direction (see Fig. 15).

Intermediate Delivering Device for Elevators. To elevate packages from the lower floor to several upper floors of a building, a recently developed intermediate discharge is used in connection with the regular simple-type pusher-bar elevator. This device, Fig. 16, consists of two endless chains connected at intervals with pushers for propelling the package over the bed and up the incline.

At each intermediate floor where a delivery is desired, a portion

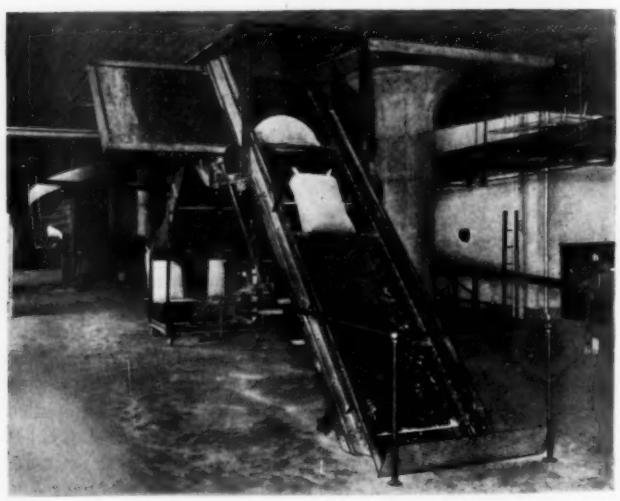


FIG. 16 INTERMEDIATE DELIVERING DEVICE FOR ELEVATORS

of the bed of the elevator is removable, that is, a panel of sufficient length to allow the largest package to pass through the bed is arranged to slide out through the side, leaving an opening in the bed. Attached to this opening is a secondary conveyor for carrying the packages away from the elevator to any point on this floor. When any panel is slid back into place, that floor becomes inoperative and the packages pass on until they come to an open gate, or until they are discharged over the top of the elevator.

Live-Roller Curve. A live-roller curve has been designed using the same principle as used on the live-roller conveyor; that is, a belt running underneath the load-carrying rolls and held

[4] Consulting Engineer, Logan Co., Louisville, Ky. Original paper contributed by the Materials Handling Division and presented at the French Lick Meeting of the A.S.M.E., French Lick, Ind., October 13 to 15, 1930.

in place by tension rolls. Through the use of ball-bearing idler rollers the belt is given sufficient twist at intervals around the curve to allow it to follow the circumference of the curve. The return belt, however, is brought back in a straight line across the chord of the circle formed by the curve. This is done by giving the belt a quarter-twist at both turning points. A take-up is arranged in the center of the return belt.

Alligator Switch. In the use of gravity conveyors, it is sometimes necessary to switch packages from a single line to two or more lines, or to receive them from two or more lines on to a single line. The usual method followed has been to place a pivoted section about 10 ft. long in the single line, this section being movable horizontally to connect to any of the multiple lines. However, when small packages were being handled, this switch section left too great an opening. A further objection was the abrupt change in direction of the rolls, which caused the packages to turn crosswise to the rolls and become jammed against the guard rail.

An "alligator" switch was therefore designed with a number of rolls at the pivot point, individually supported on steel brackets (see Fig. 17). These in turn are pivoted in the center, being tied together with a flexible lattice arrangement. When the switch section is swung at an angle to the single line of the conveyor, these individually supported rolls form a flexible portion of the switch section. They automatically arrange themselves in a radial position as regards the curve connecting the single straight



FIG. 17 "ALLIGATOR" SWITCH FOR ROLLER CONVEYORS

line of the conveyor and the switch section. This device holds the distance between the rolls to a minimum and eliminates the abrupt change in direction of the rolls, making an ideal turn for the package.

Selective Deflecting Switch. An alligator-switch section, operated electrically by means of a motor and a screw drive, has been developed to deflect packages or boxes from a main conveyor to any of several branch lines (see Fig. 18). The main conveyor here is equipped with two "floating" rolls just preceding the alligator switch and two others just following it. The branch line is also equipped with two of these floating rolls just following the alligator switch. Tote pans are handled on this conveyor. A key placed in one of the sockets on a tote pan completes an electric circuit when the pan passes over the floating rolls immediately preceding the switch section. This circuit connects with the motor on the switch section, moving the switch section to serve the branch line. As the package then passes off the switch section and over the floating rolls in the branch line, it sets the switch in the circuit for either of the two following results: If the next package is not keyed it passes over the first floating rolls and the motor for the alligator switch reverses, throwing the switch in a straight line with the main line of the conveyor. Or, if the succeeding box is also keyed, it passes over the first floating rolls without effect on the motor or the alligator switch.

It is necessary, of course, to have the packages spaced at least 2 ft. or 3 ft. apart. Where this cannot be done manually,

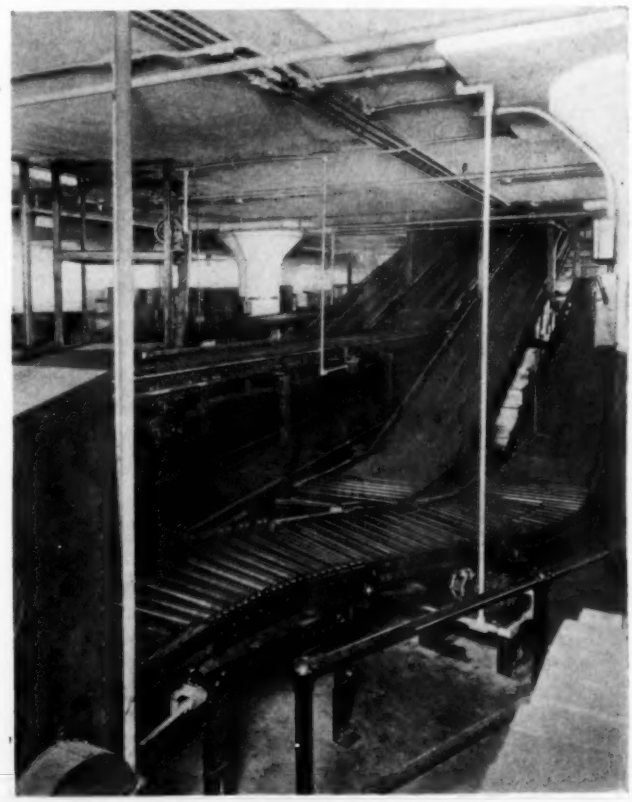


FIG. 18 SELECTIVE DEFLECTING SWITCH FOR GRAVITY ROLLER CONVEYOR

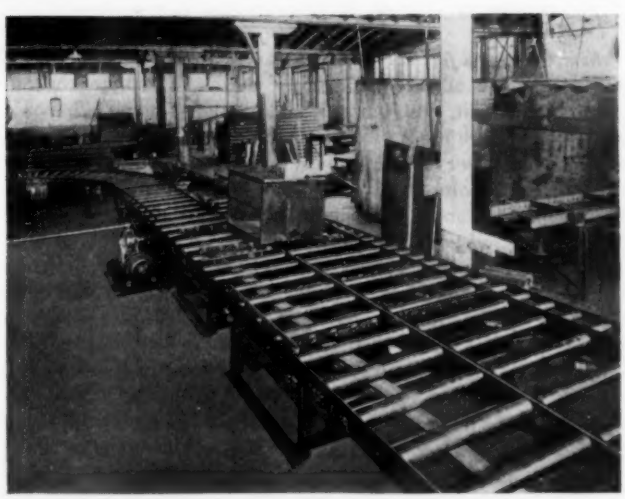


FIG. 19 ELECTRIC MULTIPLE-SWITCHING DEVICE

it is done electrically by placing in the main line, directly preceding the switch section, a power spacer. This device auto-

matically spaces packages, speeding up the first one enough to place it the proper distance ahead of the package following.

To keep the alligator switch from moving before a package is off it, or before it is ready to be moved for the next package, the circuit connecting the motor which operates it includes two recording switches. These switches are so arranged in connection with the floating rolls that a package is counted in to the deflecting zone and again counted out as it leaves it, the switch being inoperative until any package which has been counted in, is counted out.

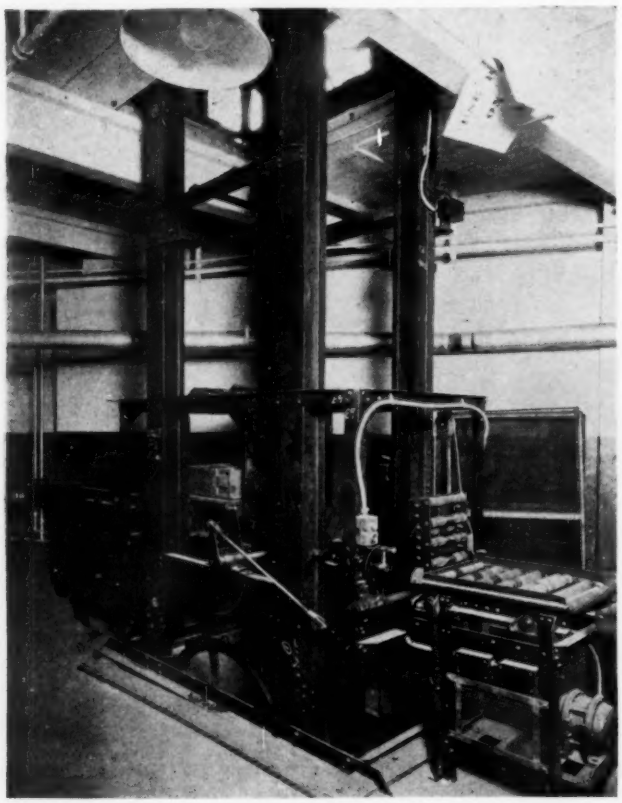


FIG. 20 AUTOMATIC SELECTIVE VERTICAL ELEVATOR

Remote Control for Alligator Switch. It is sometimes advisable to operate the alligator switch by means of remote control. An electric circuit, push buttons, and a series of signal lights on a control board are used. To move the switch to any other position it is necessary only to push a button for that particular station. The motor that operates the alligator switch is equipped with a solenoid brake, and suitable limit switches are placed in the path of the switch section to stop it at the proper points.

Electric Multiple-Switching Device. There has been designed and built an actual full-size working model of a selective, multiple-switching device, Fig. 19, by means of which it is possible to use as many as 300 or 400 selective stations. This device is operated electrically by means of a given setting of keys on some part of the package being conveyed. It is arranged with disappearing rolls placed in the main conveyor line at the deflecting point and is operated electrically. A package traveling along the main line and keyed for any certain station, raises a roller stop in its own path. This brings the package to a standstill directly over the disappearing rolls. As soon as the package comes in contact with the roller stop it closes a circuit, which elevates the disappearing rolls and the package with them. The disappearing rolls rise on an incline at right angles to the path of the package on the main conveyor, and therefore the package rolls off onto the branch line.

As soon as the package is off the main line and on the branch line, the roller stop and the disappearing rolls sink below the carrying surface of the main conveyor. Virtually any number of these selective stations may be inserted in the main roller-conveyor line. A package not keyed for intermediate stations will pass over these stations without effect, until it reaches the station for which it is keyed.

Automatic Selective Vertical Elevator. The automatic selective vertical elevator, as shown in Fig. 20, is equipped with a dial at each loading station on each floor. These dials are so arranged that they can be set for a discharge station on any floor. The operation is electrical throughout. The elevator proper consists of two endless chains connected at intervals by swinging tray carriages traveling upward on the loading side.

Some products handled, such as grinding wheels, are stacked up in piles on a flat pallet, presenting a package that must be moved very carefully to avoid spillage of the wheels. A roller conveyor leading into the loading station of the elevator on any given floor



FIG. 21 AUTOMATIC RECIPROCATING ELEVATOR

holds a pallet loaded with wheels ready for the elevator. The operator on the floor where this pallet is waiting to be picked up, sets his dial to correspond to the floor where the package is to be delivered. The remainder of the operation is automatic. Before reaching the loading station the empty carriage, moving upward, makes an electrical connection. This action automatically starts a motor by means of which the waiting package is automatically and positively moved into the pick-up station. If an ascending carriage is not empty, that is, if it carries a package from a lower floor, the particular station which is waiting to be loaded is inactive and the package does not move into the elevator.

The setting of the dial automatically sets the key on the car-

riage that is carrying the load. Therefore, when it reaches the proper discharge station on its downward travel, it automatically throws that station in and deposits the package there. The discharge is positive by means of a power conveyor. As soon as a package has been picked up from a loading station, or has been delivered at an unloading station, these stations immediately return to an inoperative position out of the path of the next carriage.

Automatic Reciprocating Elevator. The automatic reciprocating elevator, Fig. 21, as now developed, usually serves two floors, an upper and lower. Its purpose is lowering packages. When there are no packages to be lowered the elevator is inactive. However, as soon as a package reaches the loading station at the top it automatically starts the carrier down to the first floor. At the first floor the package is discharged automatically, and, when the load is cleared, the elevator automatically reverses and travels back up to the top position ready for another package. This operation continues as long as there are packages to be lowered. The same automatic type of reciprocating elevator

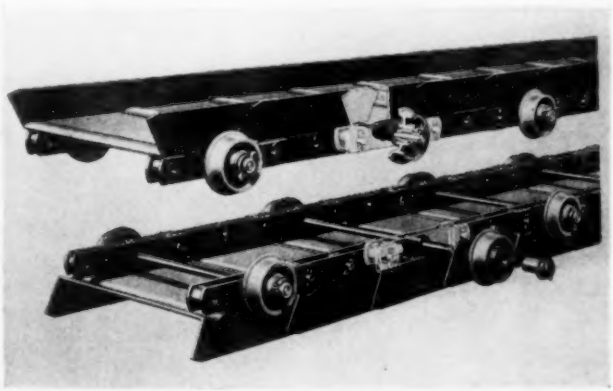


FIG. 22 LEAKPROOF APRON CONVEYOR WITH EQUALIZING SADDLE CONSTRUCTION

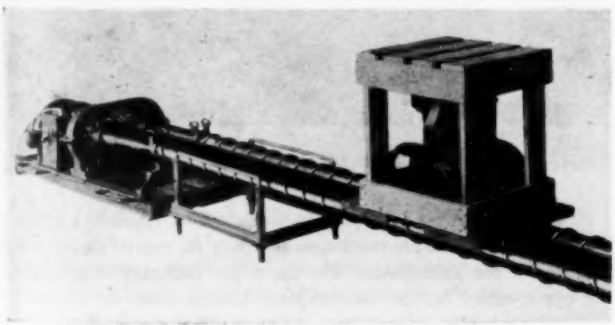


FIG. 23 TWINVEYOR USING SPRING-SUSPENDED RACK AND PALLET FOR HANDLING MISCELLANEOUS COMMODITIES

has been worked out for elevating as well as lowering, and for taking care of two stations as well as one.

Apron Conveyors. Another important development in the conveying industry has been the introduction, by the Chain Belt Company, of "Rex Leakproof" apron conveyors with equalizing saddle construction. Generally speaking, apron conveyors have always been made up with double strands of either malleable-iron or steel chains, the apron pans being attached to the inside sidebars of the chain, and the carrying rollers being placed at the articulation points of the chain between the sidebars. With this standard type of apron-conveyor construction, eccentric loads were set up on the inside sidebars of both strands of chain. It was also exceedingly difficult to construct the pans so that they would not leak. Moreover, in making repairs it was necessary to disassemble the complete apron, thereby losing valuable production of time.

For the Rex conveyor equalizing saddle construction, Fig. 22, the apron pans, instead of being attached to only one side of the chain, are fastened to an equalizing saddle. This saddle rests directly on the chain, being held in place either by through rods or pins extending through the chain and saddles. The load is



FIG. 24 TWINVEYOR TRANSPORTING AND STACKING SUGAR BAGS (1000 PER HR.)

therefore carried directly along the center line of the chain, and hence is distributed equally to both sidebars.

The chain draws straight and true, and its life is practically doubled. The saddle and through rods, together with the large-diameter outboard rollers, are entirely independent of the chain joint, and therefore the chain is called on to take tension loads only. As the dead load is practically divorced from the pulling medium, the natural wear occurs on the tread and bore of the hard white-iron outboard rollers and the hardened steel bushing sleeves which are locked to the ends of the rod. These parts can be readily and quickly replaced without disturbing the chain or apron assembly in any way, thereby cutting hours off the usual time required for apron-conveyor repairs.

Twinveyors. The Clark Tructractor Company has placed on the market a conveyor, escalator, or tiering machine (Figs. 23 and 24) of unique design for a variety of handling operations. The operating mechanism consists of two sets of tubular external screws driven toward each other by a power head. The direction of travel is reversible. The conveyor is sectional, portable, can be set up or taken down rapidly, and functions in any normal position. A change in the angle of deflection is possible at each intersection, the maximum angle being 15 deg. at any horizontal

or ascending joint and 10 deg. at any descending joint. Packages can be carried up grades as steep as 30 per cent.

Tubular screws come in lengths of 8 ft. each. The total length of the standard unit is 50 ft., consisting of five 8-ft. dual

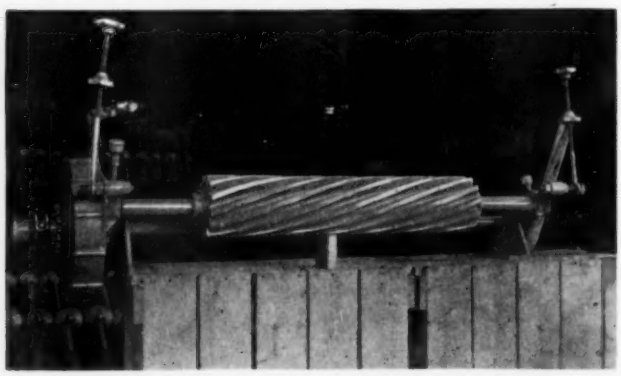


FIG. 25 RUBBER BRUSH FOR CLEANING CONVEYOR BELTS

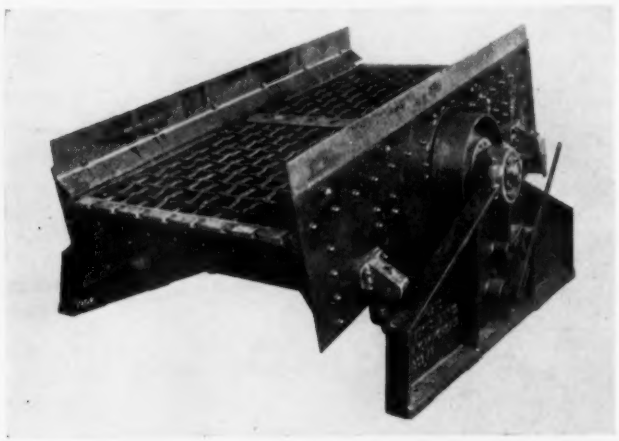


FIG. 26 IMPROVED SIZING SCREEN FOR USE WITH BELT CONVEYORS

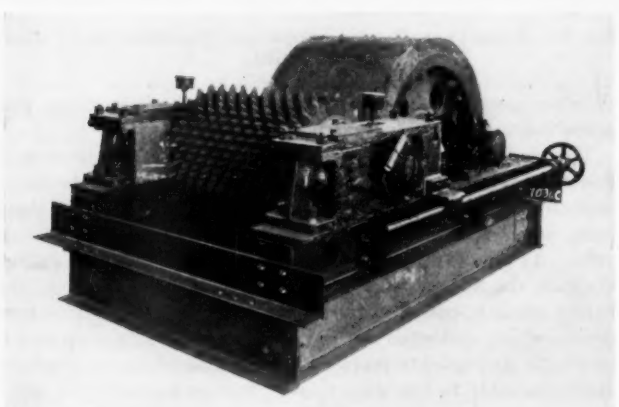


FIG. 27 COMBINED FEEDER AND CRUSHER

sections, two 4-ft. single sections, and a power head. If more than one unit is employed, a hurdle section carries packages across the intermediate power heads. Center distances between tubes are adjustable within a range of 6 in. minimum, for use with bags, up to 12 in. maximum, for large packages and bales.

Weights up to 100 lb. can be handled, and the traveling speed of the package is 90 ft. per min. Pallets, as shown in Fig. 23, pivoted to run between the tubes, are provided to handle boxes, crates, and bales which otherwise would be difficult to transport and would tend to leave the conveyor.

Right- or left-hand intersections are made possible by the use of spiral gravity chutes composed of bars curved in the direction of travel. The chutes are designed to set over power heads or tubes at any desired point.

A special hand truck, or wheel carrier, can be provided in which to move the power head. The power head is mounted on a wood base and can be kept on the truck set on the floor or set on top of a pile of bags or boxes, for stacking or unloading.

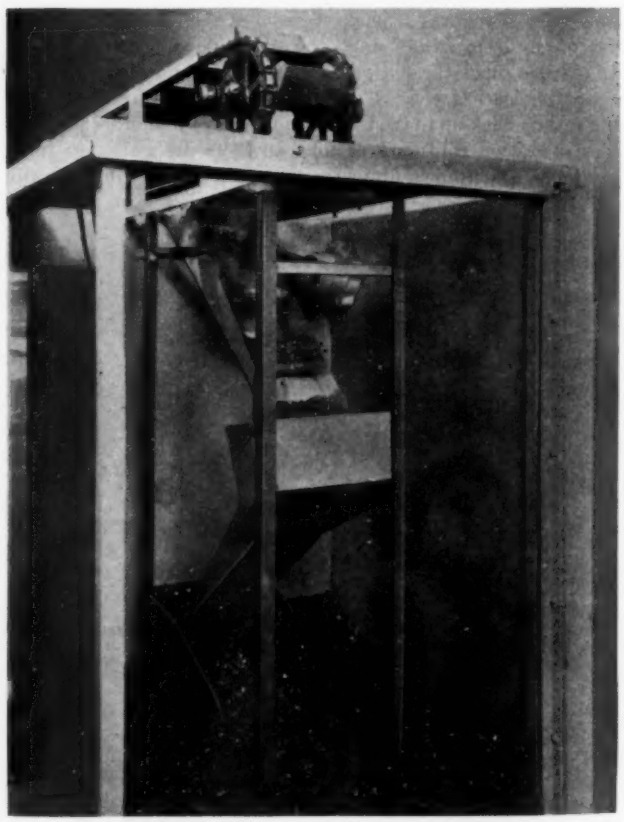


FIG. 28 "AVALANCHE" CHUTES FOR LOWERING FRIABLE MATERIALS INTO BINS

Equipment for the Foundry Industry. The Eastern Corporation has made numerous improvements in materials-handling equipment for foundries. The foundry industry has offered the opportunity for an outstanding development in the use of ball-bearing roller conveyors. Although this type of conveyor equipment has been employed in foundries for several years in a comparatively minor way, a wider range of use is now evident because of the availability of vastly improved types, quality suited to the individual application, and a more comprehensive knowledge of foundry requirements in process and arrangement.

Foundrymen have long claimed that such equipment could not be employed for the transportation of green-sand molds of intricate pattern design and core assemblies. The development of the sand-sealed ball bearing, however, which can be operated for years without relubrication, great precision in construction to provide vibrationless transit, and the ability to withstand the continual spillage of molten metal, have furnished in roller conveyors a simple and easy method of movement and a flexible mobile storage system for molds being accumulated for heats.

Other accessories to this equipment have been developed, such

as roller-conveyor discharge mechanisms adaptable to existing hand-discharge roll-over molding machines and adjustable to the pattern draw, various fixed and movable lateral-transfer mechanisms, and vibrationless turntables, as close-coupled direction changers, all constructed with sand and molten-metal protection features.

Standardization of widths and parts has been worked out for equipment to handle a wide range of sizes, types, and weights of molds, so that a single fixed width of conveyor can be utilized throughout an entire plant that may be engaged in making a great variety of work. This feature provides for periodic changes

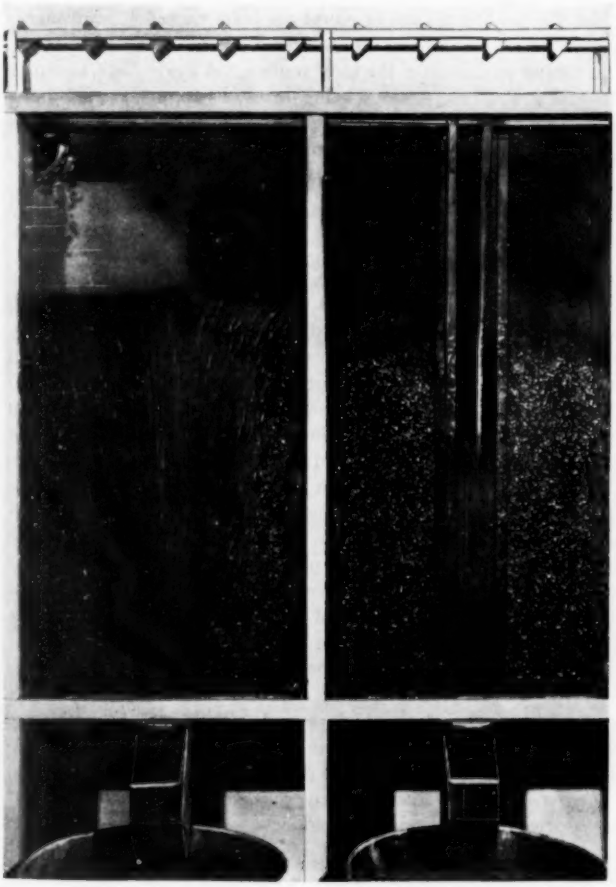


FIG. 29 "WITHDRAWAL" CHUTES FOR REMOVING FRIABLE MATERIALS FROM BINS

in arrangement, as dictated by changes in character or process of work, with practically no obsolescence.

These methods of mold transit provide the means for grouping and concentrating the work, to restrict pouring of metal to certain areas or zones in which safe and speedy overhead ladles travel over only short distances. The danger of accidents is thereby reduced.

The great flexibility of process and movement thus provided by these newly developed transit methods in the foundry has made it possible to produce a large miscellany of types and sizes of castings, in small quantities and requiring frequent pattern changes, by methods similar to those of straight-line production, and controllable by conventional planning methods. Applications of the roller-conveyor equipment have already been made to direct pouring into molds from the furnaces.

In the cleaning and finishing of castings, similar equipment



FIG. 30 CHAIN CONVEYOR IN AUTOMOBILE PLANT

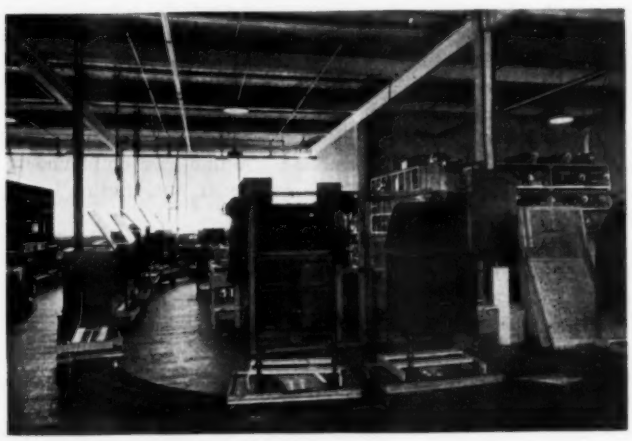


FIG. 31 SINGLE-STRAND DOLLY OR TRUCK PUSHER CONVEYOR OF THE "ON EDGE" TYPE

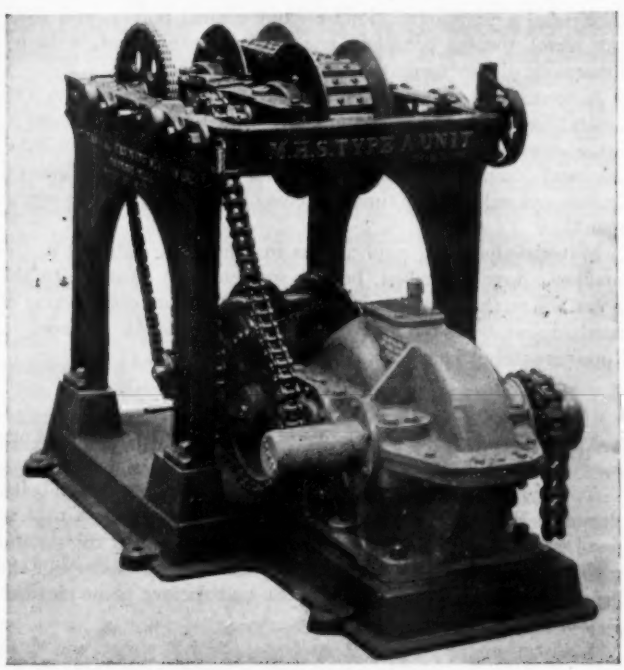


FIG. 32 (RIGHT) STEEL V-BELT VARIABLE-SPEED TRANSMISSION

and a like character of arrangement have provided the same order lines, sequence of process, and speedy manufacturing cycle as has been brought about in molding operations, with substantial reduction in semi-processed inventory, and coordination in pace for all foundry departments. These general systems, in many cases, have more than doubled the output per square foot of floor area.

Such systems have been made highly adaptable to simple and effective wage-incentive, piece-rate, and bonus plans. They have also brought about an easily maintained cleanliness, never before quite possible in the general run of the foundry industry.

Power conveyors have been developed, of higher quality, greater capacity, and wider utility for the handling of molding sands. Molding sands are now quickly removed from the foundry molding sections, screened, cleaned of metal and dust, automatically tempered, mixed, bonded, and carried to desired points in quantities and qualities to suit the various products—entirely by mechanical means that require the attention of but few individuals, and with much lower horsepower requirements.

To meet higher-capacity requirements without proportionately higher initial costs, higher speeds in conveyors and elevators have been attained without increasing the wear and tear or reducing the life of the equipment. Basic materials of higher durability to resist abrasion, and of greater strength, have been incorporated, with corresponding changes in design.

Fabrication by electric arc welding has been further developed, with the use of machined jigs for accuracy, uniformity, and symmetry of outline in assembling. A universal employment of interchangeable, self-adjustable, protected, anti-friction bearings has likewise come about.

Among the newer features are: integrally mounted, completely enclosed drives, motors, and controls; reduction of noise in operation; and lowering of breakdown hazards.

Improved sand-processing and conditioning accessories of the continuous type are now incorporated in the continuous stream of movement and coordinated with the general conveyor system. Better lubrication and adequate exhaust systems for dust abatement further improve the equipment.

Simplified methods of operation have resulted from the greater employment of high-class electrical controls, with methods of push-button operation by interlocking and grouping of units, to obtain complete functioning from central points, and from the trend toward complete enclosure of equipment, with oil immersion of moving parts.

A consistent, general standardization in design of parts and assemblies, with universal punching and uniform bolt sizes in attachments, connection plates, and flanges, and uniform widths, diameters, and sizes of frame members, has increased the general utility and reduced the time and cost of installation of the equipment.

Materials-handling applications in foundries, installed in the manner above described, have been generally effective to the point where the total labor percentage of the net dollar value of product has been reduced from an average of 48 per cent to approximately 25 per cent.

Belt Conveyors[8]

A number of useful developments have occurred in the belt-conveyor field during the past year.

Rubber Brush. The rubber brush for cleaning conveyor belts, shown in Fig. 25, is mounted on a pipe core carried on a rapidly revolving shaft. The brush proper is made of tough rubber and resembles a spiral gear of very wide face. The periphery of the brush is brought in contact with the belt surface to be cleaned. The rubber brush has demonstrated itself to be a decided improvement because of its ability to resist abrasion.

Holdback Rig. A new type of self-contained holdback rig has been developed to prevent a belt conveyor from running backward under load when the power is shut off. This rig consists of a ratchet wheel, on the hubs of which is mounted a yoke carrying a pawl. By means of springs rubbing on the faces of the ratchet wheel, the pawl is held free when the conveyor belt is moving forward and is forced into the ratchet when the conveyor belt starts running backward. This rig is a very inexpensive and effective device.

Rubber-Cushioned Idler. A rubber-cushioned idler is used under the loading points of conveyor belts carrying large pieces of heavy material which are delivered to the belt with impact. The rubber cushions on the idler pulleys act as resilient bumpers protecting the belt against being bruised or punctured by the lumps of material. The effectiveness of this unit is demonstrated by a pile-driver test, using a pointed 50-lb. weight. The weight, when dropped 18 in., will puncture an eight-ply belt supported on a plain cast-iron idler pulley, but the weight can be dropped 6 ft. without puncturing the belt when the latter is supported by a rubber-cushioned idler pulley.

Five-Pulley Idler. There is a tendency on the part of some users to abandon the type of belt-conveyor troughing idler having one horizontal and two inclined pulleys, which has been very popular during the past six or seven years, in favor of an improved design of idler having one horizontal pulley, two pulleys inclined $11^1/_4$ deg. and two pulleys inclined $22^1/_2$ deg. This new design includes such features as malleable-iron brackets, pulleys of equal length, angle bases, and improved bearing mountings.

Screens. An accessory which is of prime importance in connection with conveying equipment is a screen for sizing the product conveyed, especially in the case of coke, stone, sand, and gravel. A very popular form of screen is one having inclined screening surfaces which are given a very rapid gyrating motion of small radius. A screen of this type is shown in Fig. 26. This type of screen has been in use for a number of years, but the past year has covered a period of development of details resulting in the elimination of trouble from the great stresses set up by increasingly high speed. The improvements are principally in the design of the screen frame and the mountings on the shaft producing the gyrating motion.

Feeder-Crusher. Feeders and crushers are very important accessories to conveying machinery. Fig. 27 shows a combination of these two units, as developed for service in a plant where coal is received at times in a frozen condition. A large drum, or roll, which has a smooth face, except for widely separated small projections, turning at slow speed, serves as a feeder roll and also as a backing plate for the second roll, which is toothed and revolves at high speed. Thus coal is drawn in over the large roll and crushed between the large roll and the small roll.

Two-Motor Drives. In the past, the practice in connection with tandem drives on belt conveyors (where two pulleys are used to drive the belt) has been to gear the two pulleys together with gears which are somewhat larger in diameter than the pulleys, and to drive one of these gears, which, in turn, drives the other. There is a creep in the belt as it passes around these pulleys which is not the same on both pulleys. With the advent of extremely heavy duty came the problem of relieving the belt of the stresses set up by this belt creep. For this purpose, in place of gearing the two pulleys together, separate motors for each shafts are used. Under certain conditions it is advantageous to use motors of the same power, but under other conditions the motors may be of different sizes. Both of these arrangements are now in successful operation.

[8] Section prepared by M. J. Andrada, Chief Engineer, and E. R. Morgan, of the Robins Conveying Belt Company, New York, N. Y.

Adams Chutes. Conveyor accessories which have come into prominence recently are shown in Figs. 28 and 29. These are known as "avalanche" and "withdrawal" chutes, respectively, and are used in lowering friable material from conveyors into bins, and withdrawing the material from the bins. The photographs are those of models because, obviously, it is impossible to get such pictures of actual installations.

Fig. 28 needs very little explanation. Material delivered by the conveyor builds up on angle baffle plates so as to cover the bottom plates of the chute, and sloping avalanche plates still further baffle the material, until so much has accumulated in the chute that an avalanche takes place, permitting a large flow of material instead of the usual continuous flow. The actual savings in degradation due to this action have been compiled by several users and are quite remarkable.

No less noteworthy are the savings in degradation effected by the use of the "withdrawal" chute, shown in Fig. 29. In this photograph, two bins with bottom openings are shown in miniature, the one on the right being equipped with a withdrawal chute. The near side of the bins is a glass plate, and the photograph was taken with the gates open, so that the flow of material is shown. A large turbulent movement of material is seen on the left, which causes attrition of the material as it crowds down to the small opening. With the "withdrawal" chute, shown on the right, a vertical column of material moves down through the chute, being constantly fed from surface material only, which rolls in from the top and never below the top.

Chain Conveyors[6]

Progress in the chain-conveyor field has in general been confined to improvement of design detail and to the more extended use of existing types of equipment. There is appearing also a tendency to make the conveyor a definite part of the production process, as has already been done in the automotive field (see Fig. 30). The idea contained in the continuous molding conveyor, now so generally used in all large production foundries, is being applied to other processes, and it would appear that this development should have further and more intensive study.

Of the different types of chain conveyors available, the power-driven overhead trolley conveyor has undoubtedly found the greatest extension in application. This is probably due, in part, to its relatively recent development, but of even greater importance is its extreme flexibility, making it applicable to all sorts of handling problems—transportation, processing, and assembly.

Following the lead of the overhead trolley conveyor is the single-strand dolly, or truck, pusher conveyor of the "on edge" type, i.e., the chain operating with the pins vertical. This design, an application of which is shown in Fig. 31, uses a chain flexible in both directions, similar to the trolley conveyor. Being confined in its track by suitable attachments, and guided around horizontal turns by rollers, it can be curved or turned in either plane, run from floor to floor, and, if desired, can be made a continuous loop. Due to this flexibility of path, this type of construction may replace one or more of the conventional straight-line conveyors, thereby avoiding transfer points and allowing greater variety in layout. Its use has been largely in the automobile-body plants for assembly, paint, and trim, but installations have also been made in radio-cabinet, office-furniture, and refrigerator assembly, and in stock rooms, etc. A modification, with the load resting on and carried directly by the chain, is beginning to replace the gravity roller conveyor in some instances, notably in the dairy industry.

[6] Section prepared by N. H. Preble, Consulting Materials-Handling Engineer, Detroit, Mich. Mem. A.S.M.E. Illustrations for Figs. 30 to 32, inclusive, were furnished by Mechanical Handling Systems, Inc., Detroit, Mich.

Progressive assembly on chain conveyors, universally employed in automobile plants, is now being applied to a greater extent in other plants, particularly in the newer industries, such as radio, electric refrigeration, and the like. Its further use in other and older fields will probably be noted in the near future.

Another field for the chain conveyor which has had considerable extension during the year is in connection with high temperature in such processes as heat treatment, porcelain enameling, core baking, billet heating, etc. Various alloy chains are now available that will operate in high temperatures. Other designs, especially of the trolley type, provide for the protection of the conveyor from excessive heat.

In the various equipment parts available there have again been no radically new developments. No new chains of importance have been offered, and it would appear that progress might well be in the other direction: in the standardization on a fewer number of types and sizes, with detail improvement in those few. Greater use is being made of all-steel forged chain, not only for trolley conveyors but also for other chain conveyors, and, within proper limits, it appears that this tendency will become more notable in the future.

Greater attention is being given to the details of design which affect maintenance, especially with regard to lubrication. Centralized pressure greasing is largely employed on oven conveyors, and its use is extending to other constructions. There is some tendency toward the use of anti-friction bearings, not on account of decreased friction so much as because of the less frequent greasing required. Trolley wheels have also been improved, and can now be obtained to operate in high temperature, acid fumes, dust, etc., and can be equipped for automatic lubrication.

The chain conveyor naturally includes a power or driving unit, of either constant or variable speed, and, ordinarily, with a considerable gear-reduction ratio between the motor and the conveyor headshaft. The open gear train to obtain this reduction has become obsolete, except for extremely light-duty work, all drives now being fitted with some form of fully enclosed speed reducer. In this connection, the more frequent use of the herringbone gear reducer is noteworthy. These reducers are now available from stock in a large range of sizes and with standard ratios as high as 300 to 1. Variable speed is ordinarily required in all chain conveyors except those used purely for transportation, and, even there, speed variation may be desired in order to synchronize with other equipment. This variable speed in almost universally obtained by the use of the V-belt type of transmission, with infinitely adjustable cone pulleys. During the past year a modification of this transmission has been introduced by Mechanical Handling Systems, Inc., in which the V-block rubber belt is replaced by a steel belt with slidable, laminated steel plates which engage with raised sections on the cones (see Fig. 32). This design gives a transmission with positive drive and one which is fully enclosed, operating in a bath of oil.

Mention should also be made of the greater use of electric arc welding in the fabrication of chain-conveyor framework, giving marked economy and some decreased weight when compared with riveted construction. It is believed that this development will grow, and that the near future will see practically all conveyor structures welded.

Drive Chain, Variable-Speed Gears, and Skip Hoists[7]

Drive Chain. Bucket elevators and conveyors, slat and apron conveyors, foundry conveyors, skip hoists, and related equipment

[7] Section prepared by W. W. Sayers, Chief Engineer, Link-Belt Company, Chicago, Ill. Mem. A.S.M.E.

have not recently undergone any radical changes. Their field of application has been considerably broadened, however, by improvements and refinements in their construction. The chain, which is the principal medium used by these types of equipment for transferring, elevating, and conveying materials, has been greatly improved in its two major requirements—strength and durability.

Until recently, chain manufacturers were dependent upon mild or medium steels with no heat treatment. Medium steel was preferable to mild steel because it had slightly higher physical properties. The fact that low-carbon steel could be surface hardened by soaking in a carbon atmosphere has been known for years, this operation being called case-hardening. It did not contribute to strength, however, but only to durability against wear. By heat treatment the strength of medium steel

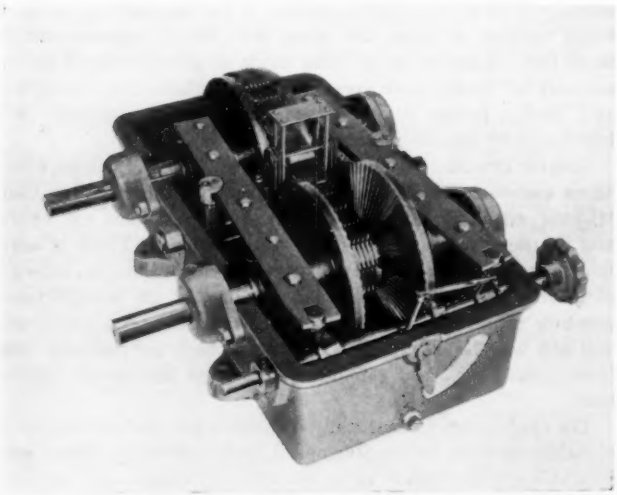


FIG. 33 POSITIVE-DRIVE VARIABLE-SPEED GEAR

is now raised from 75,000 to 95,000 lb. per sq. in. The strength of mild steel is not increased by treatment.

Medium-carbon steel bars having an analysis of:

Carbon	0.35 to 0.45
Manganese	0.50 to 0.80
Phosphorus, less than	0.04
Sulphur, less than	0.04

give a desirable leeway in design of steel-chain sidebar links, whether heat treated or untreated. The pins, or bushings, should be of either mild steel, case-hardened, or of medium steel, heat treated.

Recognizing that cumbersomeness and weight in chains are undesirable, and frequently prohibitive, for use with materials-handling equipment, manufacturers now use steels containing alloying elements, which, with suitable heat treatment, provide equivalent strength with considerably reduced size and weight.

Standard carbon steel (S.A.E. 1015-1020-1045), nickel alloy steels (S.A.E. 2320-2345), and chromium alloy steel (S.A.E. 3135-3245), chromium alloy steel (S.A.E. 5140), and molybdenum alloy steel (S.A.E. 4615), all react uniformly well to the heat-treatment control used in production methods, and are readily obtainable.

The actual example, shown in Table 1, of the development of a single size, or pitch, of drive chain used by the oil industry with rotary-type well drills, serves to illustrate the increase in strength, accompanied by a relative decrease in weight, of chains applicable to chain conveyors and drives.

TABLE 1 INCREASE IN STRENGTH OF DRIVE CHAIN FOR OIL-WELL DRILLS

Chain No.	Weight, lb. per ft.	Strength, lb.	Strength per lb.
103	4	9,600	2400
4103	5.7	22,000	3800
SS-40	6.6	28,000	4250
SS-40 Plus	6.6	40,000	6100
Hyper SS-40	8.1	75,000	9200
124	6.6	15,000	2200
4124	8.5	33,000	3880
SS-124	12.7	58,000	4570
SS-124 Plus	12.7	85,000	6700
Hyper SS-124	18.5	150,000	8100

Another steel chain of the drive type has recently been developed having a strength of over 11,000 lb. per pound of weight per foot of length, or almost five times as great as that of malleable iron, with corresponding increase in uniformity and reliability.

Cast chains of malleable iron are extensively used with elevators and conveyors because they are inexpensive and readily provided with special attachments, cast as a part of the link.

Cast chains having a hitherto unattainable combination of high tensile strength and great wear resistance, have been recently developed, and are now available. These chains are admirably suited for drives, elevators, conveyors, etc. A high-grade malleable iron is processed in such a way that it becomes practically a different metal, with vastly improved qualities.

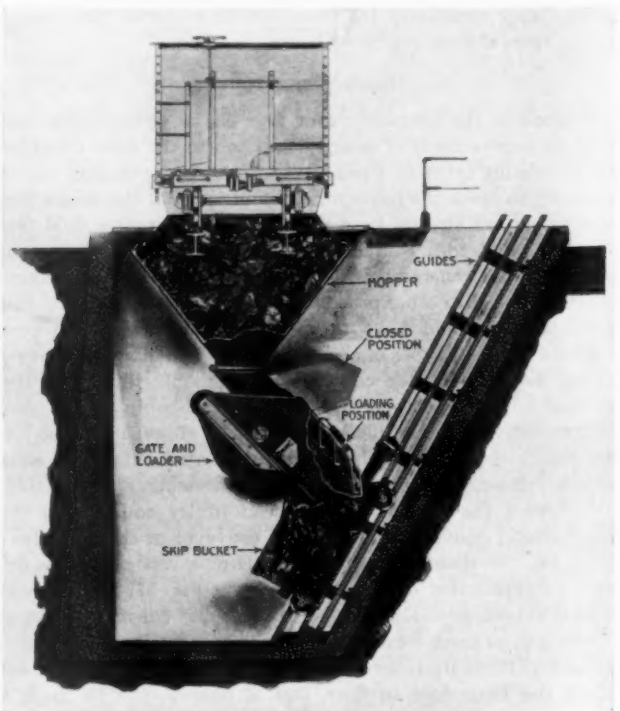


FIG. 34 AUTOMATIC LOADER FOR SKIP BUCKET

This new method of processing produces a metal with a perfectly balanced combination of the three qualities desirable in a sprocket chain—great toughness to resist tension and shock without permanent stretch; great strength in proportion to weight and bulk; and characteristics to resist abrasive wear.

Tests of this new metal, furnished in Table 2, afford an interesting comparison of its characteristics with those of other cast-chain materials.

Due to the structure of this new metal it resists abrasive wear far better than the usual cast chain. Worthy of mention is its exceptionally high yield point (25 per cent higher than that of

TABLE 2 PHYSICAL CHARACTERISTICS OF MALLEABLE-IRON CHAIN

Item	Processed malleable	Average malleable iron
Average yield point, lb. per sq. in.	45,000	36,000
Average ultimate strength, lb. per sq. in.	65,000	54,000
Brinell hardness	170–190	110–130

average malleable iron), which engineers recognize as of greater importance in chains than ultimate strength. In addition to having this high yield point, the metal is tough and ductile, and therefore better withstands shock and fatigue loads.

Cast steel of higher carbon content, and manganese steel having higher yield point, strength, and hardness, are available, but these are more expensive and require special patterns, whereas the new processed chains can be made at any malleable foundry having processing equipment, from standard patterns, at a cost slightly more than that of good regular malleable chain.

The improvements in wrought-steel and cast chains mentioned above greatly enlarge their field of application by making available chains of reduced weight having much greater strength and reliability. Weight and strength are exceedingly important factors with long or high conveyors and elevators.

Variable-Speed Gears. Conveyor and elevator driving machinery and terminals have been greatly improved by substituting speed reducers and anti-friction-type bearings in place of open gearing, and by shafts provided with plain pillow blocks distributed on separate supports. The constant attention required for lubrication is practically eliminated, and much more dependable operation and service are provided.



FIG. 35 TROLLEY MECHANISM OF AUTOMATIC TROLLEY BUCKET SYSTEM

A positive, infinitely variable-speed gear, shown in Fig. 33, requiring only a minimum of space, has been recently introduced by the Link-Belt Company. This speed gear can be readily and advantageously used for driving any elevating and conveying equipment whose speed of travel must be occasionally or frequently varied to obtain the best operating and production results. The positive-drive gear consists of a specially constructed chain and pair of sprockets, by means of which infinite variations in speeds may be obtained continuously (without steps) while running, and in ratios up to 6 to 1. The entire mechanism operates in an oil bath, within a fully enclosed housing. Due to its small size and other features, it is also used for driving machine tools and equipment of all kinds where the speeds must be positively and infinitely varied from time to time.

Skip Hoists. The winding machine for skip hoists has recently been improved by providing it with a welded steel housing, of great shock-resisting properties, in which cut herringbone gears operate noiselessly in an oil bath. The shafts for gears and drum are carried in anti-friction bearings. The rope sheaves, which usually are located in rather difficult or inaccessible positions, are also provided with anti-friction bearings. The steel housing of the machine is built integral with a large substantial steel subbase, on which the motor and traveling cam limit switch are mounted, thus insuring and maintaining correct alignment of all parts.



FIG. 36 VIEW SHOWING COMPLETE INSTALLATION OF AUTOMATIC TROLLEY BUCKET SYSTEM

A unique, clever, and simple loader which automatically controls the filling and operation of the skip bucket was developed during the past year. With this device, which requires no weighing mechanism, the bucket is hoisted only when it is fully loaded. It remains in loading position until filled, when it is immediately hoisted and continues to be hoisted, dumped, lowered, filled, and hoisted, as long as sufficient material is available to load the bucket. This automatic loader insures maximum hoisting capacity, and no attention from the operator is needed to start and stop the equipment. (See Fig. 34.)

A modified form of skip hoist, known to the trade as the trolley bucket, which has been generally used by wholesale and retail coal and material yards, has recently been improved by making all its operations entirely automatically controlled. The bucket is automatically filled, hoisted, trolleyed, lowered, and

emptied at the desired point; then rehoisted, trolleyed back, and lowered for loading at the track hopper. The attendant has only to adjust the selector dial and push the start button to accomplish the automatic repetition of the complete cycle until the stop button is pressed.

The equipment consists of separate motors and drums, electric control apparatus, a selector governing the points at which the bucket is lowered and emptied, and a loader for filling the bucket automatically. The trolley mechanism is shown in Fig. 35, and a complete view of an installation in Fig. 36.



FIG. 37 TERMINAL OF PNEUMATIC TUBE FOR SMALL MESSAGE WORK

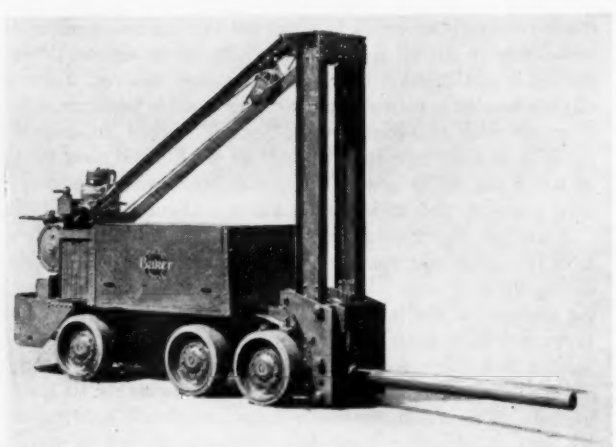


FIG. 38 8000-LB. RAM TRUCK WITH ARTICULATED FRONT AXLE

PNEUMATIC TUBES[8]

After many years of experimenting, the Lamson Company has recently introduced what is called the "Pneu-Art terminals" for pneumatic tubes. Former types of terminals were constructed of heavy metal. The new types are made to harmonize with the decorative surroundings in stores, offices, banks, and other business and manufacturing establishments. Colors are made to correspond with the office trim.

The terminals (Fig. 37) are constructed of a light metal, are rectangular in shape, and both receive and dispatch the pneumatic-tube carriers. They can be placed in fixture panels and therefore are practically hidden. Doors are spring-closed and fitted with felt gaskets to operate silently. Sizes correspond to those used for former types of installations, with which the new equipment is interchangeable.

HAND LIFT TRUCKS AND PORTABLE ELEVATORS[9]

Hand Lift Trucks. The outstanding development in the hand-lift-truck field during the past year has been its greatly increased use in connection with the shipment of goods on skid platforms. The growth of skid shipping has continued in the paper industry until nearly all customers specify skid packing, due to the saving in cost and greater ease in handling. In the larger cities, most trucking contractors have equipped themselves with hand lift trucks to handle skidded shipments consigned to printers and jobbers. An increasing variety of automotive assemblies are now shipped to assembly plants on skids. Skidded shipments

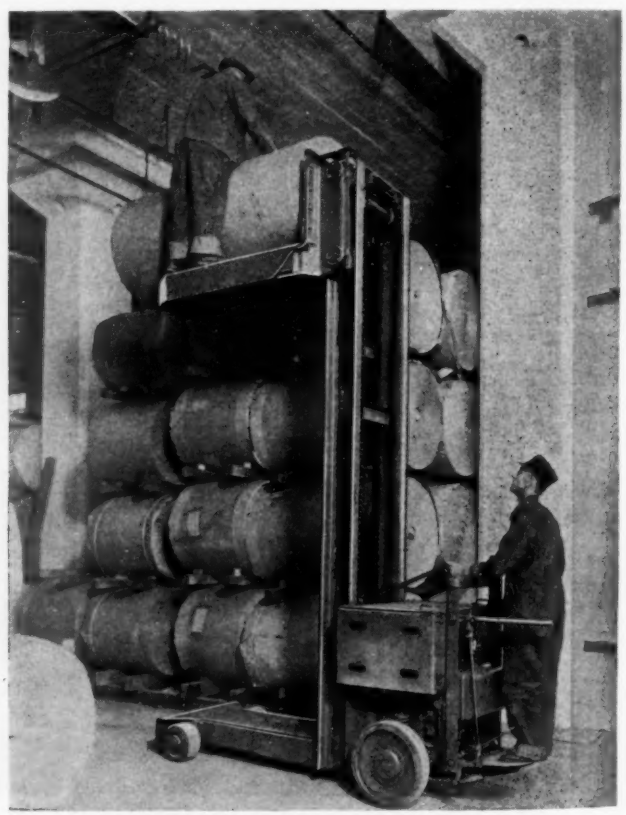


FIG. 39 6000-LB. HIGH-LIFT TRUCK

[8] Section prepared by C. S. Jennings, Chief Engineer, The Lamson Company, Inc., Syracuse, New York. Mem. A.S.M.E.

[9] Section prepared by F. J. Shepard, Jr., Treasurer, Lewis-Shepard Company, Watertown, Mass.

of slate, firebrick, canned goods, and other similar commodities are now being transported both by rail and water at a substantial saving in packing, handling costs, and damage to goods.

The use of hand lift trucks and skids in railroad stores departments has continued to expand during the year, and practically all the larger railroad systems have made initial installations or are increasing present equipment, with gratifying economies. Several intercoastal steamship lines are actively soliciting skidded freight, because economies in handling make this an attractive cargo.

Hand lift trucks of standard and special design are being widely used for handling desks, refrigerators, stoves, safes, machine tools,

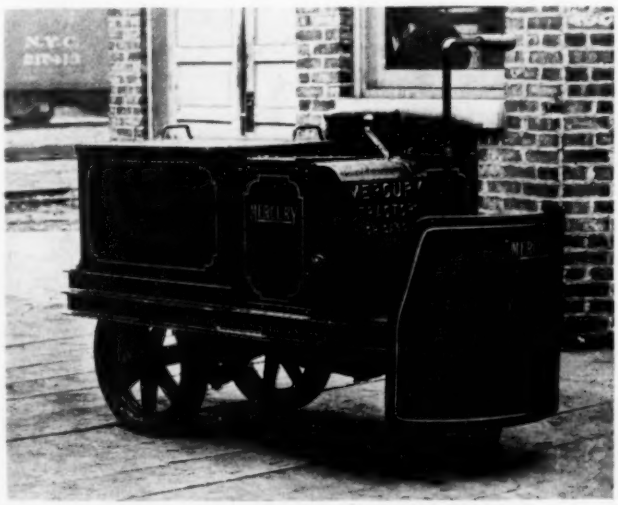


Fig. 40 Truck Tractor But 32 In. in Overall Width and With Correspondingly Short Turning Radius

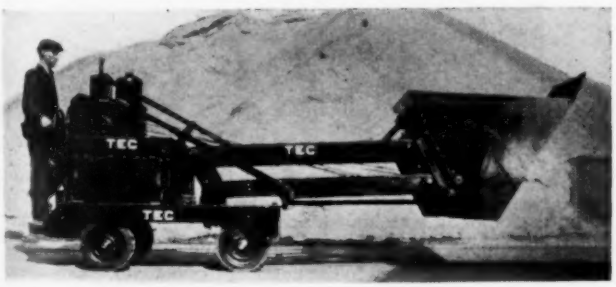


Fig. 41 Shovel Truck for Tunnel Work With 6 Ft. 6 In. Overhead Clearance

and other commodities that are provided with legs allowing entrance of the truck. Open-end lift trucks are also found satisfactory for handling reels of wire, crates, boxes, tote pans, etc., where platforms are not required. Many manufacturers are giving thought to redesigning their product to allow of handling in this way, as a marked saving is gained both throughout production processes and in warehousing.

A new type of truck, known as the Footlift, has been brought out by the Lewis-Shepard Company during the past year, which, as its name implies, is elevated by the operator's foot, using the leg and thigh muscles instead of the arm and back muscles to elevate the load. Among other advantages claimed for it are ease, quickness of operation, safety, and less strain on the operator. All manufacturers of hand lift trucks are giving more attention to refinement in design and safety features. It is now generally recognized that the jack-type, or multi-stroke, trucks should be used for handling heavier loads, and all hand-truck manufacturers are prepared to furnish this type of equipment.

Portable Elevators. In the portable-elevator field a large increase is noticeable in the number of special machines designed to perform operations that previously had been considered as requiring brute strength. Among the many types which are coming more into general use may be mentioned: warp-beam

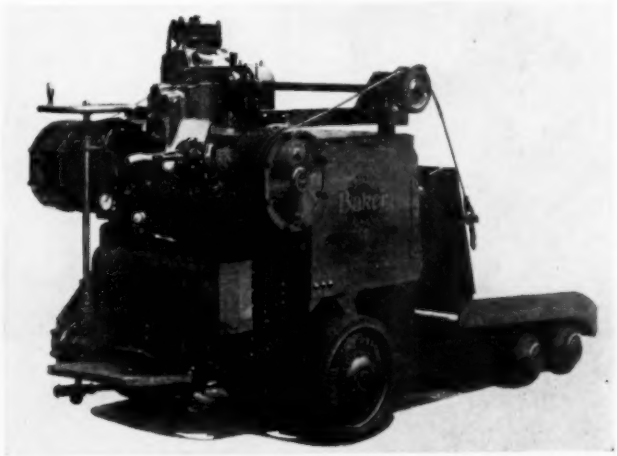


Fig. 42 Truck for Handling Cable Reels

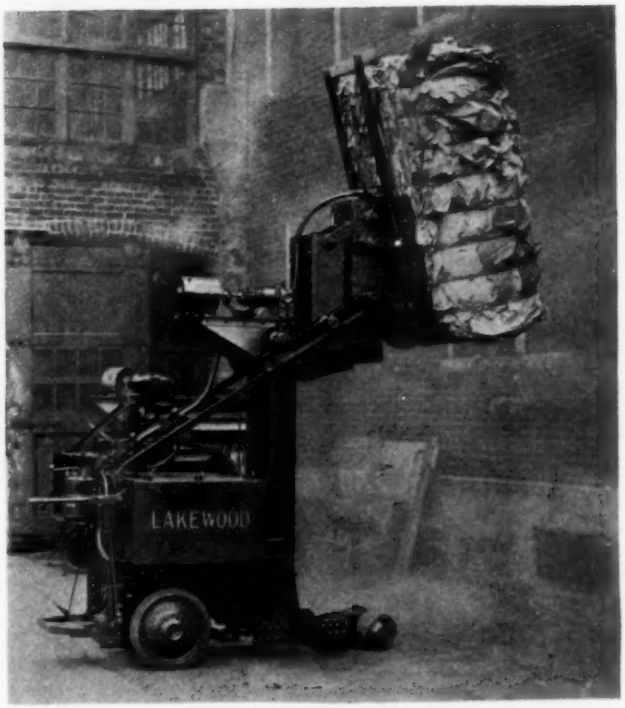


Fig. 43 Tilting Fork-Type Machine for Handling Bales of Waste Paper

carriers, barrel- and drum-lifting machines, oil-switch-tank lifters, refrigerator cranes, die lifters, and special-purpose machines for handling large work in process.

Many machines are handling loads not exceeding 200 to 300 lb., and, to make the machine as light and portable as possible, special section members are often used, constructed by arc welding.

Many installations of portable elevators are used, in connection with skid and lift-truck systems, to pile or rack the loaded skids. It is not uncommon to pile 2000-lb. loaded skids six high, with a corresponding saving in floor space. Machines of heavier capacity have been placed on the market during the past year, and installations of 6000 to 10,000 lb. capacity are not unusual.

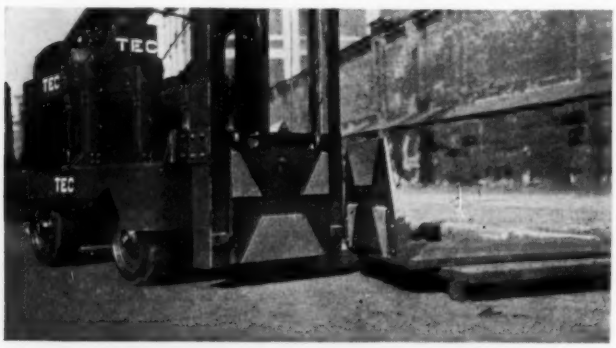


FIG. 44 DOVETAIL ARRANGEMENT FOR RAPIDLY ATTACHING A SHOVEL, PLATFORM, AND FORK OR RAM TO A TRUCK BODY

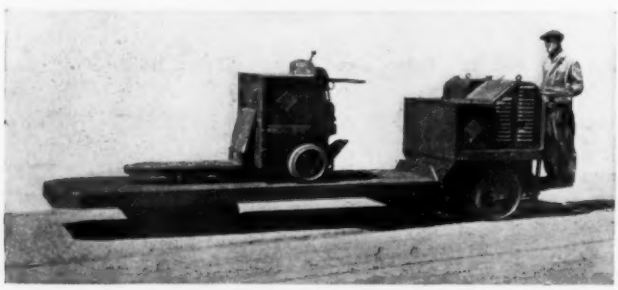


FIG. 45 ELECTRIC TRUCK WITH 12-FT. PLATFORM FOR HANDLING L. C. L. CONTAINERS
(1¾-ton truck resting on platform.)

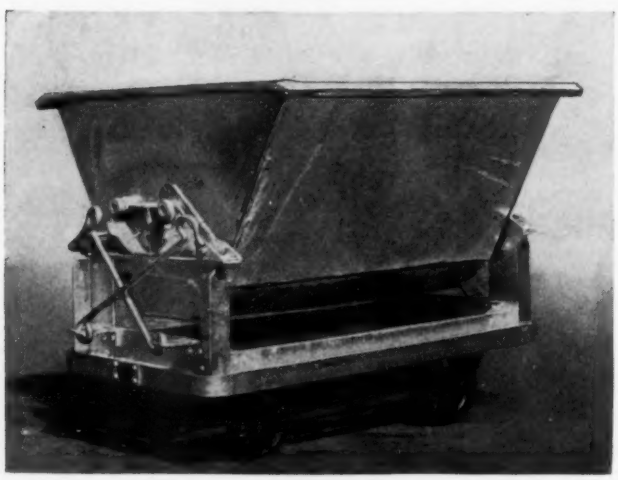


FIG. 46 GALVANIZED TRAILER FOR USE IN PACKING PLANT

Four-post or double-end machines, supporting the carrying platform at two sides, are being used in increasing numbers for adjustable loading docks. Capacities of from 4000 to 8000 lb. are not uncommon in this type of equipment.

A sharp increase in the percentage of power-operated machines has been noticeable, because it is far more economical to elevate commodities by electricity than by hand-power. Hand-power machines are now rarely specified for other than extremely intermittent use.

A gasoline-operated power machine has been developed for

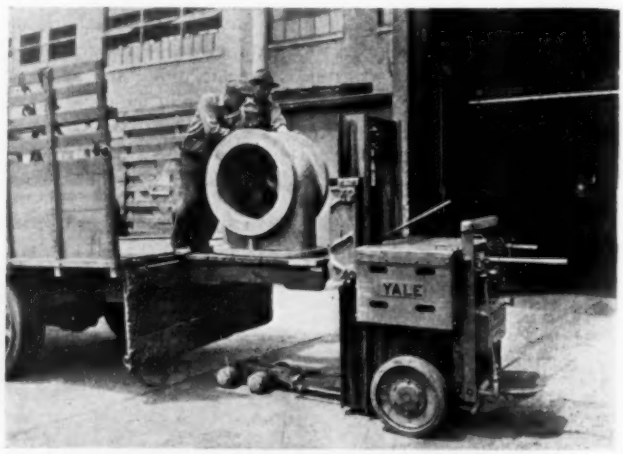


FIG. 47 400-LB. HIGH-LIFT TRUCK

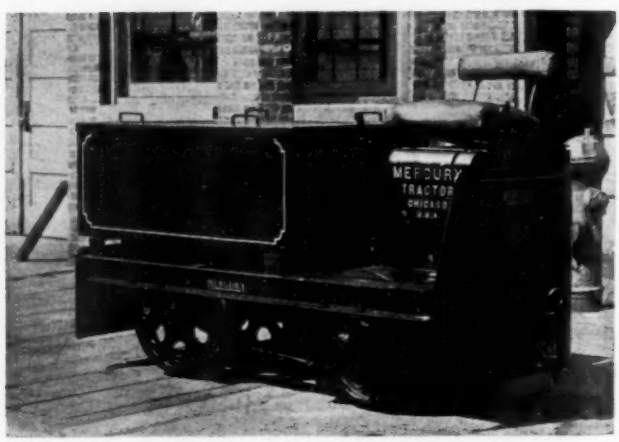


FIG. 48 NEW TYPE OF NARROW-GAGE FLANGE-WHEEL TRACTOR

use on docks, or in warehouses not wired for power. This machine is equipped with a small self-contained air-cooled motor operating through a clutch to the hoisting mechanism. Lowering is controlled by a centrifugal-type governor acting through a band brake. This type of machine is especially useful in loading automobile trucks and handling material on construction jobs and other out-of-the-way places.

ELECTRIC INDUSTRIAL TRUCKS[10]

Progress in the design and use of electric industrial trucks during 1930 has followed the general course witnessed during the past few years. The new developments have been, in general, in the shape of refinements, although one or two striking improvements have been seen.

Increased Capacities. During the past year, electric crane trucks have been developed with a rated capacity of 42,000 ft-lb. overturning moment. Another machine has been placed in service at the Jones & Laughlin plant which has a bifurcated platform and handles 20,000-lb. bosh tanks in the tinplate mill. Ram trucks, in general, have been limited in capacity to 3000

[10] Section prepared by C. B. Crockett, Secretary, Industrial Truck Association, New York, N. Y. Assoc-Mem. A.S.M.E.

and 4000 lb. By using an articulated front axle, the truck shown in Fig. 38 now handles 8000 lb. These machines have also been developed with a twin ram instead of the single one as shown. By the use of counterweights the capacity of the fork-type truck has been increased to over 10,000 lb. This year has also witnessed the increase of drawbar pull in tractors, the maximum available now being 7500 lb.



FIG. 49 INDUSTRIAL TRACTOR DEVELOPING DRAWBAR PULL OF 26.64 HP.

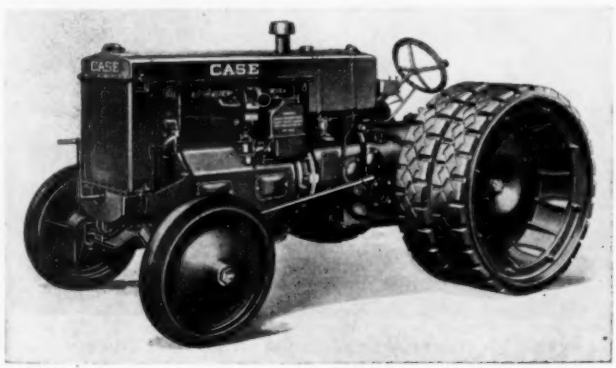


FIG. 50 INDUSTRIAL TRACTOR WITH TURNING RADIUS OF 156 IN. DEVELOPING DRAWBAR PULL OF 26 HP.

Not only has the weight capacity of these machines been made greater, but the hoisting speed of cranes and the lifting heights have been increased as evidenced by the high-lift truck of 6000 lb. capacity and 160 in. overall height, shown in Fig. 39.

Increased Maneuverability. Along with the introduction of newer and larger machines have come many developments to increase the maneuverability of the equipment. The leading manufacturers of storage batteries have developed a new type of cell providing a larger amount of power per square inch of area occupied by the cell. This development has resulted in the introduction of a new heavy-duty-type tractor which has the same clearance and turning radius as the standard machine, but with approximately 30 to 40 per cent more power. There has also been developed a tractor with the standard drawbar pull but only 32 in. in overall width, and with a correspondingly short turning radius. (See Fig. 40.)

Fig. 41 shows a shovel truck used to carry coal through a tunnel with 6 ft. 6 in. overhead clearance, and at the same time deposit its load in hoppers 8 ft. above the ground. This same principle of handling at an elevation, and yet maintaining low headroom, has been developed in the telescoping type of high-lift truck. One machine of this type can pass through the ordinary factory door, but can raise its load to a height of 10 or 12 ft. if necessary.

Safety features have been improved by means of switch interlocks, and trucks are now in use where motors, controllers, and limit switches are equipped with air-tight, gasketed covers so that the machine may safely be operated in a highly inflammatory atmosphere.

New Attachments and Machines. New and novel atachments and machines include the following:

1 A self-operated clamp for handling tinplate.

2 A fork-type truck in which the tilting mechanism operates automatically from the lift.



FIG. 51 THREE-WHEEL TRACTOR OF SHORT-TURNING RADIUS



FIG. 52 6000-LB. LIFT TRUCK

3 A truck for handling cable reels (see Fig. 42) with a limit switch on the winch, operated by the reel on the platform of the truck.

4 A mechanism for reeling in and out a predetermined distance.

5 A special tilting fork-type machine for handling bales of waste paper (see Fig. 43).

A new 6000-lb. high-lift truck has appeared during the year with the following features: hydraulic lift, flat springs, under-

driving wheels, internal brakes, and ball bearings in steering knuckles. A new method has been evolved for attaching tires to trailer wheels by which the wheels are sent to the tire manufacturers and the rubber tires are molded directly on the wheels.

There has been a marked development of attachments that may be used with standard equipment. Fig. 44 shows a dovetail arrangement by which a shovel, platform, and fork or ram may be attached to a truck body in less than a minute. A truck with beveled forks is now being used for handling paper rolls from the floors of freight cars.

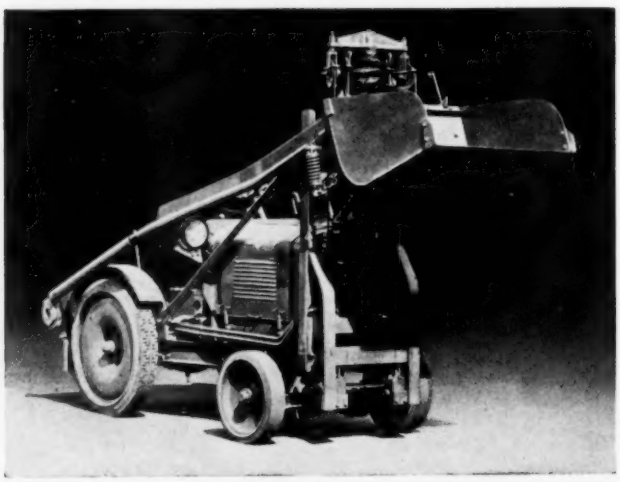


FIG. 53 POWER SHOVEL BUILT ON CHASSIS OF 40-HP. TRACTOR

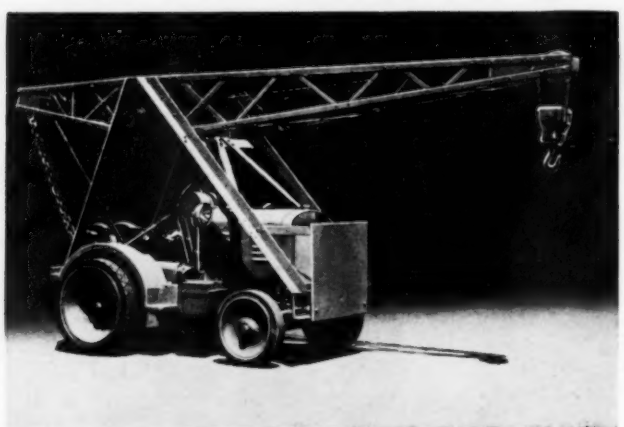


FIG. 54 ADJUSTABLE-HEIGHT BOOM CRANE BUILT ON CHASSIS OF 40-HP. TRACTOR

New Uses for Electric Trucks. New uses for electric industrial trucks include: the use of a tractor for regular car spotting at the repair shops of the Chicago Rapid Transit Company: the delivery of oil and grease by the means of a pump and tank mounted on a skid platform; the handling of concrete in plant construction; the use of equipment for plowing off snow; and, most important, the extensive handling of l.c.l. containers. Machines for handling these containers have platforms 12 ft. in length by 40 in. wide. Fig. 45 shows one of these machines with a $1^{1}/_{4}$-ton machine resting on the platform.

Fig. 46 shows an interesting combination of materials-handling and production equipment. These galvanized trailers are used in a packing plant to receive beef bones. When filled, the trailers are moved by tractor to another location where they are filled with hot water, and a steam line is connected to a manifold which runs through the water at the bottom of the trailer. A cooking process is thus carried on and the meat juices extracted. The fluid is then drained off and the bones are transferred in the same container for utilization in other departments.

During the past year railroads and large industrial plants have made considerable progress in eliminating the rehandling of waste products. At several locations, installations have been made in which unit containers, either mounted on skids or trailers, have been spotted at various points throughout the plant. These containers, when filled, are moved by lift truck or tractor and dumped in gondola cars or a pit. In addition to the saving of labor, the rehandling, or sorting, of scrap or waste material is thus eliminated by providing a sufficient number of containers.

Coordination of Handling Equipment With Other Types of Equipment. The Great Lakes Transit Company has coordinated the use of tractors and trailers with conveyors for unloading, sorting, and transporting flour from ships to warehouse and freight car. The conveyor is used to bring the bags from the hold, and becomes an automatic sorting table, the bags being lifted off the conveyor and placed upon the proper



FIG. 55 THREE-WHEEL TRUCK OF 500 LB. CAPACITY

trailer according to the various brands and consignees. In Chicago, l.c.l. freight is loaded directly from the tailboard of the truck to trailers at various freight depots throughout the city. These trailers are run directly into freight cars and moved without rehandling to the transfer house 16 miles outside the city. Because material for only one destination is placed on each trailer, these trailers are moved at the transfer house directly to the proper car, so that only two handlings have to be made, one at the unloading of the truck, and the other at the loading of the freight cars that carry the material to its final destination.

In coordinating the use of power lift trucks and hand lift trucks, the machine shown in Fig. 47 has been developed during the past year. This is a 400-lb. high-lift truck with a platform $6^{1}/_{2}$ in. high, in lowered position, for use with hand-lift-truck skids, but with a four-wheel compensating front axle for two-ton loads.

The stores department of the Chicago, Milwaukee, St. Paul and Pacific Railroad has successfully coordinated the tractor-trailer and skid-lift-truck systems. The Milwaukee shops, which cover some three square miles, are served by 11 receiving stations. Each of these stations is served by two trailer trains, each train starting at Station No. 1 and making the complete circle, traveling in a direction opposite to the other. The trains operate on regular schedules and provide pick-up and delivery service for each station at approximately 45-min. intervals. In this

way the long-haul transportation is accomplished by means of the tractor-trailer system, while the goods are handled by lift truck in and about the shops and storerooms. The trailers are designed for two 6000-lb. skids, although average loads are not so great as this amount. The average train consists of approximately eight loaded trailers and four empties for picking up loaded skids at intermediate points along the route.

A new type of narrow-gage flange-wheel tractor, or electric "locomotive," has wheels and axles so made that the gage can be adjusted within the limits of 18 in. minimum and 30 in. maximum. This truck is shown in Fig. 48. Another new four-wheel tractor is equipped with an extra large battery consisting of 24 cells of the 23-plate type.

Gasoline Trucks and Tractors[11]

Considerable activity has been manifest among the manufacturers of gas-powered industrial haulage vehicles during the year 1930. Twelve new gas-powered tractors and lift trucks have been introduced by seven manufacturers. Three large

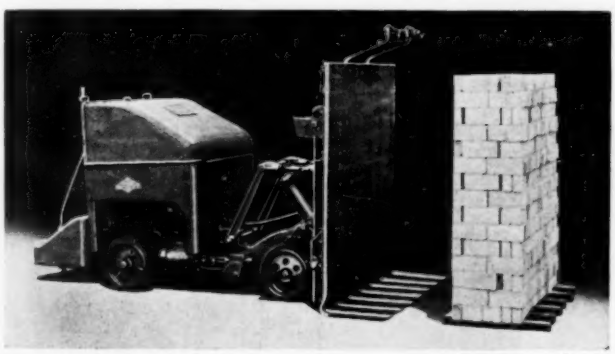


Fig. 56 Gas-Powered "Brick Hacker" for Handling Green and Pressed Brick

manufacturers of agricultural tractors have entered the industrial field in a substantial way, and the introduction of a new light passenger car by one of the large automobile manufacturers has led to the introduction of several light gas tractors powered by this automobile engine.

The Allis-Chalmers Manufacturing Company introduced this year its model "U" industrial tractor, Fig. 49, developing a maximum drawbar pull of 24.64 hp. The weight of the standard tractor is 4125 lb., and it comes equipped with a power take-off pulley and a number of special attachments, such as crawler treads, sweepers, side fillers, scrapers, air compressors, and other useful industrial equipment.

A new industrial tractor was placed on the marked this year by the J. I. Case Company. It is especially designed for use with industrial attachments, such as street sweepers and snow brushes, bucket loaders, cranes, and lift platforms. Model "C-I" has a drawbar pull of 17 hp. and a turning radius of 120 in. The larger model "L-I," Fig. 50, has a drawbar pull of 26 hp. and a turning radius of 156 in. This company also announces a special narrow-tread model of its "C-I" tractor, with a turning radius of 109 in., overall width of 48 in., as compared to the standard model having an overall width of 61 in. All models may be equipped with steel wheels for industrial use, and also with winter cabs and electric lights, when required.

The Charlier Manufacturing Company has developed three types of gasoline-motor-powered vehicles. A new "Brownie" three-wheel tractor, Fig. 51, has an exceptionally short turning radius of 72 in., a drawbar pull of 1200 to 2500 lb., weighs 1950 lb. and can be ballasted to suit the job. The spring-suspended third wheel allows the tractor to travel over uneven and rough roadways with the ease of the four-wheel type. It is powered by a Ford Model AA engine and has speeds of from 1 to 30 miles per hour (four forward speeds and one reverse). A heavy $^3/_8$-in. steel bumper plate is attached to the front.

A new four-wheel tractor made by the same company is similar to the three-wheel type, except that it steers on two wheels instead of one.

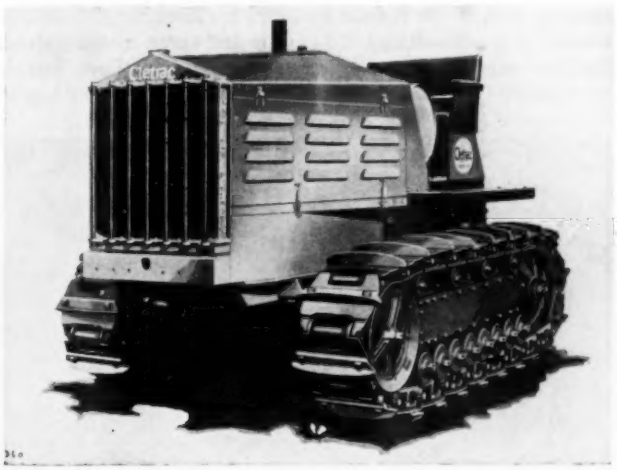


Fig. 57 60-Hp. Crawler-Type Tractor Equipped With 6-Cylinder Engine

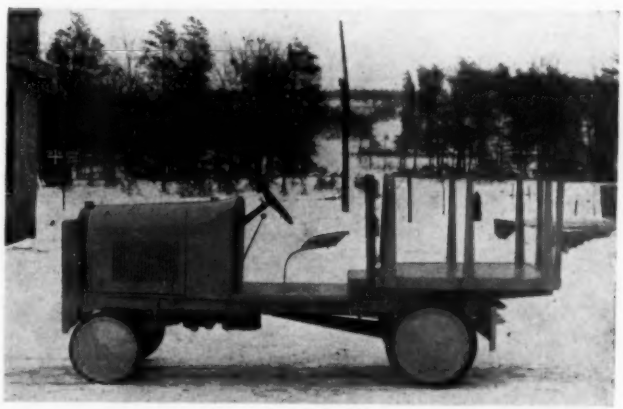


Fig. 58 "Shop Mule" Powered With Ford Model "T" Engine

A new "Brownie" lift truck, Model LT-3, made by the Charlier Manufacturing Company, and illustrated in Fig. 52, has a capacity of 6000 lb., a platform height of 11 in. or 17 in., a safety tread platform 56 × 24 in., a lifting speed of 4 seconds up and 3 seconds down, a turning radius of 120 in., is powered by a Ford Model AA engine, travels two to eight miles per hour, and weighs 2575 lb.

The Clark Tructractor Company, in May, 1930, introduced a new 40-hp. industrial tractor known as the "Clarktor." It is equipped with either solid or pneumatic tires and carries generator, battery, self-starter, and lights. The turning radius is 108 in. With this tractor as a chassis, the same company is putting out a high and low power shovel, Fig. 53, known as the "Clarktor," diggers, and also an adjustable boom crane, Fig. 54, with capacity from 1600 to 4000 lb. and with a maximum hook

[11] Section prepared by Ezra W. Clark, Vice-President, Clark Tructractor Co., Battle Creek, Mich.

height of 13 ft., for use on docks, piers, and railroad yards.

What is said to be one of the smallest straight industrial tractors has also been put out by the same company. It is known as the 4-wheel "Clarkat," has a turning radius of 92 in., a drawbar pull of 2000 lb., and a towing capacity, on the level, of 25 tons on trailers. It is designed primarily for factory, mill, and railroad work.

An improved model of the same company's 3-wheel "Clarkat" has been introduced. It has a turning radius of 51 in., and is equipped with a steel box platform 33½ in. wide by 29½ in. long, of 500 lb. load capacity, at the rear (see Fig. 55). This platform adapts the tractor as a stock chaser for the prompt delivery of heavy castings and tools in and about machine shops, foundries, and manufacturing plants. The platform can be removed in two minutes, making the standard trailer coupler

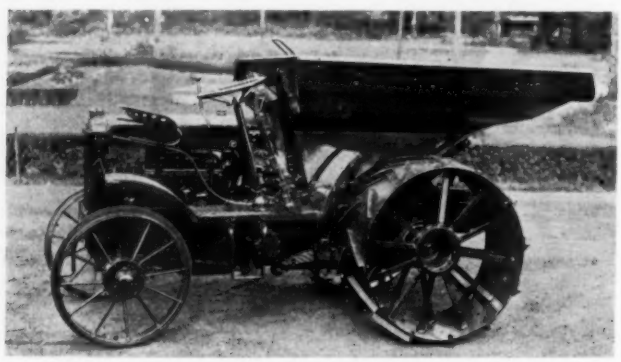


FIG. 59 THE "IRON MULE," A 2-TON DUMP TRUCK BUILT ON THE CHASSIS OF AN INDUSTRIAL TRACTOR

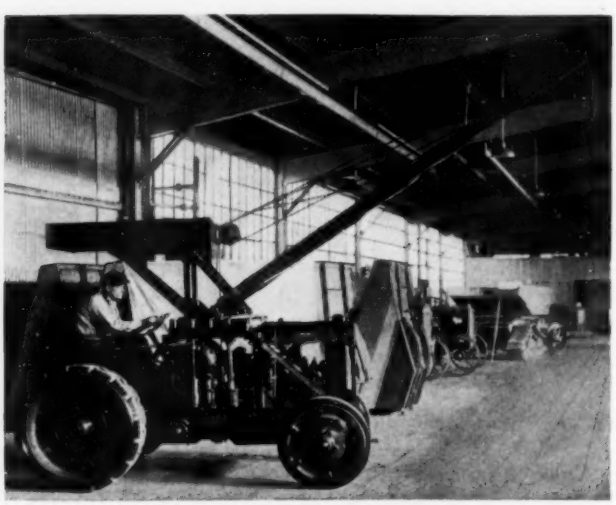


FIG. 60 SWINGING-BOOM CRANE ("ROUSTABOUT")

available so that the machine can be used as a regular factory tractor. The same company has also introduced to the brick and clay block plants a gas-powered "Brick Hacker," Fig. 56, with a capacity for handling a pile of 500 bricks. This machine is designed for the handling of bricks from kiln to storage, storage to shipping platform, to car, and also for barge movement.

The Cleveland Tractor Company, manufacturer of crawler tractors, introduced this year one new model, known as the 80-60 "Cletrac," shown in Fig. 57. It is rated at 60 hp., but is said to be capable of developing 80 hp. It is powered with a six-cylinder four-cycle engine governed to 1050 r.p.m., is equipped with Delco-Remy starting, lighting, and ignition, and, when specified, is furnished with power pulley and take-off.

The "Shop Mule," which for years was powered with the Model T Ford engine and parts, has been put out this year by W. F. Hebard & Company in new form, and is now equipped with the Model A Ford engine and necessary parts. It is four-wheeled, is built with a spring seat for service as a tractor, and has

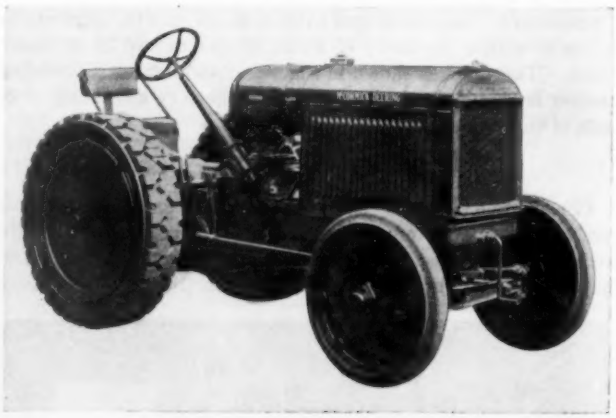


FIG. 61 36-HP. INDUSTRIAL TRACTOR

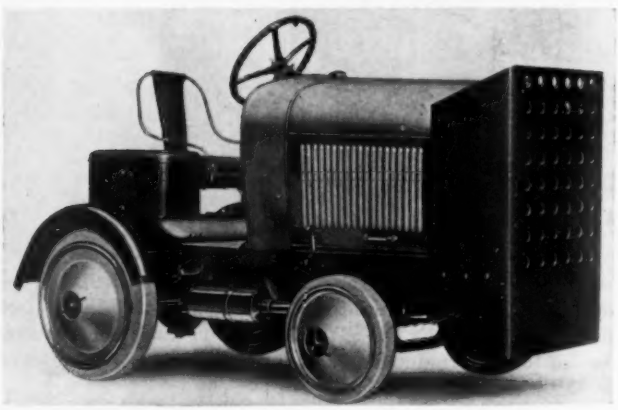


FIG. 62 SMALL 4-WHEEL GAS-POWERED INDUSTRIAL TRACTOR—THE "INDUSTRIAL TUG"

a capacity of 15 to 20 tons on trailers. With a longer platform, the model K will carry a load and also two trailers, and as model L shown in Fig. 58, it is a load-carrying tractor-type "Shop Mule." The Hebard machine is the oldest of the Ford-powered industrial tractors, and the three new models put out this year embody all of the former features, with certain refinements and improvements that make this an economical small tractor for general use.

Using the Allis-Chalmers industrial "U" tractor as a base, the Hughes Keenan Company builds what is known as the "Iron Mule." Equipped with a 2-yd. body, as in Fig. 59, it finds a wide usefulness in the road-building field.

A swinging-type, general-utility crane known as the "Roustabout Crane" is also a new product this year of this company, being built both to the Allis-Chalmers model "U" and to the McCormick-Deering industrial tractor. It is a swinging-boom crane, Fig. 60, with overhead counter-balancing weight, with a rated capacity of 2000 lb. It will pass through a door 5 ft. 6 in. wide by 7 ft. 6 in. high with ample clearance. Control levers are within easy reach of the operator. It is believed that it will have a large field of usefulness among the railroads.

The International Harvester Company has placed on the market a new model, No. 30, of its McCormick-Deering industrial tractor, which provides a flexible and conveniently controlled tractor for heavy industrial jobs (see Fig. 61). It is built to the general design and construction of the previous model, No. 20, but is of larger capacity, and therefore available for heavier duty. Among its important improvements are heavy-duty, internal, self-energizing brakes. The driver rides at the rear of the machine and is always in control of the tractor. Rear rims and tires are quickly removable. The four-cylinder, heavy-duty tractor engine develops 36 hp. on the level.

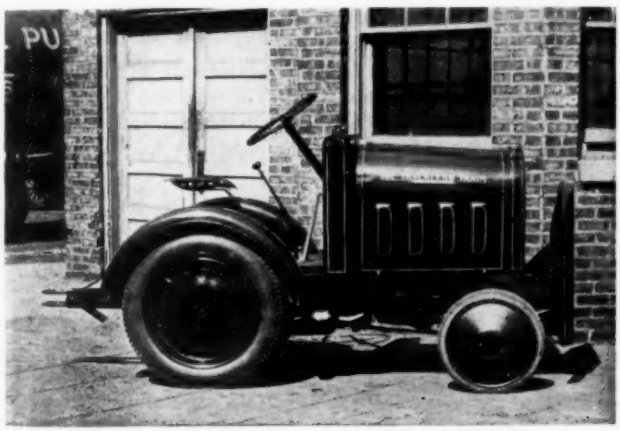


FIG. 63 NEW INDUSTRIAL-TYPE GAS TRACTOR

Removable cylinders permit of convenient, quick servicing. Special equipment provides for lighting, power take-off, and either solid or dual pneumatic tires. The turning radius is 198 in.

The Whitehead & Kales Company is a new entrant into the field of gas-powered industrial tractors, and is putting out a small four-wheel tractor, Fig. 62, known as the "Industrial 'Tug.'" It has a drawbar pull of approximately 1500 lb. and is powered with a Ford engine and parts. It is equipped with a tool box beside the driver's seat, and a heavy front bumper.

Using this tractor as a chassis, the same company has introduced the Whitehead & Kales golf-course tractor, equipped with steel wheels fitted with sod-puncture pins. The wheels are drilled to take extension rims. With a maximum of 1500 lb. on the rear wheels, this golf tractor has ample traction without excessive weight to pack the turf.

A new industrial-type gas tractor, Fig. 63, has been developed by the Mercury Manufacturing Company. This new tractor can be furnished equipped with dual solid tires on the rear wheels, and extra wide tires on the front wheels, with single solid tires, or with pneumatic tires on the rear wheels. It has a standard Model A Ford motor with electric starter, including an 80-ampere-hour battery, Ford radiator, and 6-gal. gravity-feed gasoline tank mounted on the dash. The normal drawbar pull is 1000 lb., with a maximum of 3000 lb. The machine has a turning radius of 104 in. and weighs 3600 lb.

The tractor was developed for outside work, mainly in yards and places where large wheels are desirable. It can be equipped with a power take-off to operate various attachments, such as brooms for sweeping snow, and other devices commonly used in handling materials in industrial plants.

INDUSTRIAL ELEVATORS[12]

Much publicity has lately been given to the really marvelous developments in automatic passenger-elevator equipment, but very little has been said about the increasing efficiency of freight elevators and the rapidly increasing standards of performance now demanded for this type of elevator in all classes of buildings. Even up to a few years ago, office buildings generally attempted to move freight, such as furniture, stationery, etc., in one of the regular passenger-elevator cars which was temporarily assigned for certain periods in the day for freight purposes only.

The tendency now, however, in these larger and monumental office buildings which are going up all over the country, is to install regular freight elevators for that purpose only. These elevators in the higher buildings are of the gearless-traction type, with car-switch control, running at speeds up to 600 and 700 ft. per min. Thus they are practically the same as the passenger elevators in the older office buildings. For instance, the Empire State Building, in New York City, now under construction, will have five distinct freight elevators.

The progress in freight-elevator equipment during the past

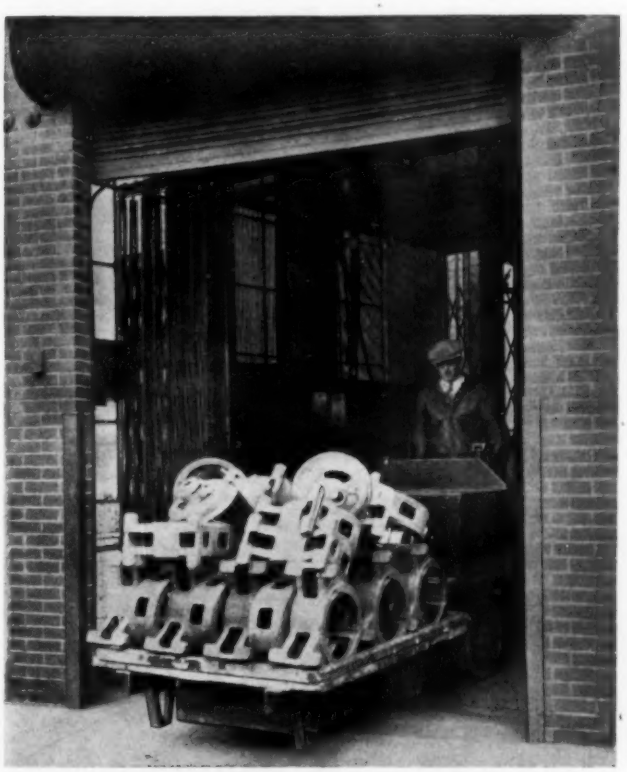


FIG. 64 USE OF LIFT TRUCK ON ELEVATOR

year has been more one of increasing standards of performance rather than the development of any particular new type of apparatus. The micro-leveling feature on freight elevators is rapidly coming into general use, owing to the increased efficiency of such elevators. Micro-leveling eliminates "inching" at the landings, thus obviating the possibility of damage to trucks or merchandise when the trucks are run on or off freight elevators (see Fig. 64). This operation also, by eliminating the false stops, reduces current consumption, cuts down wear and tear on the apparatus, and prolongs its life.

Another development of the smaller freight elevator, in locations where the service does not require an operator constantly on duty, is the use of the double-button type, which is rapidly superseding the old-style hand-rope control. The hand-rope elevator had its use where it was desired to have the elevator operated by those who were using it, rather than by an operator on

[12] Section prepared by E. A. Fitch, Vice-President, Otis Elevator Company, New York, N. Y. Assoc. A.S.M.E.

the car. The hand rope could be pulled from any floor, bringing the car to that point. This necessitated safety devices, such as a rope lock, to prevent the car being taken away from any one who was using it. It was also necessary, of course, to have self-closing gates or doors, which would be held open while the car was standing at the floor, and would close automatically when the car left, so as to prevent the possibility of any one falling down the hatchway under the impression that the car was at the landing.

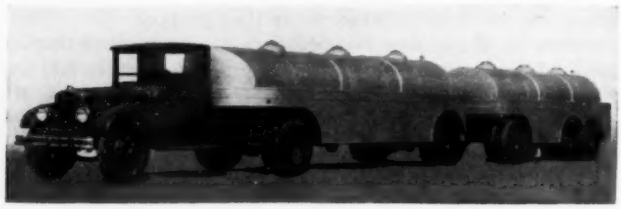


FIG. 65 MOTOR TRUCK WITH OIL-TANK TRAILERS



FIG. 66 MOTOR TRUCK WITH 8-WHEEL DRIVE



FIG. 67 BUNK BUILT BACK OF DRIVER'S SEAT IN MOTOR-TRUCK CAB

This new double-button type is operated from each floor by the pressure of the button, which calls the elevator to that floor if it is not already in use. "Up" and "down" buttons in the car then operate the elevator in the desired direction, constant pressure being exerted on the button employed until the destination is reached. These elevators are also equipped with gates or doors at each floor for protection.

Not so many years ago the freight elevator was not considered of very great importance in manufacturing plants warehouses, or, in fact, in any type of building. Many owners felt that any elevator which would get the freight up and down was good enough. This feeling has been gradually changing, however, and the last year has seen a material alteration of viewpoint, looking toward the installation of more efficient and time-saving apparatus.

Another factor of prime importance is the capacity and size of elevator as compared to the electric and hand lift trucks used in industrial service. Many a plant could save considerable money by installing freight elevators of sufficient size and lifting power to carry the largest electric trucks, with maximum load, that the plant could effectively operate.

It is now realized that freight elevators are as important in their place as passenger elevators. The machines manufactured by leading concerns in the elevator industry are now of the same high class in every respect as the passenger machines. They

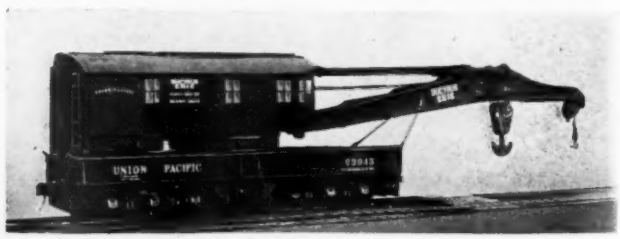


FIG. 68 NEW 200-TON WRECKING CRANE



FIG. 69 NEW TYPE OF HEAVY LOCOMOTIVE CRANE

are equipped with practically all of the same types of safety devices—so as to prevent accident either to the materials carried, or to the elevator apparatus itself—and also with devices to prevent interruption of service. This development is resulting in much more efficient operation, reduced operating expense, and a saving of considerable time in the transfer of freight, with consequent speeding up of production in the factories.

MOTOR TRUCKS AND TRAILERS[13]

So numerous and so complex have been the year's developments in the motor-truck industry, that any attempted report must be confined to a few generalities in order to conform to prescribed space limitations.

Most significant is the increasing tendency of industry to appreciate the flexibility with which trucks can be adjusted to the individual traffic and materials-handling problems of manufacturers. As an outgrowth of this realization, motor trucks

[13] Section prepared by Edward F. Loomis, Secretary, Motor Truck Committee, National Automobile Chamber of Commerce, New York, N. Y.

are coming more and more to be regarded as highly specialized units of machinery, each designed to perform a specific type of operation with the utmost efficiency.

Developments Resulting From Research. Research undertaken cooperatively by truck, chassis, equipment, and parts manufacturers has been productive of uncounted contributions to the more efficient and economical use of motor trucks in materials handling. To cite but a few instances:

1 The perfection of lighter and stronger metals in body construction, permitting a substantial increase in the truck's pay-load capacity. Aluminum and alloy metals in some cases have subtracted from 1500 to 2000 lb. from the truck's dead weight (see Fig. 65).

2 Devices for the chemical or mechanical heating and refrigerating of trucks, permitting the transportation of materials and produce under regulated temperature conditions.

3 Bodies and appliances for the safer movement of combustibles and explosives.



FIG. 70 RAILWAY MAINTENANCE CRANE

4 Improvements in appliances for the preparation and mixing of concrete and other plastic building materials in transit.

5 Bodies and equipment to expedite and to simplify loading and unloading of vehicles. In this division should be included auxiliary appliances, conveyors, improved side- and rear-dump-truck hoists, cranes for handling container-car units—the list is almost endless.

6 Mechanical improvements (see Fig. 66) in truck, tractor, trailer, and dolly equipment which have contributed materially to the economy of truck operation, or have made the truck capable of transporting bulky or unwieldy machinery and materials. One instance is the development of the double-deck trailer for transporting automobiles.

An index of the trend toward the use of balloon and pneumatic tires is provided by records showing that solid tires are supplied as standard equipment on less than twenty per cent of the models, and less than four per cent of the trucks, produced during the year.

Widening Field of Use. More than passing emphasis is merited by the increase in the use of truck-and-trailer, and tractor-semi-trailer-and-trailer articulated units, due to improvements in coupling and braking devices, and a better general understanding of the economic advantages of pneumatic- and balloon-tire equipment. This single development contributes to the greater efficiency of the power unit by permitting its operation while extra trailers are being loaded and unloaded, to say nothing of the increased pay load that can be transported at one time with such equipment.



FIG. 71 HEAVY-DUTY PILE DRIVER

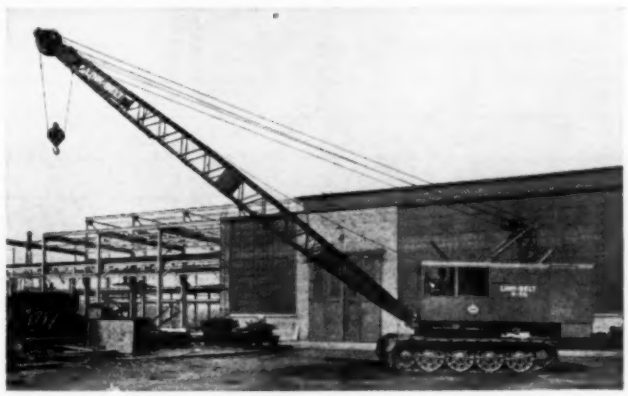


FIG. 72 CRAWLER-MOUNTED SHOVEL CRANE

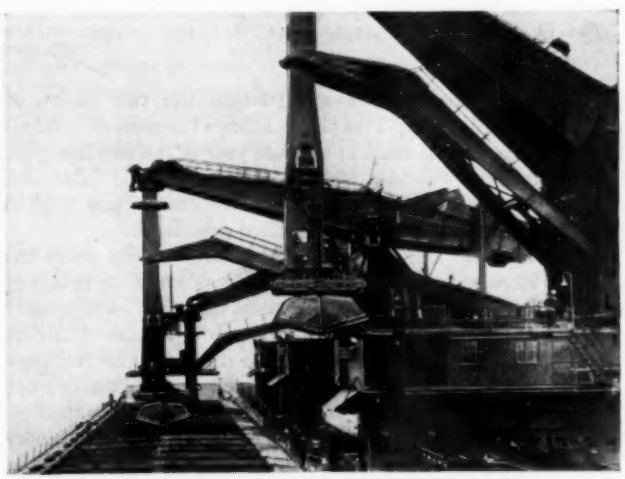


FIG. 73 UNLOADERS AT PRESQUE ISLE DOCK, TOLEDO, OHIO

Use of trucks by railroads in developing their shipper-to-consignee service has made spectacular advances, the number of

trucks operated by railroads exceeding 6000 vehicles, exclusive of the Railway Express Agency's 10,000-truck fleet.

Extension of operating radii is predicted by two additional developments—sleeping compartments in truck cabs for relief drivers, and radio receiving sets for trucks. While the last-named development has not yet been so used, traffic experts are seriously considering the possibilities of short-wave radio equipment in the dispatching and routing of trucks. This device has already been proved satisfactory by the police departments of several large cities. The relief driver's sleeping compartment is now manufactured by several companies. In most cases it consists of a bunk built into the cab, as shown in Fig. 67, directly over or back of the driver's seat.

Both of these developments, it is believed, will contribute particularly to the expansion of common-carrier, inter-city truck operation.

Locomotive Cranes[14]

For the purpose of this Report, "locomotive cranes" will include railroad wrecking cranes, although the two classes of equipment are distinct. This plan is followed because of developments that have been made in the past year in the wrecker type. The designation will also include crawler-mounted machines, although originally the term "locomotive crane" was applied only to 4- or 8-wheel cranes for standard-gage track.

Recent Trends. One of the chief trends of recent years has been the increased use of crawler mountings, enabling cranes of small capacity to work close to their loads, instead of using heavier cranes, with long booms, operating on remote tracks. Crawlers have also given mobile crane service to users with no trackage. On the other hand, where tracks are already laid, the railroad type has many advantages: faster operation, easier propelling, ability to switch cars, and less wear and tear on machinery from traveling on uneven ground. Each type will continue in demand, meeting efficiently particular combinations of needs.

Another trend is toward internal-combustion drives, due to their greater economy for intermittent work, and freedom from the limitation of small boiler capacity. Diesel drives are growing in popularity, and their high first cost is justified by operating economies where a high load factor can be maintained. Internal-combustion drive, however, imposes an entirely different demand on clutches and other elements than does steam, and therefore the best gasoline or Diesel machines are those specially designed for these prime movers. Steam continues in demand with many users and under many conditions. Where other conditions warrant it, a steam-driven crane will probably outlive a gasoline or Diesel machine.

There is a noticeable demand for increase in speed, all-steel construction, alloy-steel parts, frictionless bearings, splined shafts, interchangeability of parts, and, to some extent—with gasoline and Diesel drives—variable-speed transmission.

Wrecking Crane. A new 200-ton capacity Bucyrus Erie wrecking crane, Fig. 68, has been introduced, having holding-down rear rollers and equalized twin main-hoist lines. Its operating weight is approximately 370,000 lb., with low center of gravity for high speed in transit. Boom hoist and slewing are operated by spur gears, the former having an automatic self-locking pawl.

Steel Erecting Crane. In conjunction with one of the large steel-erecting companies, two sizes of a new type of heavy locomotive crane have been built by the Industrial Brownhoist Corporation, suitable for use with extraordinarily long booms up to 200 ft. in length, and designed to handle very heavy loads with shorter booms. The smaller crane, shown in Fig. 69, handles 75 tons at 13 ft. 6 in. radius, and the larger crane handles 150 tons at 17 ft. radius. Both are adapted to form the power plant for heavy bridge-builders' derrick cars of larger capacity than anything heretofore used as regular equipment. Each crane includes a separately driven high-speed, low-power hoisting drum which is a great steam saver, speeds work, and saves the main machinery.

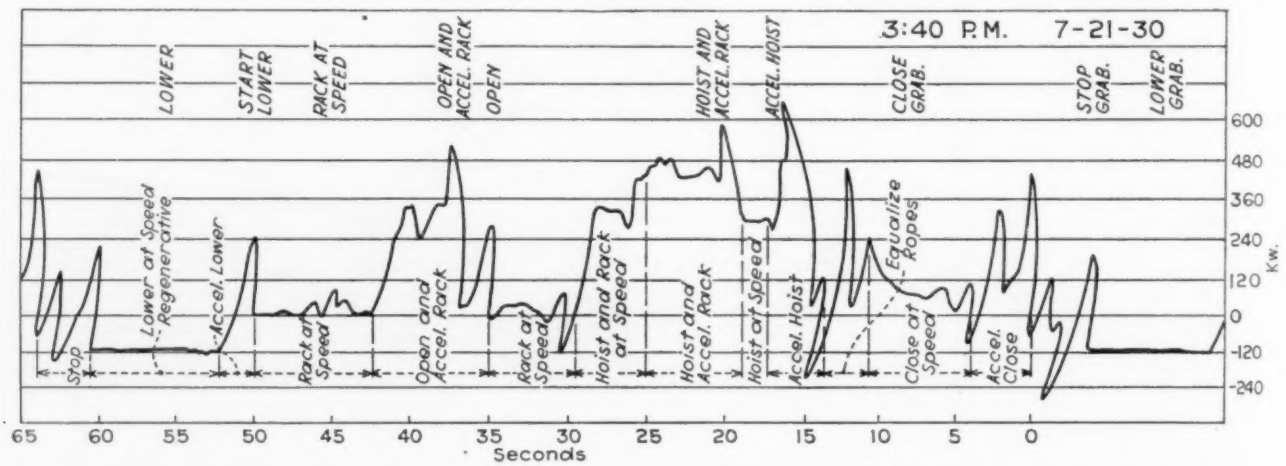


Fig. 74 Recording Graphic Chart Showing Amount of Power Turned Back Into Line When Lowering Load From Bridge Regeneratively

Gas-Electric Wrecking Crane. Two modern double-ended gas, electric, storage-battery wrecking cranes are being built to meet peculiar conditions in electrified zones of the Cleveland Union passenger and New York Central Riverside freight terminals. Each crane weighs 436,000 lb.; is 67 ft. long; is carried on four 4-wheel trucks equalized in pairs; and will handle 105 tons through a 180-deg. arc at either end of the crane. The motive power consists of two 220-hp. gasoline engines direct-connected to electric generators, plus a 208-cell storage battery of large capacity. The propelling equipment is comparable to that of electric locomotives.

Railway Ditcher. A heavy-duty, high-speed railway ditcher and maintenance crane has been developed by the Industrial Brownhoist Corporation for traveling up to 18 miles per hour and handling a spreader, in addition to working between 30-yd. air-dump cars, and cleaning out ditches with a 1¼-yd. shovel,

[14] Section prepared by H. H. Perry, Vice-President, Industrial Brownhoist Corporation, Cleveland, Ohio. Assoc-Mem. A.S.M.E.

which reaches to the far end of the dump car. It handles its own train of tender, dump cars, and spreader, from work to dump and back. The shovel and crane boom are readily interchangeable. The machine weighs about 200,000 lb. and, as a crane, will lift up to 60 tons. The engines are 10-in. × 12-in., and steam is supplied by a 75-in.-diam. 200-lb. fire-tube boiler.

Railway Maintenance Crane. A number of railway maintenance cranes of a new type, Fig. 70, have been delivered by the same company to one of the eastern roads, whose object was to obtain the largest possible lifting capacity without outriggers and without fouling passing traffic by a long rear overhang. The crane carries a machinery unit and boom, operator's stand, and counterweight, all of which revolve through about 110 deg. either side of the track center line, without projecting the rear end beyond clearance. The boiler, tanks, and auxiliary equipment are on the other end of an unusually long and heavy car, mounted on trucks made specially heavy to carry the crane loading, which is concentrated at one end. The maximum free capacity is 12 tons at 15 ft. radius. The crane weighs about 200,000 lb. when equipped with an 80-ft. boom.

Heavy-Duty Pile Driver. To meet the needs of railroads in replacing wooden trestles with concrete structures of economical design, a special, heavy-duty pile driver has been developed to drive large concrete piles and set large concrete-slab decks upon the piles after driving. This machine, shown in Fig. 71, is manufactured by the Industrial Brownhoist Corporation.

The machine is a compact, heavily built, full-revolving, railway pile driver with detachable truss and leaders, which may be replaced by a heavy, curved crane boom, when the machine will handle loads about equivalent to a 120-ton wrecking crane with boom swung over the end of the car. The crane boom is used for setting the concrete-slab decks.

All the desirable features of the most modern railway pile drivers are present in this machine, including a locomotive-type boiler and a two-speed travel mechanism. The leaders fold up and down quickly, and the driver handles itself alone on the main line. Inclined piles may be driven, not only in a plane at right angles to the track, but in a plane parallel with the track. The maximum leader capacity is 22,000 lb. at 31 ft. 11 in. radius.

Crawler-Mounted Shovel Crane. An impressive 2-yd. heavy-duty crawler-mounted shovel crane, or dragline trench hoe, has been introduced by the Link-Belt Company. Its maximum lifting capacity is 40 tons at 12 ft. radius, and the machine weight 125,000 lb. as a crane. The maximum dragline pull is 40,000 lb. The crane is illustrated in Fig. 72.

Bridges, Car Dumpers, Towers, and Handling Systems for Bulk Materials[15]

Greater attention has been given recently to the economic phases of installations for bulk handling, where formerly the technical features received the closest investigation. This demand for high economy has resulted in mechanical refinements, such as pressure lubrication, roller bearings, heavier construction, special selection of metals, etc., to reduce maintenance charges. Likewise a more general use of generator-voltage control—the advantages of which afford simpler control apparatus, greater accuracy of speed control, regenerative retardation, low maximum kva. demand and kw-hr. consumption per ton of material handled—reflects a reduction in operating costs.

Car Dumpers. The car dumper built for the New York Central Railroad at Toledo, in 1929, by Heyl & Patterson, Inc., represents a distinct advance over previous machines for loading coal into Great Lakes freighters. Novel features are the car retarder on the cradle and the new arrangement of ropes and sheaves.

The car retarder is made up of nine units (six having individual control), so that the desired braking pressure can be exerted against all wheels of the incoming loaded car without gripping the wheels of the empty car as it is leaving the cradle. The loaded car can be stopped from a speed of 8 ft. per second in a distance of about 22 ft.

Instead of the usual gravity "hump" yard requiring a crew of car riders, this dumper is fed from a flat yard by electric pusher locomotives. Standard railway retarders and electrically operated switches on the run-off track control the passage of empty cars to their yard. This arrangement of yard with the efficient retarder on the dumper gives higher speed of operation at reduced cost.

The rope system designed for this dumper, using $1^1/_8$-in. rope on 75-in. sheaves with forged high-carbon-steel grooved slip rings, together with a new arrangement or location of sheaves, for the first time in the history of the Great Lakes coal trade has permitted the dumping of over 5,000.000 tons of coal with one



Fig. 75 General View of Coal-Dock Gantry Bridge

set of ropes (which from present indications will handle 7,000,000 tons). The best average for previous dumpers has been slightly over 2,000,000 tons.

The maximum rate of dumping is one car per minute (cars up to 120 tons capacity), and the average over cargo, including time for shifting the boat, etc., is 40 cars per hour.

The dumper is electrically driven with a Ward Leonard system of control for the cradle and Barney haulage engines, each of which has two 500-hp. motors. Forced ventilation of filtered air to the motors and motor-generator sets prevents accumulation of coal dust in the windings.

Unloaders. A noteworthy development by the Wellman Engineering Company, in 1929-1930, is incorporated in the new Hulett unloaders recently installed by them at the Presque Isle Dock of the Hocking Valley Railway Company (now C. & O.) at Toledo, Ohio, and shown in Fig. 73.

The general type of these machines is the same as past installations, the machine traveling on runways having a span of 72 ft. 6 in., and having the trolley with walking beam and stiff bucket leg with 17-ton ore shells. All machines of this type in the past have been operated by direct current, but in this case a radical departure is made and all the functions of the machine, including bridge travel, larry travel, hopper disks, larry gates, and bucket-leg rotation, are operated at 440 volts by 3-phase 60-cycle 6600-volt alternating current which is stepped down from the incoming line at the substation.

The remaining functions, including beam hoist and trolley travel, are operated by direct current at 275 volts, with variable voltage control, this current being supplied by a 3-unit motor-generator set located on the walking beam. This set includes a 500-volt synchronous motor, operating on a primary voltage of 440 volts, 3-phase, 60-cycle, flexibly connected to a 200-kw. d.c. generator

[15] Section prepared by C. D. Bray, New York, N. Y., Vice-President of the Mead-Morrison Manufacturing Company, East Boston, Mass. Mem. A.S.M.E.

for the hoist, and a 125-kw. d.c. generator for the trolley travel.

The advantage of variable voltage control is fully realized in this installation, and the use of alternating-current motors for the functions described seems to produce an operation which is entirely satisfactory.

Bridges. Among the outstanding developments in the past year is one connected with a large gantry bridge designed and built by the Mead-Morrison Manufacturing Company for the Consolidation Coal Company (North Western Fuel Company) for their Milwaukee, Wisconsin, coal dock. This bridge handles a 10-ton coal grab and has been developed to operate on 440-volt, 3-phase, 60-cycle alternating current. Most of the man-trolley bridges developed in recent years have been for use on direct current, in order to obtain the advantages recognized as inherent in direct-current service.

This alternating-current bridge, however, has more than met expectations, in that the combination of regenerative braking with graduated air braking gives full control of all movements of the man trolley itself, as well as of the grab bucket. This system has worked out very successfully, as is indicated on the recording graphic chart, shown in Fig. 74, which shows clearly the amount of power turned back into the line when lowering regeneratively. The power consumption itself, per ton handled, is as low, if not lower, than on any existing direct-current bridge. The man trolley is fully equipped with Timken roller bearings.

Fig. 75 shows the general outline of the bridge, and Fig. 76 the 7-ton clean-up grab, which has a spread of 24 ft. and a cutting edge of 10 ft.

Shiploader. A loading tower of novel design has been placed in operation by the Robins Conveying Belt Company, New York, at the plant of the Alabama State Docks Commission in Mobile. The outstanding feature is its flexibility and range of operation, which are such that, in addition to cargo loading, it is applicable to bunkering operations, reaching all hatches irrespective of location, a feature which most loading towers are lacking.

The tower, which is movable along the dock face, is arranged to receive material from a trunk-line conveyor and deliver it to ship or to storage inshore over the loading belt. The loading conveyor is mounted in a shuttle frame, which, in turn, is supported in a pivoted cradle, which swings through an arc 20 deg. above and 20 deg. below the horizontal.

In operation, the tower is moved to the proper spot, the cradle swung to the desired angle, and the shuttle is then run out between the shrouds, over the ship's rail or life boats, and the end of loading belt is registered over the hatch. Further flexibility is provided by a hinged telescopic chute. For delivering to storage, the shuttle is run inshore and the conveyor direction is reversed.

Industrial-Railway Equipment[16]

In quarries, a growing use is being made of industrial cars for hauling stone from the quarry face to the crusher. Bodies of similar sturdiness have also been mounted on trucks used for the same purpose. The opportunity is opening up to apply bodies of these special types to contractors' trucks used for handling stone dug from excavations for buildings in large cities, and in other types of construction work. Rough bricks and heavy stones likewise can be handled by the same equipment.

In the handling of transformers in large electric power plants there is used what is known as a transformer transfer car. This is a special car built for a special purpose and ranging in capacity from 20 to 150 tons.

[16] Section prepared by W. E. Farrell (New York, N. Y.), President Easton Car & Construction Company, Easton, Pa.

Practically all the newspaper plants erected in recent years use industrial cars and track for handling newsprint into storage, and out of storage to the reels on the presses. A notable installation of such equipment is in the new building of the *Daily News* in New York.

In the erection of most tall buildings, the concrete for the floors has been handled by a kind of a two-wheel scoop body, but for handling this material in the new Empire State Building in New York City, dump cars, industrial track, and turntables have been used.

Large-capacity structural turntables find many applications for industrial purposes. These turntables vary in capacity from 25 to 150 tons, range from 8 to 15 ft. in diameter, and are sometimes equipped with standard-gage track of heavy rail. The General Electric Company and John A. Roebling's Sons Company use such turntables for handling heavy reels of wire at points where it is desired to change the direction of rolling of such reels.

Development in Aerial Wire-Rope Tramway Construction[17]

Double Reversible Tramway. An instance of the adaptability of aerial wire-rope transportation is found at the sand and gravel plant of J. F. Fitzpatrick, Auburn, Massachusetts. A double reversible type of tramway, having a capacity of 75 tons hourly, was installed by the American Steel & Wire Company. It carries raw material from the pits to the screening and washing plant on top of the storage bins. About a year ago a single reversible tramway was installed to take excess sand and gravel from the bins, at a point just above the truck-loading chutes, and form storage piles for contract shipments and winter use.

Recently, a ready-mixed concrete plant has been installed. The aggregate bins were located under the track cable of the single reversible tramway. The operator who loads the bucket is provided with a set of push buttons by which he throws the tripper into place over the proper bin. The tramway bucket automatically dumps its load, and the operator returns the bucket to the loading terminal by means of the drum control of a slip-ring-type induction motor. Whenever there is an excess accumulation of either sand or gravel that is not required in the concrete mixing plant, the bucket is run out over the storage piles and dumped automatically.

Automatic Tripping Carriage. An automatic tripping carriage has been developed by the American Steel & Wire Company which has resulted in direct economies in the operation of reversible tramways. Merely by reversing the direction of the tramway carrier, the carriage operates levers on the bucket, which discharges the load before the carriage has traveled 30 in. in the reverse direction. The operation is simple. As the loaded carrier runs out toward the dump pile, a revolving ratchet passes by a beveled pawl at each revolution of the carriage wheel. When motion is reversed, the ratchet and pawl engage. The pawl is mounted in a chain wheel which is made to revolve with the carriage wheel, gradually winding up a short length of chain connected to the bucket latch system. The motion thus obtained is utilized to trip the load. While the chain wheel has slowly revolved, the pawl mounting has engaged a stationary cam, which gradually raises the pawl out of engagement with the ratchet.

The outstanding advantage of this device is that the dumping point on a reversible tramway is not dependent on any attachments on the track cable. In the case of a single reversible, or one-bucket, tramway the operator may change the dumping

[17] Section prepared by E. J. Lloyd, Engineer, Tramway Engineering Department, American Steel & Wire Company, Worcester, Mass.

point at will, making it possible to increase the hourly tonnage during rush periods by making short hauls, and filling up the more distant end of the pile during slack periods. Since it is no longer necessary to send a man out in a bucket, or on the cables, to change the position of a tripper, the safety of operation is greatly increased. When the dumping point of a double reversible, or two-bucket, tramway is changed, it is necessary to release the traction-rope clamp on one bucket and run the other to the new dumping point.

Banana-Plantation Tramway. A tramway has been installed at a banana plantation in an 8300-ft. circuit, capable of being extended to a 25,000-ft. circuit. Single stems of fruit weighing about 70 lb. each are loaded at supports about 300 ft. apart while the rope is in continuous motion. The rope may be run in either direction, to accommodate the harvesting operations, the direction being determined by the shortest haul to the discharge point. Motive power is supplied by a 30-hp. Diesel engine geared to an 8-ft. plain-groove sheave. Constant tension is maintained by



FIG. 76 CLEAN-UP BUCKET USED ON BRIDGE SHOWN IN FIG. 75

passing the rope around a sheave which is mounted in a light steel structure, in turn supported on standard-gage railroad trucks, and the whole attached to a suitable floating counterweight. Here we have an out-of-doors counterpart of the overhead conveyor chain, capable of being loaded and unloaded while in motion. The field of application of such a system has yet to be fully explored. For instance, bags of coffee or light mechanical parts can be carried long distances.

Colliery Waste-Disposal Tramway. A waste-disposal tramway of the continuous type was designed by the American Steel & Wire Company and installed late last year by the Houston Collieries Company, near Kimball, West Virginia. It is illustrated in Fig. 77. A special feature is that loads weighing about 4600 lb., gross, are raised vertically 443 ft. in 879 ft. horizontally. Taking the cable slope into account, the maximum slope is 78 per cent. The friction grip is required to develop a minimum resistance to slippage of 2830 lb. This is probably the most severe duty required of any similar system in the United States.

After the buckets ascend to the top of a ridge, they pass over an almost level 1600-ft. span, at the end of which is a structure containing track-cable counterweights and a 16-ft.-diameter sheave. The buckets pass around this sheave automatically at a speed of 500 ft. per min. and return to the loading terminal. A 36-in. feeder conveyor with push-button control loads the buckets instead of the usual bin gate. Buckets are sent out from the loading terminal by an electrically operated bucket spacer and dispatcher, which eliminates much manual labor and insures uniform spacing.

Grip Sheave for High-Speed Transmission of Power by Wire Rope. A notable improvement has been made in the design of a grip sheave for the high-speed transmission of power by wire rope under high tension. The Hallidie design of grip sheave

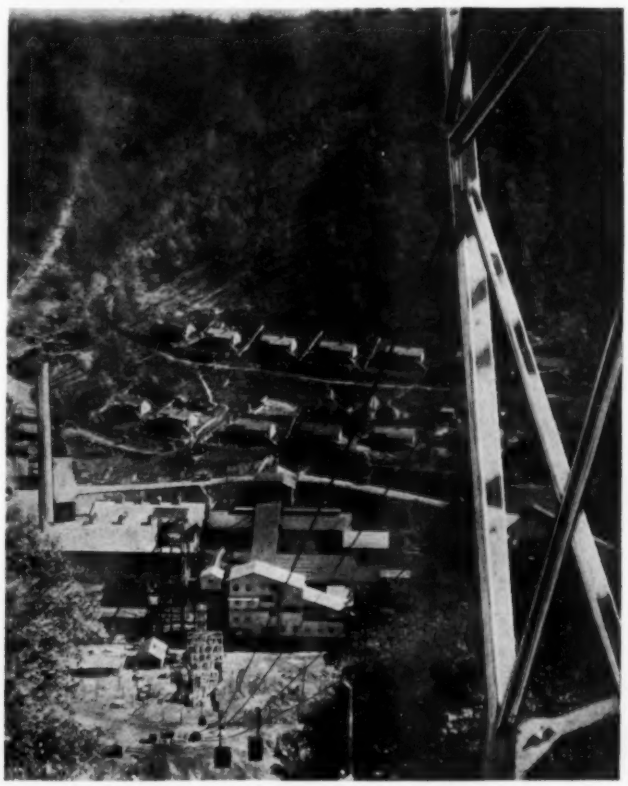


FIG. 77 COLLIERY WASTE-DISPOSAL AERIAL TRAMWAY OF THE CONTINUOUS TYPE

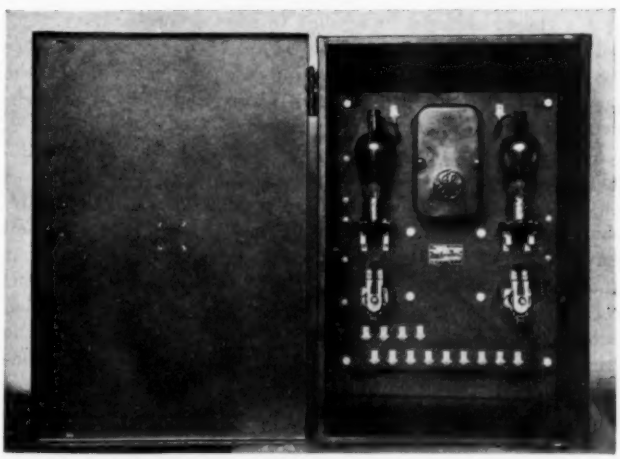


FIG. 78 THYRATRON CONTROL FOR COKE CONVEYOR

has generally been used in this country where a half-lap drive was necessary, and where the power to be transmitted could not safely be entrusted to a plain grooved or wood-filled sheave. At rope speeds over 700 ft. per min. the wear on the toggle jaws, pins, and segments increases rapidly.

When confronted with the problem of transmitting 250 hp. at 1800 ft. per min., the American Steel & Wire Company designed a grip sheave in which the jaw pins are eliminated. Each jaw is machined all over. Two pairs of jaws are placed in each of 48 machined pockets. The side-clearance tolerance is 0.007 in., maximum. Adjustment for rope size is secured by shimming back of the jaw retainers. The jaws are chrome-nickel-steel castings, heat treated to give a Brinell hardness of about 450. The sheave runs in a vertical plane, completely enclosed, and is splash lubricated. In operation, it is practically noiseless, and the rope shows less than normal wear after several months' use. There is no apparent wear on the sheave.

Application of Electrical Equipment to Materials-Handling Machinery[18]

From the standpoint of electrical apparatus, materials-handling equipment may be divided into two classes: first, that which

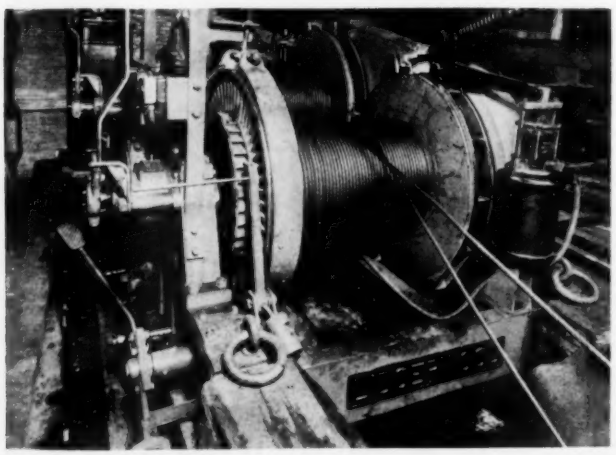


Fig. 79 Contractor's Electric Hoist With Electrically Driven Hydraulic Device for Operating Safety Emergency Brake

is driven by simple stock equipment, such as a single-speed motor and starter; and, second, that which requires special equipment which must be applied to meet a specific duty.

There is little chance for progress in the first class, so far as the application of electrical equipment is concerned. In the second class, opportunities frequently arise, and some accomplishments have been made in the last year.

Electronic Tubes. Perhaps the most interesting and novel development has been the use and proposed use of electronic tubes for controlling materials-handling machinery. In one case, where the amount of material fed by one conveyor on to a second conveyor must be kept proportional to the amount of a different kind of material already being conveyed by the second, this result is accomplished by a thyratron control, Fig. 78, which varies the speed of the first conveyor motor in proportion to the weight of material conveyed by the second. The second conveyor crosses a weigh pan located ahead of the discharge of the first conveyor, and the slight up and down movement of this pan due to the varying weight of material controls the grid circuit of the thyratron, which in turn controls the speed of the d.c. motor on the second conveyor by varying its field. Thus the proportion of the two materials mixed together on the second conveyor is kept very close to constant.

A photoelectric tube is used to provide interlocking between two machines on a grain-car unloader so that the second movement cannot start until the first is completed. It gives greater accuracy than a limit switch will give for this duty, avoids mechanical and electrical wear and tear, avoids the use of gearing or chain drive, and a failure of the tube or light source introduces a safety feature, because it prevents operation of the second motion, thus notifying the operator that it needs attention.

Other applications of electronic tubes have been proposed for controlling materials-handling machinery. Some of these, no doubt, will go into operation. All electronic-tube applications of this kind should be scrutinized carefully, however, because the tubes are a relatively new development and a few more years of experience in the industrial field are necessary to demonstrate their specific fields of greatest usefulness.

Remote Control. An interesting application of remote control has gone into successful operation on the cableway at the Owyhee Dam in Oregon. A 400-hp. induction motor, driving the hoist and trolley motions of the cableway, and a 75-hp. motor, driving the tower, are accurately controlled, including the operation of the air brakes and clutches, from a point 1300 ft. distant from the motors. Full-magnetic reversing control is used, and the air brakes and clutches are handled through electrically operated valves. This is believed to be the longest distance through which the control of a complicated, heavy reversing duty has ever been accurately handled by any means.

Another interesting remote-control application is found on a riderless transfer-car system for handling stone out of the quarry of the Trinity Portland Cement Company. There are four cars, each propelled by a.c. motors with solenoid brakes, operating on a track about a mile long between the shovel and the crusher. This track is divided electrically into about 20 sections and has a turn-out about the middle for passing other cars. The cars are controlled entirely by men in two towers, and by the shovel operator who has control of the track sections adjacent to the shovel. He can thus prevent collisions while loading, dispatch loaded cars, and bring empties to the shovel for filling. The two towers control all other operations, including switching, and the cars are kept correctly spaced by the inherent speed characteristics of the a.c. motors.

Generator Field Control. The use of generator field control (often known as Ward Leonard) continues to expand on materials-handling equipment and has been successfully used on coal and ore bridges during the year. In one place it replaced straight a.c. drive on a bridge, and an increase in capacity was obtained, with greatly reduced maintenance work, and a decrease in power consumption. A car dumper equipped with generator field control broke all records for dumping coal on the Great Lakes in August, 1930. Two hundred and forty-three cars (12,763 tons of coal) were dumped into a boat in 4 hours and $12^1/_2$ minutes, including all time for shifting the boat. This is an average of $57^3/_4$ cars per hour. The cost per ton was considerably less than with steam operation.

Most blast-furnace skip hoists installed within the year have used generator field control, and it has been ordered for use on a Hulett ore unloader for the first time. High-speed, high-capacity coal towers have been using it successfully for several years, and on a repeat order placed during the year, generator field control was again selected, due mainly to its freedom from delays, low maintenance cost, low power consumption, and flexibility of operation.

Ballast Cleaner. A rather unusual type of materials-handling equipment using electric drive is a railroad ballast cleaner which has operated long enough to prove its worth. This machine digs the rock ballast from between the tracks with grab buckets, which dump it on a conveyor, from which it goes through a screening and cleaning process. It is then deposited between

[18] Section prepared by James A. Jackson, Industrial Engineering Department, General Electric Company, Schenectady, New York.

the tracks again. The entire machine is pulled by a locomotive to the place where ballast is to be cleaned. The locomotive then runs ahead and sets its brakes. A steel cable is attached to the locomotive and goes to an electric winch on the front end of the ballast cleaner. The entire cleaner is pulled slowly along by operating this winch, while electric motors operate all other motions of the cleaner. Electric power is supplied from a gasoline-engine-driven d.c. generator.

Safety Emergency Brake. A novel use is made of a small electrically driven hydraulic device, shown in Fig. 79, for operating a safety emergency brake on contractors' hoists in New York. The setting of an emergency brake must not occur too abruptly, because this action may break the rope or overturn the derrick. When power is cut off from the hydraulic-device motor, the stored energy of the revolving armature and centrifugal pump causes the rotation to continue, but at a diminishing rate, thus applying the braking torque at a rate slow enough to prevent damage to the equipment, but fast enough to meet the safety requirements. This equipment has passed severe tests successfully.

Variable-Speed Transmission Systems as Applied to Materials Handling[19]

New Designs. In the development of the variable-speed transmission, mention should be made of several new designs which have increased its utility, making it adaptable for many special requirements. These new designs include: an electrical remote control, through which the speeds may be regulated from one or more push-button stations; an electrical automatic control, which provides entirely automatic speed regulation for certain requirements; an enclosed design, in which the operating parts are encased for protection against steam, acid fumes, dust, grit, and dirt in severe-service installations; a vertical unit, designed to occupy minimum floor space, and which may be enclosed and equipped with motor base; and a special unit with frame designed to receive motor and speed reducer, this forming a variable-speed power-driving unit.

The electrical remote-control variable-speed transmission makes it possible to locate several push-button stations at convenient points throughout the length of a conveyor, thus eliminating the necessity for the foreman to go to the driving end each time a speed change is necessitated. For other requirements the control stations for several conveyors, or sections, may be centrally located on one master control board, on which electrical tachometers may be mounted, indicating the speeds of each separate conveyor unit.

In general, the need for speed regulation is determined by: changes in operations; number of workmen on the "line;" change in size, shape, or construction of product; fluctuations in temperature; variation in quality of material in process; skill of workmen; exact synchronizing of two conveyor sections; change in composition of material; and changes in complete production schedules.

New Installations. The following descriptions of typical recent installations will serve to illustrate several special requirements, the results obtained, and the savings effected.

Sign Enameling. In a sheet-metal works manufacturing enameled metal signs, these signs are delivered from the stamping and enameling machine on to a small belt conveyor. From this conveyor the signs pass to a second conveyor that carries them through a continuous oven to bake the enamel. This second conveyor is on a lower plane than the first belt conveyor, and is equipped with vertical pins, spaced at regular intervals, between which the signs stand on edge while being carried through the oven. A Reeves variable-speed transmission is used not only to vary the speeds of the conveyors for different sizes and kinds of signs, but also to synchronize the two conveyors so that the signs will always drop between the pins in proper position.

Previous to the installation of the variable-speed transmissions it was necessary to use a workman at the intersection of the two conveyors, in order to guide the signs into position on the second conveyor. The variable-speed units timed the conveyors so accurately that the signs dropped into position perfectly, and this workman was found unnecessary.

Veneer Cutting. An interesting application of the variable-speed transmission is in connection with the conveyor which is used to carry the veneer away from a rotary veneer lathe as it is cut. Ordinarily, the lathe is so speeded that the log turns about 40 r.p.m. constantly. At this speed, on a log 24 in. in diameter,

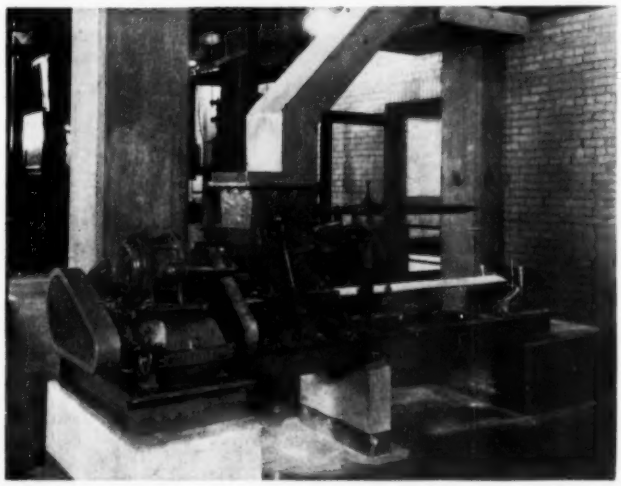


Fig. 80 Poidometer or Automatic Weighing Machine at Municipal Water-Purifying Plant, Minneapolis, Minn.

the lathe cuts 250 ft. of veneer per minute. As the log decreases in diameter, the production of veneer declines in exact proportion, until, at a 6-in.-diameter core, it is only 62.5 ft. per min.

In order to utilize a conveyor for taking the veneer away from the lathe, the conveyor speed must be synchronized with the veneer production and automatically reduced in exact proportion to the output of veneer. This is accomplished through a somewhat special type of variable-speed transmission just recently developed. Instead of the standard speed-shifting screw and handwheel control, a wedge-shaped cam is mounted between the ends of the shifting levers which govern the position of the disks and regulate the speeds. This cam is actuated by means of a connection to the knife carriage of the lathe, so that, as the knife moves into the log to cut veneer, the cam acts on the variable-speed transmission unit to slow down the conveyor automatically in exact proportion to the decline in production of veneer. As the knife carriage moves backward, the cam moves with it, thus increasing the conveyor speed in proportion to the diameter of the next log to be cut.

Bacon Slicing. At the plant of Oscar Mayer & Company, Chicago, short packing conveyors are used in conjunction with bacon slicers, both driven with separate Reeves variable-speed transmissions, which, however, are shifted simultaneously. Through this exact timing of the conveyor with the slicing machine it is possible to lay the bacon strips automatically on the conveyor in such precise manner that the packers, girls, who sit

[19] Section prepared by D. W. Clem, Vice-President and Manager of the Reeves Pulley Company of New York, Inc. Assoc-Mem. A.S.M.E.

along the edge of the conveyor, can pick up the correct amount of bacon of all sizes and thicknesses, to fill the box, and the weight will be uniform. The synchronization of speeds is responsible for this exact "lay," as it is called.

Poidometers. The variable-speed transmission has come into general use for regulating the feeding conveyor on automatic weighing machines or poidometers, such as built by the Schaffer Poidometer Company. These poidometers, of various sizes, are used in handling cement, lime, heavy chemicals, stone, etc., the material being fed accurately by weight instead of by volume. Fig. 80 shows a poidometer installed at the Municipal water-purifying plant Minneapolis, Minnesota.

The material, deposited through a hopper, trips an automatic scale beam when the predetermined weight has been deposited on the conveyor. By reason of the varying specific gravity of the materials, it is necessary to be able to regulate the speed of the conveyor in order properly to remove the materials being fed. Each poidometer is equipped with a recording device which

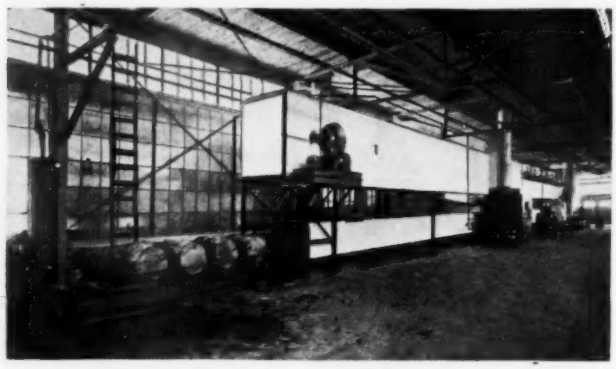


FIG. 81 DRUM-HANDLING EQUIPMENT PROVIDED WITH VARIABLE-SPEED TRANSMISSIONS

enables the operator to determine just what amount of material has been handled in any given unit of time.

Automobile Washing. A recent development in automobile washing, or laundry, conveyors is represented in the circular type called the "Merry-Go-Round," installed by the Sinclair Refining Corporation, in Cleveland. While conveyor systems for automobile washing, equipped with variable-speed transmission, have been in use for approximately three years, this is the first circular type, where the cars are discharged through the same door at which they enter.

Automobile Assembly. A recent development in the automotive industry has been the introduction of the American Austin "Bantam" car. Materials-handling systems are extensively used in the manufacture of this small automobile, and Reeves variable-speed transmissions are used on motor assembly, chassis assembly, and body "lines," timing the production accurately to the desired production schedule, and eliminating overproduction in any one department.

Foundries. The use of conveyor systems in foundries is expanding rapidly, and there is a possibility that foundry mold conveyors may ultimately revolutionize this industry. Variable-speed control on mold conveyors is essential because of changes in size of castings and consequent variation in length of time required for the mold to "set."

Paint Drier for Steel Drums. The drying operation is of major importance in the painting of steel drums. After being tested, the drums must pass through a predrier for drying off surface moisture before going to the painting machine. The drums are received at the painting machine, where they are spray painted as they revolve on their chimes. The ideal painting machine will handle all types and sizes of drums, rolling them on their chimes only.

After painting, the drums go through another drying oven, where they are dried uniformly and hard. The conveyors that carry the drums through the ovens must be capable of traveling at various speeds, depending on the oven temperature, method of heating oven, and kind of paint used. By means of the variable-speed transmission the conveyor also keeps in step with the other operations in drum handling. The speed through the predrier is controlled so that the drums do not pile up in front of the painting-machine operator before he is ready for them. Also, the drying rate of the drums after painting may be varied to suit delivery conditions at the discharge end of the oven.

In Fig. 81 are shown some units of Bartlett & Snow drum-handling equipment equipped with Reeves variable-speed transmissions. The tested drums are entering the predrier, through which they travel to the painting machine. The freshly painted drums of all sizes are then conveyed through the drying oven, located immediately above the drier.

Vanishing-Cream Conveyor. The conveyor installation at the Ponds Extract Company, Clinton, Connecticut, holds interest not only in the slow speed used and the nature of the product handled, but also in the amount of material handled, the total weight of the fully loaded conveyor amounting to 10,000 lb. of vanishing cream in jars while cooling after filling, this whole load being handled by a $^3/_4$-hp. motor.

The jars for filling are placed on a short cross-conveyor, whence they feed into the filling machine. They are placed on the main conveyor, from the filling machine, by hand, closely arranged on the belt, which moves from 10 in. to 30 in. per minute for a total distance of approximately 175 ft. This belt, 30 in. wide, is closely packed its entire length with these jars.

At the tail end of this belt the jars are pushed by their movement across a short plate, with its farther edge slanted to effect a staggered arrangement, on to a very short length of 30-in. belt conveyor, moving at just twice the speed of the main conveyor. This short conveyor, together with the preceding plate, acts as a feeder on to the cross belt at the right end, from which the product is finally removed for packing.

Metal-Stamping Conveyor. Another interesting installation is that of a 4-in. I-beam caterpillar-type monorail conveyor for handling stampings, installed at the Kelsey-Hayes Wheel Company, Detroit, Michigan, by the Whitehead & Kales Company.

The conveyor carriers are spaced on 32-in. centers and consist of suspended steel hangers with four shelves. The three upper shelves hold six stampings in their various stages and the lower shelf holds six blanks. The driving mechanism consists of two individual caterpillar units. Each drive has a Reeve variable-speed transmission with separate motor, the motors being wired to the same starter. This installation furnishes an example of the very accurate timing which can be secured by means of the variable-speed transmission, keeping the drives in step and giving production control.

Paint Drier for Automobile-Wheel Rims. A corner-sprocket-drive conveyor installation, shown in Fig. 82, has been made at the Studebaker plant at South Bend, Indiana. It is equipped with a typical Reeves variable-speed transmission. The conveyor is 450 ft. long, with a mean speed of 12 ft. per min., and it handles hand-painted rims through the drying ovens. The variable-speed feature is necessary for production and proper drying, according to the characteristics of the plant, oven temperature, and atmospheric conditions.

ACTIVITIES OF THE MATERIALS HANDLING DIVISION

The Materials Handling Division has carried on a number of important activities during the past year, among which those

outlined below are considered to be of particular significance.

Labor-Saving-Equipment Formula. The study and development of the labor-saving-equipment formula, also known as the materials-handling formula, has been furthered this year through the action of the subcommittee consisting of S. H. Libby, Chairman, Harold V. Coes, William F. Hunt, James A. Shepard, and Fairfield E. Raymond. By action of this committee the detailed study of the formula has been assigned to the Department of Economics of the Massachusetts Institute of Technology, where the work will be under the direct personal supervision of Professor Raymond. Professor Raymond is likewise a member of the Management Division Subcommittee on Formulas, and in this way the research activities on the labor-saving formula will be coordinated with similar studies to be carried out under the auspices of the Management Division committee. For the financing of the work on the materials-handling formula the project is being submitted to the Research Division of the Society for action.

Manufacturers' Committee of the Materials Handling Division. Undoubtedly the most important move made by the Division during the year has been the organization of the Manufacturers' Committee of the Materials Handling Division. The chairman of the Division, after considerable discussion and correspondence regarding the need and opportunity for cooperation among manufacturers in the field of materials handling, to further the science and its application throughout industry, laid a proposal before the Executive Committee of the Division early in 1929, and was authorized to proceed with its development. A dinner meeting was held at the National Management Meeting in Detroit in May, 1929, and the manufacturers present cordially endorsed the suggestion and indicated a willingness to cooperate with the Materials Handling Division in some practical plan to attain the objective. Individual units of the materials-handling field, among which are the electric-crane manufacturers, the hoist manufacturers, and the electric-truck manufacturers, already had trade associations. For this reason, and also because the interests of the entire industry were too widespread for united action along commercial lines, it was decided to carry out the development of cooperation along professional, engineering, and educational lines, under the auspices of the Materials Handling Division, rather than to attempt to form a large trade association for the industry. Moreover, the Division could not ethically participate in the advertising and sales-promotion activities that naturally are performed by a trade association.

A meeting to organize the Manufacturers' Committee of the Materials Handling Division was called on Friday, June 27, 1930, in New York. To this meeting representatives were invited from various representative manufacturing concerns and from trade associations in the materials-handling field. An attempt was made to give representation at this initial meeting to every important division of materials-handling equipment. Seventeen men attended the meeting, which was presided over by R. H. McLain, of the General Electric Company, organizing chairman, and a former chairman of the Materials Handling Division. The following program was observed at the meeting:

1 Call to order by the chairman
2 Presentation of the Society's viewpoint. R. H. McLain
3 Place of the Materials Handling Division in the Society. P. T. Wetter, Assistant Secretary, A.S.M.E.
4 Plan of coordination between the Manufacturers' Committee and the Materials Handling Division. G. E. Hagemann, chairman of the Division
5 Presentation and discussion of tentative program of activities:
  a Nomenclature
  b Standardization projects
  c Simplification (reduction in number of types and sizes of parts)
  d Research
  e General classification of equipment
  f Collection and publication of engineering data
  g Development of the economic phases of materials handling
  h Development of cost-accounting methods for materials handling
  i Securing papers for materials-handling meetings
  j Collecting leads from which magazine articles could be prepared
  k Arranging for lectures in colleges
  l Ethics concerning engineering services to prospect users
  m The field of the consulting materials-handling engineer

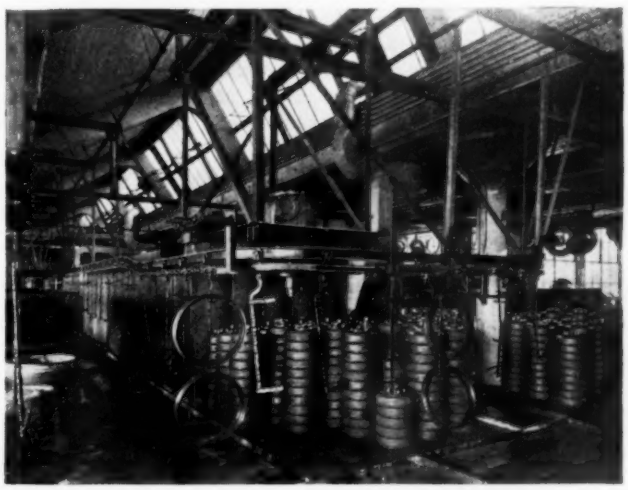


FIG. 82 CORNER-SPROCKET-DRIVE CONVEYOR INSTALLATION IN AN AUTOMOBILE PLANT

  n Classification of materials that are handled in industry
  o Compilation of an engineering library on materials handling
  p Assistance to the Division in preparing the Annual Progress Report
6 Plan of organizing the committee on a permanent basis
7 Election of officers of committee
8 Fixing of dates and places for future meetings
9 Planning for a special session for manufacturers at the Annual Meeting in December
10 Other business
11 Adjournment.

It was decided that the committee should stand as at present constituted, until the representation could be more fully studied so that other groups not now included could be provided with representatives. Each definite field of materials-handling equipment eventually is to be represented by at least one man, and plans will be developed for the election, or appointment, of these representatives impartially from among the various handling-equipment manufacturing plants, with terms of office to run for regularly set periods. In other words, it is the aim of the committee to reach, interest, and secure the cooperation of every manufacturer of materials-handling equipment, so that the coverage may be complete.

Because the full committee of sixteen regular members seemed rather large for the holding of frequent meetings, it was decided to elect an executive committee of five to function in behalf of the whole committee. This executive committee consists of five members to serve at present for one year each the first term of office expiring in December, 1931. It was further decided that three officers should be elected—a chairman, vice-chairman, and secretary-treasurer—these to be three of the five members of the executive committee. The officers and members of the executive committee elected to serve until December, 1931, are:

Chairman, R. H. McLain, General Electric Company
Vice-Chairman, W. E. Farrell, President, Easton Car & Construction Co.
Secretary-Treasurer, C. B. Crockett, Secretary, Industrial Truck Association
Additional members of executive committee:
George L. Morehead, Vice-President, Link-Belt Company
F. J. Shepard, Treasurer, Lewis-Shepard Company.

After an extensive discussion, at both morning and afternoon sessions, of the various subjects that might properly come within the jurisdiction of the committee, and deciding to hold an open meeting for manufacturers at the Annual Meeting in December, the committee voted to adjourn to meet again in December.

Subcommittees of the Manufacturers' Committee. At an executive committee meeting held on September 9, the chairman of the Manufacturers' Committee, R. H. McLain, was authorized to appoint the following subcommittees to get the work of the group under way.

Subcommittee on Nomenclature and Classification of Materials-Handling Equipment:
George L. Morehead, Vice-President, Link-Belt Company
L. J. Kline, General Manager, Mercury Manufacturing Co.
S. Buckley, Shepard Niles Crane and Hoist Company.

Subcommittee on Performance Standards of Locomotive and Crawler-Type Cranes and Shovels:
H. H. Perry, Vice-President, Industrial Brownhoist Company
F. A. Smythe, President, Thew Shovel Company
A. J. Fries, District Superintendent of Motive Power, New York Central Lines, Buffalo, New York.

Subcommittee on Formulas for Figuring the Capacity of Containers:
W. E. Farrell, President, Easton Car & Construction Company
H. J. Brendlin, The Haywood Bucket Company
M. J. Andrada, Chief Engineer, The Robins Conveying Belt Company.

Subcommittee on Research:
Brinton Welser, Vice-President, Chain Belt Company.

Subcommittee on Materials Handled and Types of Equipment Used in Each Case:
M. W. Potts, Engineer, Alvey-Ferguson Company.

Subcommittee on Leads for Papers and Magazine Articles on Materials Handling:
C. B. Crockett, Secretary, Industrial Truck Association.

It is anticipated that from this beginning the Manufacturers' Committee of the Materials Handling Division will develop into a most effective medium to advance the professional, engineering, operating, and economic phases of materials handling in industry through constant and active cooperation with the Division. It is especially desired to stress the fact that the committee is absolutely impartial, and earnestly solicits the support and cooperation of every individual manufacturer of materials-handling equipment. Although actual individual representation on the committee by every manufacturing concern would make the committee far too large for effective operation, nevertheless each manufacturer is asked to consider himself and his company field members of the committee, with full liberty at any time to submit suggestions and otherwise participate in each of its activities in which he may be particularly interested.

National Meetings. The National Meeting, held annually in the spring at various important centers (Philadelphia in 1928, Detroit in 1929, Chicago in 1930, and Cleveland in 1931) continues to grow in interest. Commencing with the Detroit Meeting, the National Meetings have been combined with those of the Management Division because the fields of the two divisions are closely associated. Attendance at these meetings in the respective years was 300, 500, and 880, for those held thus far. The success of previous meetings; the care with which the 1931 National Meeting program is being arranged; the active cooperation of the Cleveland Local Section, the Cleveland Engineering Society, and a large local committee of manufacturers' representatives; and the opportunity for a number of most instructive inspection trips—all should serve to attract a still larger group to the 1931 meeting.

Materials-Handling Exposition. The first independent exposition of materials-handling equipment was held in Chicago in April, 1930, in connection with the combined National Management and Materials Handling Meeting. The show was arranged under the auspices of the Mid-West Exposition Association, because it was of a commercial nature and hence could not be sponsored by a division of the Society. It was endorsed and actively supported by the Materials Handling Division, however, and, in addition to its highly educational advantages, was satisfactory from an advertising and sales aspect to those who subscribed for space and exhibited.

The results in bringing manufacturer and user together to their mutual advantage were sufficiently definite to cause the management to plan a second exposition for the forthcoming National Management and Materials Handling Meeting to be held in Cleveland, April 13 to 17, 1931. At the time this progress report was prepared, 55 manufacturers had negotiated for exhibition space. The 1931 exposition should be unusually effective from the standpoint of the potential economies from efficient materials handling in the production cycle, which it will cause to stand out with particular emphasis. This feature is of special significance when dull business periods necessitate unusually careful measures to conserve the resources and energies of a company and to lower unit manufacturing costs to the minimum.

Discussion

C. M. Wilkinson.[20] The Executive Committee of the Materials Handling Division deserves much credit for the ably prepared Progress Report, for it represents an unusual amount of time and effort. It is especially noted that, in this report, the materials handling industry has been divided into several classes or groups. The Executive Committee for the Materials Handling Convention and Exposition in Cleveland has completed a classification which, it is believed, should be made the permanent basis or designating these classes or groups in the materials handling industry and which is as follows:

1 Electric hoists; 2, chain hoists; 3, overhead cranes; 4, electric trucks and tractors, 5, overhead monorail; 6, pneumatic conveyors; 7, package elevating and conveying; 8, overhead chain conveyors; 9, bulk elevating and conveying; 10, portable conveyors; 11, industrial railways; 12, gasoline tractors; 13, hand lift trucks and skids; 14, hand trucks and trailers; 15, locomotive cranes; 16, air hoists; 17, accessories.

Apparently only recently has the importance and size of the materials-handling industry as a whole or in part been understood or realized.

Attention was called recently to the fact that class 15, locomotive cranes, was doing a business of nearly $100,000,000 annually. With other classes, such as class 9, bulk elevating and conveying, doing a business of more than $15,000,000; with class 7, package elevating and conveying, having upward of $10,000,000 annually; with class 5, overhead monorail, doing upward of $5,000,000 in annual business, and the other classes doing business in much the same extensive way, one may obtain a comprehensive idea of the importance of the materials handling industry in the business of the country as a whole.

The materials-handling industry could well be called a basic industry, as there could be no buildings, steel plants, electric lights, etc., without first having some part of the materials-handling industry move the raw materials from locations where nature has left them, and then handle these raw products from clay banks, iron mines, coal mines, etc., to points of manufacture and thence to the ultimate consumer.

[20] C. M. Wilkinson Company, Akron, Ohio.

MH-53-2

Materials Handling in the Shoe Industry

By K. D. HAMILTON,[1] BROCKTON, MASS.

The growth of the shoe industry is described from its earliest establishment, from the point of view of mechanical handling of products. It describes the typical factory-production problem and the development of the mechanical conveyor in shoe factories. Mechanical drying and tempering of leather, as well as the adaptation of the individual motor to shoe machinery, are described, and also the installation of standard equipment for power house, dust collecting, packing, and shipping, to show that the shoe industry is searching for all methods to reduce production costs.



THE manufacture of shoes is one of the oldest in the country. During the pioneer days shoemakers went from town to town making shoes for the families in those early settlements. It was a hand-sewed shoe and simply designed, but substantially made.

Enterprising shoemakers in some of the larger towns opened small workshops near their homes. As the business increased, neighbors were called in to assist. These shoes were usually sold in a local market until the development of transportation made it possible to branch out for wider markets.

The machinery was elementary, although a few mechanical devices gradually had been developed. It was not until after the Civil War that the shoe industry really got its start and mechanical devices were invented to speed up production and to reduce cost, as well as to produce a shoe which would be better and more uniform in design and appearance.

Whenever an ingenious shoemaker developed a machine for performing an operation on a shoe he might form a company for manufacturing and marketing it. In a few years, therefore, there were several companies manufacturing shoe machinery of different types. Most of these have been absorbed into one company which manufactures the larger machines used in the industry today, although there are several independent manufacturers making machines for specific operations.

The trend in shoe machinery will be the development of equipment to speed up schedules and to simplify operations, eliminating as much as possible all hand work, in order to meet changes in style. New methods are being developed to eliminate operations, and thus there is a constant change in the design of shoe machinery. One instance is the development of the so-called stuck-on shoe, whereby the sole is cemented to the upper instead of being stitched on, as in a welt shoe.

Shoe manufacturers as well as machinery-equipment companies are beginning to find the need of economy through the introduction of new-process machinery and a speeding up of schedules in order to meet customer demand and style changes.

It was common practice 10 to 15 years ago to take from 20 to 24 days to make a pair of shoes. Today it is possible to reduce the schedule to 5 or 6 days in cheaper grades and to 11 to 15 days in the higher grades.

The method of moving shoes about the factory always has been by using a shoe rack. Shoes were made in lots of 24, 36, and 48 pairs. It was felt necessary to keep the shoes on the last a certain number of days to maintain fit and workmanship. Mechanical drying was considered injurious to the leather. Temperature and humidity control were unknown. The manufacturer realized that his shoes did not dry well in the damp, humid summer weather, while in the winter the factories were too warm, resulting in cracky sole leather and other difficulties.

The shoe manufacturer finally realized that it was necessary to reduce cost, to speed up the turnover of equipment and lasts, to reduce the amount of money tied up in goods in process, and to make better deliveries. Conditions were changing rapidly, and he could not continue on the 20- to 24-day schedule.

Conveying apparatus had been perfected for the automobile industry, and some of the more enterprising medium-grade shoe manufacturers looked into the possibility of mechanical conveyors. There were a few who made a start, these factories making in most cases a uniform grade of shoes of few styles and patterns.

It was found that in order to develop mechanical conveying it was impossible to introduce production schedules on a 24- or 48-pair basis as the racks were too large to be handled conveniently. A 6-pair rack was developed, and in some cases even a 1-pair rack system was anticipated. Other manufacturers reduced the manufacturing unit with a 12-pair rack system to obtain speed and a quicker turnover of goods in process. Reducing the factory unit was a radical step, but in actual practice a material saving in the time schedule resulted. Shoe production was kept flowing at a more constant rate with a smaller unit, and a greater output per operator was realized.

Another radical change was in the drying of the shoes. The time that the shoe was on the last was one of the retarding factors in increasing the production schedule. This resulted in the design and adoption by most of the progressive shoe manufacturers of drying and tempering equipment, as more uniform results were obtained with temperature-control apparatus. These radical ideas in the routing of shoes through the factory upset all accepted theories on the drying and conditioning of shoes on the last.

The problem of shoe-factory design and construction was one of the least troubles of the early shoe manufacturer. A building was put up or one was leased regardless of its practicability for the process, and the machines were located in the building without regard to light or power transmission.

Factory construction today, however, is thoroughly analyzed to conform with process requirements and machinery detail. Route charts for the material are made to conform to straight-line motion and efficient handling of product. The old-time factory was from 30 to 40 ft. wide, with headroom seldom over 10 ft., and the lighting conditions were poor and the windows

---

[1] Mechanical Engineer, George E. Keith Co. Mem. A.S.M.E. Mr. Hamilton was graduated from the Massachusetts Institute of Technology in 1913. He is a member of the American Society of Safety Engineers, the Plant Engineers Club of Boston, the National Fire Waste Council, and the Advisory Board of Engineers of the National Transmission Association.

Contributed by the Materials Handling Division for presentation at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, 29 West 39th Street, New York, N. Y. All papers are subject to revision.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

small. The factories of recent design are standardized, from 40 to 60 ft. wide, with headroom from 10 to 13 ft., insuring good light and ventilation.

The modern type of factory lends itself to the installation of conveying equipment because there is sufficient headroom for overhead ducts, driers, and conveying apparatus. It is wide enough to obtain three, four, or even six rows of machines, where



FIG. 1 ONE OF THE LARGER CONVEYORIZED SHOE FACTORIES

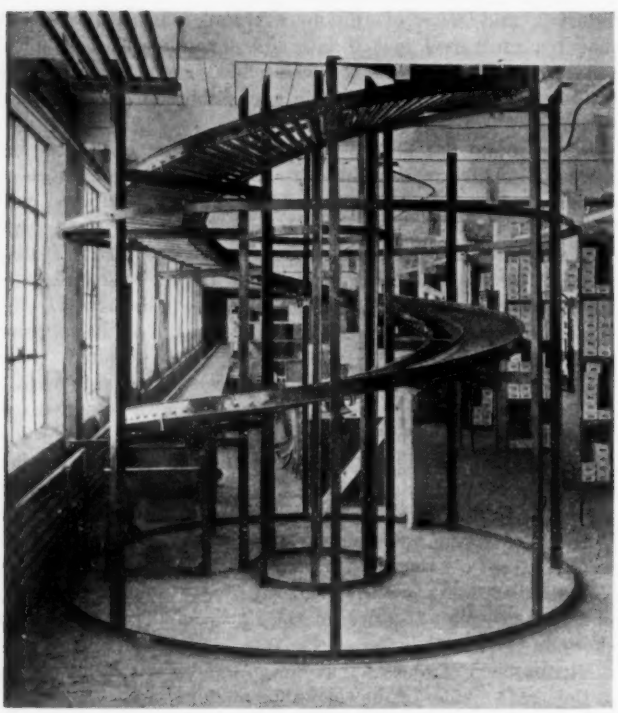


FIG. 2 SPIRAL CHUTE FOR MOVING GOODS FROM ONE FLOOR TO ANOTHER

the older factories could have only two rows of equipment, located near the windows of the outside walls.

The type of conveyor which was adapted for the major operations in the making room was of the roller design, and several installations were made in the larger factories. Some of these have since been discontinued because of change in manufacturing conditions, while others are being used with good results. Gravity rolls were used for flat sections, and the racks were sometimes moved by the operators, while in some other installations the rolls were power driven.

It was soon found that it was impractical to build mechanical conveying equipment for large units and that it was more practical to build smaller sections and to duplicate these units for the required capacity of the factory. Here again the question of shoe-factory design and construction developed as an important feature, in order to adapt this new idea to the present equipment and process.

The first installations of conveyor equipment were in New England and were operated for several years. There have been several installations through the Middle West which proved satisfactory for the type of shoe for which they were designed, and it is believed that several of these conveyorized shoe factories are still in operation. Considerable publicity has been given to a conveyorized factory in Europe, which has been operating steadily for several years at large capacity.

These installations have shown the trend of pioneer thought to adapt mass production to the shoe industry. Factory problems

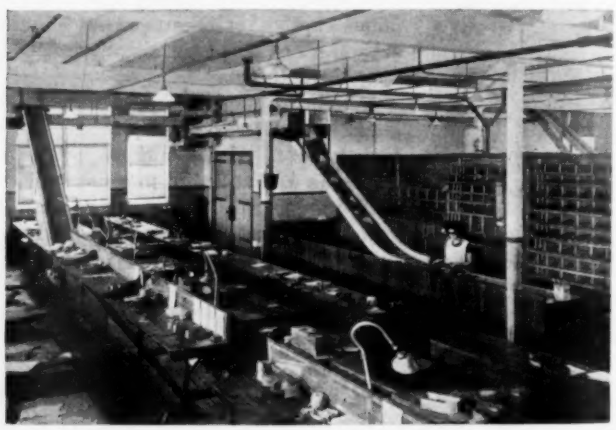


FIG. 3 BELT CONVEYOR FOR RETURNING PATTERNS TO A CENTRAL STOCK ROOM

of style and capacity have retarded the adoption of this idea by the majority of shoe manufacturers, and although practical applications have been made and are now in operation, there are many problems to be solved in view of the satisfactory results recently realized with the 12-pair rack system.

Only assumptions can be made for the future, but no doubt some additional adaptations of this idea will be made where specific shoes of standard design are anticipated.

The foregoing applies entirely to the leather shoe industry. Conveyors for the manufacture of rubber boots, canvas shoes, and allied products are common practice because of the standardized nature of the product. Belt and roller conveyors are indispensable with present-day mass-production methods in the large rubber-shoe organizations.

The illustrations herewith of shoe-manufacturing conveyor machinery show that there are two rows of rollers, one section of rollers being used for a through-way transmission path and the other for intermediate positions for the various operations. This system required the development of stops at different operations so that the shoes could be moved from the main path of travel to the operator at the machine. The operators were timed, depending upon the capacity of the equipment, and the shoes were kept in constant travel through the room.

Whenever it was necessary to dry shoes they were elevated to

especially designed driers. It is impossible to dry leather in a so-called dry atmosphere, because leather must be dried from the inside out rather than to seal the surface with a high temperature. If leather can be dried in a slightly humid atmosphere, this will eliminate the cracking of the grain.

For this reason it was necessary to install humidity-control apparatus, placing the shoes in a moist atmosphere at a moderate temperature, and subsequently carrying the shoe into a relatively higher temperature of 100 to 110 deg. for a definite period. The shoe was then cooled to room temperature before allowing it to proceed to the next operation. This system is still being employed in many factories.

The introduction of drying and mulling equipment, however,



FIG. 4 FACTORY EQUIPPED WITH SHOE CONVEYORS

has been made in a great many factories where the rack method is still employed for routing material throughout the factory. Conveying equipment is employed as a part of the drying apparatus, as will be shown later.

Belt conveyors have been employed for carrying small parts to and from their locations in the room. In one large cutting room all patterns are brought from the individual cutters to a central stockroom for patterns. Here they can be sorted and reassembled for other jobs in the room. This idea can also be employed for transporting dies to and from the stockroom. Overhead wire-basket conveyors on trolleys are sometimes used for distributing parts in the cutting and stitching rooms from a central point such as an upper-leather storeroom, a findings department, or a pattern department. Each application must be studied so as to adapt standard equipment to the individual problem, and so that the relative speed traveled and the design of the installation with relation to other apparatus will fit into the scheme of production without interfering with the flow of material through the factory.

The subject of drying has been mentioned several times, and as the drying equipment of larger size could not conveniently be installed without the application of conveyors, and as the conveying apparatus could not be installed without the application of driers, it seems fit to give attention to this apparatus. In some of the illustrations there is shown a chain conveyor used to transport the machines and the racks through the tem-



FIG. 5 INTERIOR OF A RETURN-TYPE DRIER SHOWING CONVEYOR AND SHOE RACKS



FIG. 6 OVERHEAD CARRIER TO TRANSPORT SMALL ARTICLES

pering room. This conveyor can be either of the drag-link or of the chain type. The reason for this equipment is to speed production in drying and not be dependent on natural atmospheric conditions. The results have been gratifying, and the time schedule has in some cases been reduced two to three days. The shoes are more uniform in texture, defective shoes are reduced in number, and there is a saving in goods in process because of a quicker turnover of lasts and racks. The decrease in production time per unit meets the customer demand for quicker delivery and reduces the amount of money tied up in merchandise in manufacture. It also results in a saving in floor area. The conveyor, of course, must be designed for each

specific job in order to meet capacity requirements for present output as well as for future requirements.

All conveyors should be installed so that they can be loaded and unloaded with the least amount of labor, and the maintenance cost must be reduced to a minimum. The speed of operation will depend upon the amount of moisture to be removed from the shoe and the varying temperature and humidity conditions which exist at different times of the year in the factory. The usual speed will run approximately 1 ft. per min., and the temperatures will range from 70 deg., with a slight amount of humidity, to 100 to 110 deg.

There are other departments of the larger organizations where conveyors of all types are employed. Such equipment as spiral chutes, elevators, lifts, and portable conveyors are used in the packing and shipping room for distributing materials to and from different floor levels and to freight cars. These installations of course are more or less standard in factories of today, each equipment being adapted from standard parts to meet specific building problems. Those concerns which are large enough to make boxes and cartons use conveyors of smaller types to transport materials.

The larger shoe companies maintain power plants of considerable size which employ coal conveyors of familiar types. Ash-handling apparatus is also employed.

The dust-collecting problem is a very important one in shoe-factory operation, and it is being given much thought by plant engineers in charge of equipment.

The author has designed several large dust-collecting units for shoe-factory installation and has found that a detailed design must be made for all piping, layout for fan equipment, and a design of the hoods which are attached to the individual machines. Sizes of suction pipes to each machine must be specified and the equipment so designed that a suction of at least 2 to 3 in. of water is obtained at the hood. Branch machine connections will vary in size from 2 to 6 in. State regulations will also affect such a design.

The location and the installation of fans and dust-collecting equipment are of paramount importance. Engineers who anticipate power-factor conditions frequently find it possible to employ synchronous motors to drive these fans, as they are in operation throughout the working day and furnish an ideal opportunity because of the steady power load. The installation of slow-speed, low-powered fans is advisable on account of the power saving, and where practical the size of the fan should be reduced to the minimum, even to the extent of separating the system into small units. It frequently occurs that various rooms operate at different intervals, when it might be advisable to shut down the fans for one of the other departments.

Conveyors are also used in some of the larger offices in shoe-factory organizations, even employing pneumatic tubes, basket conveyors, and lifts to assist in distributing office material about the building. The transportation department frequently employs belts, lifts, spiral chutes, and other ideas for expediting shoe cases from the shipping room and stock department.

Another item which affects the material-handling problem is the change in the distribution of power. The first shoe factories, as with all industries in this country, started to develop power from the engine drive and belt transmission. Large belts transmitted power through shafting to all corners of the factory, and it was necessary to place the machines so that they would be operated from the line shafting. In many of these old-type shoe factories the friction load has sometimes been as high as 70 per cent of the total load. With the adoption of the induction motor and the adaptation of the group drive to shoe-factory planning, it was found that the shafting problem could be partially eliminated and the machinery be grouped in smaller units to obtain flexibility. It was possible to drive stitchers, welters, trimmers, or other groups of one classification with a separate motor, and that unit could be independent of all others.

There are many benefits from individual motors in the shoe trade which are peculiar and perhaps would not occur in other industries. First, the small size of motor required lends flexibility to the machine. The changes in style in shoe design and manufacture require the frequent moving of this equipment from one location to another. Machines can be placed at the proper angle with the windows to obtain perfect light.

There are specific conditions where it may be possible to introduce mechanical means to transmit shoe products through the factory when applied to large units of standardized grade. The rubber-shoe factories have already adopted the conveyor as standard practice. To thoroughly develop this method an extensive study must be made of all problems of manufacture in each individual organization. A whole-hearted cooperation must be used in the operation of driers and tempering rooms. Engineering skill must be employed to cooperate with all phases of management to overcome the customs which are so strong in this industry. The economic trends are forcing themselves upon this industry and the necessity of reducing production schedules and operating costs will form an incentive to enterprising manufacturers to work out new methods to create the most practical work unit, whether it be transported about the factory on a conveyor or by the simple shoe rack.

MH-53-3

Materials-Handling Methods in the Fisk Tire Plant

By C. E. MAYNARD,[1] CHICOPEE FALLS, MASS.

The paper discusses the part that handling equipment has played in a program of cost reduction and quality improvement in the Fisk Tire Company Plant. In recent years there has been a tendency for the plant to become more "conveyorized," coordinating groups of operations. The Trucking and Mechanical Departments are mainly responsible for the materials-handling equipment. The Engineering Department, a subdivision of the Mechanical Department, is expected to develop improvements. The cost of improvements has to be justified by a reduction in operating cost or an improvement in quality. There are no limitations on the cost of materials-handling equipment providing a saving can be made within a reasonable length of time. Comparing trucking costs of 1925 with the present, a 50 per cent reduction is noted. In spite of the tremendous savings that were made in materials-handling methods during the last four or five years, the possibilities of further improvement are far from being exhausted.



LARGE reductions in manufacturing-labor costs have been made at the Fisk Rubber Company in the last few years. Part of this reduction has been due to improved handling methods. This paper will attempt to show what improvements in materials-handling methods have contributed to a lower operating cost and an improved quality of the product. During the last few years materials-handling methods have received much more attention in the tire industry than formerly. Tire plants in the past were more "departmentalized." Production men and engineers looked upon the tire-manufacturing process as a group of departments, each one more or less separate, but contributing to the finished tire. The straight-line idea of production began to appear in the tire industry but a few years ago, as it did in other manufacturing endeavors. With the straight-line idea in mind, the production engineers, unconsciously perhaps, began to look upon tire manufacturing as a single continuous process rather than as a group of processes. The first effect of this change in point of view was to tie departments more closely together by improved materials-handling methods. Development along these lines has been going on continuously for the last few years, and the savings have been extensive. In many cases, improved handling methods have obliterated some departmental lines completely, whole groups of operations being so coordinated that it was no longer logical to maintain certain departments as separate units.

The Fisk Rubber Company has been a pioneer in these developments. The production buildings of the Fisk plant are modern; consequently it has been possible, in spite of many improvements in the methods and layouts, to work out a satisfactory handling system.

Control of Material-Handling Equipment

At the present time two departments are mainly responsible for the care and maintenance of the materials-handling system—the Trucking Department and the Mechanical Department.

The Trucking Department has been assigned the general responsibility of operating all elevators, all interdepartmental trucking, where the use of electric trucks or tractors is involved, the laying out of all trucking aisles, the determination of the amount of trucks and trailers to be used in the various departments, and, finally, a primary responsibility for the maintenance cost of the aforementioned equipment. The supervisor of the Trucking Department reports directly to the factory superintendent.

The Trucking Department supervisor has two types of workers under him—elevator operators and electric-truck drivers. It is his responsibility to see that the elevators are operated with an optimum load, and also it is his responsibility to determine the fixed routes and the size of loads that the truck drivers shall handle. As production varies it is customary for the supervisor to send into equipment storage the excess trucking equipment, such as trailers, live and dead platforms, and special departmental trucks.

Other materials-handling systems, which are mainly conveyors of various types, are under the supervision of the mechanical superintendent, who in turn reports to the factory superintendent. He is responsible for the upkeep of the conveyors and is held accountable for any losses in production which occur because of their breakdown.

The Engineering Department, which comes under the Mechanical Department superintendent, is expected to develop improvements in the materials-handling system, whether it be conveyors or special trucking equipment. They, of course, do this with the cooperation and assistance of the Production Department supervisors. In many instances the ideas involving improvement of the materials-handling system actually come from the Production Department rather than from the Engineering or Trucking Departments.

The cost of every improvement in the materials-handling methods has to be justified by a reduction in operating cost or an improvement in the quality of the product. In some instances it is also a matter of saving valuable floor space. Incidentally, there never has been any limitation placed on the cost of materials-handling equipment to be used, providing the saving to be made could be paid for within a reasonable length of time.

[1] Factory Superintendent, Fisk Rubber Company. Mr. Maynard was born on January 13, 1895. He received the B.S. degree from Amherst College in 1917. His business history includes The Good-Year Tire & Rubber Company, 1917–1918; Naval Aviation Corps, 1918; went with the Fisk Rubber Company, 1919; Manager of Process Development Department, 1926; Mechanical Superintendent and then Factory Superintendent, 1930. Mr. Maynard is the author of several papers and the inventor of numerous machines and processes relating to the rubber industry.

Contributed by the Materials Handling Division for presentation at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers, 29 West 39th St., New York, N. Y. All papers are subject to revision.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

General Layout of the Main Production Building

It might be advisable to turn for a moment to the general layout in the main production building of the plant. Most of the crude materials come into the building on the second floor, this being the track level. From there they drop one floor to the Mixing and Milling Department. Adjacent to the Milling Department is located the Calender Department. It has been found advisable to place these two departments on the first floor on account of the weight of the machinery used. Rubberized fabric from the Calender Department is carried to the top, or fifth, floor of the building, where some of it is cut to the various dimensions required for tire assembly. The storage for this material is maintained there also. Some of this fabric is brought to the

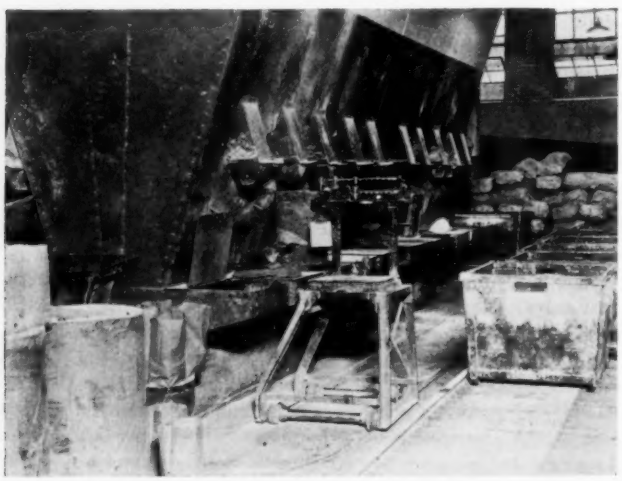


Fig. 1 Hopper System in the Compound Department

fourth floor, where cutting machines and band-building units are located. The cut materials from the fourth and fifth floors are then conveyed to the third floor, where the Tire Building or Assembly Department is located. From the third floor the raw tires are conveyed to another section of the first floor, where they are cured or vulcanized. On account of the curing equipment being heavy, it is necessary to locate it also on the first floor.

From the Curing Department the tires are conveyed to the Tire Finishing and Inspection Department, located on the fourth floor, which is the most convenient level from which to convey the tires to the storage building.

Course of Production, Compound Department

There can now be followed in detail the tire-manufacturing processes throughout the plant, noting as far as possible the materials-handling methods used.

The first department in the tire-making process is called the Compound Department. The storage place is located on the car level and very near the unloading point. The pigments, which come in bags or barrels, are generally emptied into sifters directly over the hopper system. These hoppers are large enough to hold from one day's to one week's supply of pigments. The outlets of the hoppers, which are on the floor below, are brought within the smallest possible area so as to reduce the walking time of the operator who weighs the materials. A roller conveyor circles the outlets of the hopper system and carries the batch boxes into which are loaded the various weighed pigments and rubber. The boxes of materials are checked for weight while on the conveyor, after which they are hauled to the Milling Department on live platforms. The Milling Department is on the same floor level and adjacent to the outlets of the hoppers.

The rubber for the compound batches is unloaded in two locations, some of it coming from the cars in the basement of the building near the washing lines and the remainder being taken off the cars on the second floor of the building near the rubber cutters and drying room, these being the operations that ordinarily follow the unloading of certain types of rubbers. The rubber taken off at the latter point is used for feeding the Banbury mixers, the loading openings of which are on the second floor. When it is necessary to move the rubber, it is handled in either large box-like trucks or in large batch boxes placed on live platforms.

Electric trucks are important pieces of equipment in the Compound and Milling Departments. They are used for moving the platforms loaded with batch boxes full of rubber and pigments and for hauling large box trucks filled with crude rubber to the various operations, such as the cutting, drying, and washing.

Milling and Calendering Department

As previously mentioned, the Compound Department is located on the same floor level and adjacent to the Milling Department, where the pigments and the rubber are mixed previous to their use on the calenders or extruders. This step in the course of handling the material has not been worked out

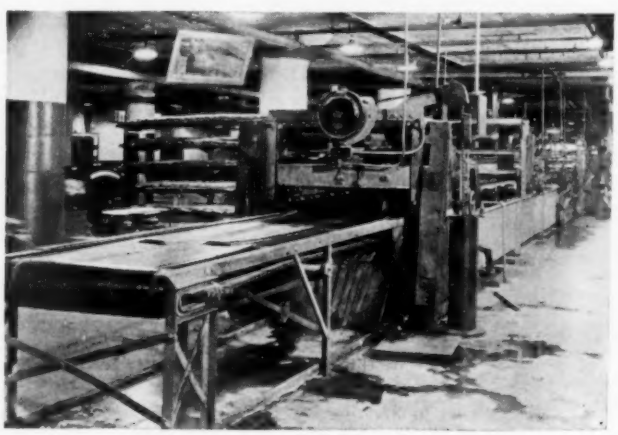


Fig. 2 Conveyor System in the Tread Department

to complete satisfaction. At present the pigments and rubbers in batch boxes are hauled to the mixing mills by means of electric trucks. After the mixing operation has been completed, the compounded rubber batch, which is then in slab form, is placed upon live platforms and hauled to the warming mills adjacent to the calenders or extruders.

Until recently it was common practice to store mixed stock in a storage department before transferring it to the various calender or extruding operations. This break in production tended to prevent delays encountered in the milling operations being transmitted to the calenders, and at the same time the calenders could quickly swing from one stock to another with an assured supply available. It also was claimed that the stock had to cool and age in storage to give standard processing conditions. This scheme, however, resulted in large inventories of stock and in comparatively high handling costs. It was found that this was not necessary; consequently, the stock now goes in most cases directly to the calendering or extruding operations without any delay. This of course reduced the inventory of material and also the handling costs.

In one section of the Milling Department the mixing opera-

tion is tied by means of conveyors directly to the warming mills of the calenders, which reduces the handling cost of the material between the mixing operation and the calendering to a minimum. It may be possible some time to follow this same idea throughout the Milling Department.

Although the warming mills were always near the calenders to which they supply stock, until recent years it has been customary to feed the latter by hand. Now it is common practice on calenders coating tire fabric to use a conveyor to carry a strip of stock from the warming mill to the feed bank of the calender. Because this continuous strip yields a more uniform feed, greater accuracy in weight of the coated fabric can be obtained than with the old hand method. The coated fabric as it emerges from the calender is wound into rolls containing as much as 500 yards. It is wound up in a fabric liner to prevent it from sticking together. The total weight of one of the rolls is about 1800 lb. A live platform with a steel superstructure which will hold two rolls in a horizontal position is used to transport them to the Stock Preparation Department.

Tread Department

In addition to coating fabric the Calender Department has as one of its main duties the production of treads. The tread is,

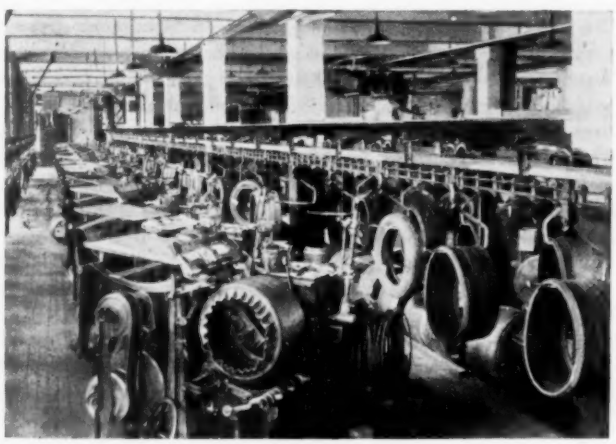


Fig. 3 One Section of the Tire Room
(On the left, the conveyor carrying the cut materials to the tire machines and on the right, the uncured, finished tires being taken away to be cured.)

as the name implies, the tread section of the tire and is formed by a calender in a continuous strip. Until a comparatively few years ago no attempt was made to link the tread-calendering process to any of the operations that directly preceded it, such as the mixing of the stock, or to any of the operations that followed it, such as the weighing of the treads and cutting them to an exact length. In other words, in order to produce the tread, from the mixing of the stock to the finishing of the tread to length and weight, three separate groups of operators or departments were required. Improved materials-handling methods were installed a few years ago which tied these three groups together, so that now the process is continuous and considered as a single departmental function.

The whole first floor in one of the main buildings was laid out so that it was possible to compound the stock, mix it, feed it directly to the calenders, cool the treads, weigh them, and cut them to a finished length. To do this four Banbury mixers were installed in such a way that the batches of pigment and rubber could be assembled on a conveyor and then fed directly to the mixers without any intermediate trucking. This conveyor is a combination of two types—pallet and roller gravity. It forms a rectangle about the Banbury mixers. The pigments and rubber are weighed out in batches and placed in boxes on one side of this rectangular conveyor system. This side of the system is at floor level and is a gravity conveyor. Boxes are carried by a pallet conveyor up an incline to a roller gravity section which runs parallel to the loading section at floor level and along the feed openings of the Banbury mixers. By using a roller gravity section here a small bank of batches can be kept ahead of the operator who unloads the boxes into the mixers. This section being on a higher level on account of the height of the mixer's feed openings makes it possible for the operator to easily return the empty boxes to the loading conveyor by



Fig. 4 Conveyor System in the Curing Department

merely shoving them down a sloping gravity conveyor section. In other words, the boxes make a complete circle of the Banbury mixers on the conveyor, being loaded on one side and emptied on the other without necessitating their removal from the conveyor system.

After the stock is mixed, it is conveyed directly from the outlets of the Banbury mixers to a series of blending and feed mills by means of belt conveyors, and from there carried to the feed of the calenders.

The calenders form the treads, which are then conveyed in continuous strips through the cooling tanks to shrink the stock. The strips, after leaving the cooling tanks, pass over continuous weighing scales in order to be certain that they are within the proper weight tolerances, and are then conveyed through automatic cutters, which cut them on a bevel to the proper length. The treads are then taken from the belt conveyor and piled on what is called a book truck. This is a live platform about 10 ft. by $3^1/_2$ ft., on one side of which is a steel frame $4^1/_2$ ft. high. This frame supports rods upon which cloth leaves slide, as they do in a loose-leaf binder. This truck is provided with 100 leaves, which give it a capacity for 200 or more treads. The leaves are used to keep the treads from sticking together. This truck thus serves as a means for storage as well as transportation. These trucks are hauled to the storage place adjacent to the feed conveyor of the tire-building units.

Due to the fact that the rate of productions of the tread calenders and the tire machines respectively are not the same, it is necessary to maintain a few hours' bank of material. Here a mobile bank, such as these trucks afford, proves very satisfactory.

This improved method of handling treads, together with the

mechanical improvements, practically wiped out a Tread Cutting and Weighing Department. At the same time it reduced the inventory of stock to a fraction of what was previously maintained and reduced the unit cost of calendering the tread about 38 per cent.

The foregoing description gives one who is not familiar with the tire plant very little idea of what has actually been accomplished in the improvement of handling conditions in the Milling and Calendering Departments. When the plant was handling as high as six or seven million pounds of mixed stock per month, it was not uncommon to have more than 800,000 lb. of mixed stock either in the Stock Storage Department or in transit. Under present conditions, even though the production tonnage is at times higher than in the past, about 300,000 lb. is considered normal. With compounded stock costing on an average over a



FIG. 5 CONVEYOR SYSTEM IN THE TIRE FINISHING AND INSPECTION DEPARTMENT

period of years around 30 cents per pound, it is quite evident that the interest charges on this first-mentioned inventory would amount to a considerable figure. The handling costs were tremendous in comparison with the present conditions.

STOCK-PREPARATION DEPARTMENT

As previously mentioned, the coated fabric after it leaves the calender is transported in rolls to the fifth floor, where some of it is cut and some stored, to be cut later on the fourth floor. This department is generally termed the Stock-Preparation Department, and is the department where the stock is cut to various lengths and widths for final assembly in the tire. It contains cutting machines of various types and also equipment for making bands containing two plies of tire fabric, one ply being laid diagonal to the other. A considerable amount of handling of rolls is required in this department to and from the cutting machines and also to and from the band-building machines. In order to facilitate this handling there has been developed an electric truck[1] with a large scoop which is capable of picking up and unloading these heavy rolls very easily. The scoop has an elevating attachment so that it makes it possible to deposit rolls in the supports of the machines regardless of their height or, if necessary, to place the rolls upon the floor or pick them up either in a vertical or a horizontal position.

[1] A product of Elwell-Parker Company, Cleveland, Ohio.

TIRE-BUILDING DEPARTMENT

Most of the material from the Stock-Preparation Department is carried in spools on conveyors directly to the tire builders. There are four conveyors, each about 300 ft. long, running between the Stock-Preparation Department and the Tire-Assembly Department. These conveyors run parallel to the tire machines on the floor below. They are provided with hooks that carry the treads and also the spools containing the tire bands, chafing strips, and padding and breaker combinations. There are a sufficient number of these hooks on the conveyors to provide for a bank of material so that delays will not arise from an insufficient supply or from the normal irregularities in production which exist between the two departments.

The installation of a conveyor at this stage of the process—namely, between the Stock-Preparation Department and the Tire-Assembly Department—not only reduced the materials-handling costs but also tended to insure a fresh supply of stock for the tire builders. This is a condition of considerable importance in a tire plant. Uncured rubber tends to bloom and lose its adhesive characteristic if allowed to age too long before vulcanization. Dry stock, as it is called in the industry, is liable to result either in higher building costs or in a poor product.

In the tire room there is practically no trucking of materials. As previously mentioned, the materials for assembly of the tire are conveyed to the tire builders on conveyors running parallel to the various lines of machines. Along the rows of these machines, running parallel to the feed conveyors, is located another chain conveyor for carrying the uncured finished tires away from the builders directly to the Curing Department, which is located on the first floor. This conveyor is 2800 ft. in length. It is a standard chain type, capable of moving in two planes, thus allowing it to pass by the various tire-building units and also by the inspection point before it drops to the heater room, where the tires are unloaded at two different points.

In years past it was common practice to transport the various materials from the Stock-Preparation Department to the Tire Department on dead platforms and also to transport by means of platforms and electric trucks a big portion of the tire production from the building machines to the Curing Department. The installation of these conveyors eliminated this trucking cost.

CURING DEPARTMENT

The Curing Department, as just indicated, is the next step in the tire process. The tires, after they are taken from the conveyor, are dusted and shaped, providing they are not already built to shape, and airbags are inserted. An airbag is like a heavy inner tube, in which pressure is applied during the vulcanization of the tire, the tire being forced against the mold by this pressure, thus forming the tread design and the lettering which normally appear on the finished tire. When the airbag is inserted, the tire is ready for the mold.

In order to handle the molds, which weigh on an average 600 to 700 lb., to and from the vulcanizers where the tires are cured, a rather elaborate conveyor system has been installed. It might be mentioned here that a vulcanizer or heater is a deep cylindrical tank on the bottom of which is a ram used for loading purposes and also for pressing the molds together during the vulcanization period. When these heaters are loaded with molds and the cover is on, steam pressure is applied, which, together with the internal air pressure on the airbags, brings about the vulcanization of the tires. These heaters are arranged in rows containing from 18 to 24 units, parallel to which runs one side of a rectangular conveyor system. There are four complete systems in this department. Each conveyor system is really a set of pallet conveyors of heavy steel construction.

As just mentioned, the heaters lie along one side of the rectangular conveyor system, and on the side opposite are located the various workmen who open the molds, strip the cured tires from them, and reload them with raw tires. The upper halves of the molds, for part of the distance along the loading section of the conveyor, are carried overhead after having been broken free. They are finally brought down into their proper position over the lower halves after the tires have been inserted. The loaded molds are then carried by the pallet conveyor to whichever heater is being filled. On this pallet conveyor are located presses for closing the molds tightly previous to their being deposited in the heaters. There are also spraying devices which spray a coat of lubricant on the inner surface of the mold in order to make it possible to easily remove the tire at the end of the curing process. The conveyor thus serves not only to convey the tire to and from the heaters, but it also becomes an integral part of the process of curing the tire. After the tires have been cured they are carried on a chute to the bag-pulling machines, and from there are conveyed to the Inspection Department on vertical lift chain conveyors.

This description is necessarily brief, but it suffices to show that the problem of materials-handling in curing a tire is of tremendous importance. The installation of these conveyors reduced the cost of curing a tire about 30 per cent, and at the same time considerable saving in space was made.

This conveyor installation is so important to the process that any breakdown in equipment disrupts production in this department completely. In order to avoid this possibility a weekly inspection of every part of the conveyor is made, and periodically, about on an average of every four or five years, the chain is completely replaced. Systematic check and replacement of worn parts have made breakdowns extremely infrequent in spite of the fact that these conveyors have been in operation for over 12 years.

The conveyors are not the only material-handling units in the Curing Department. When the raw tires are taken off the conveyor from the tire room, they have to be sorted according to size for the various groups of molds that are in the heaters. This necessitates a bank. It has been found through experience that it is advisable to maintain here a movable bank. In order to do this, a truck was developed upon which about 80 tires can be stored. At the same time it can be readily pushed from the raw-tire conveyor to the storage point and to and from the tire-shaping machines. This truck might be termed a "tree" truck in that it has a steel structure supporting a number of arms upon which the tires are hung.

Inspection and Tire-Finishing Department

The Inspection and Tire-Finishing Department follows the curing of the tire. The tire is carried from the heater room on vertical lift conveyors. Although there are several outlets to the main curing units, they all converge at one place in the Inspection and Tire-Finishing Department, which is located on the fourth floor of the building, and which is the floor most advantageous for conveying the finished tires to the storage building.

Until a few years ago the Inspection Department contained mainly dead platforms loaded with tires and tables upon which to trim and inspect them. There were no conveyors used except one to carry the tires to the storage building. It is necessary in this department to trim from the tire the mold overflow, balance it, inspect it, paint the tire completely, and also paint the trade mark upon the tire. In order to carry the tire through the various operations and not encounter delays, a series of belt conveyors and live-roller conveyors were set up parallel to each other. Between these conveyors are placed various groups of workers required to perform the aforementioned operations. Two of these conveyors are double-decked in order to save space, the lower conveyor being used for incoming tires and the upper conveyor being used for conveying away tires upon which the operations have been completed. The live-roller conveyors, which are always the feed conveyors to the operators, allow the tires to bank up, so that at all times the workers have tires available. At the completion of the last operation, which happens to be the trade-mark painting job, the tires are then carried on a chain conveyor to the operators who sort the tires for the wrappers. The sorters place these upon small steel round-top platforms. About 16 tires can be piled on one of these platforms. Due to the fact that they have four casters, they can be readily pushed about. The actual dimensions of one of these platforms are $6^5/_8$ in. high and 40 in. diameter at top. These platforms can be stacked in a very small space when not in use. This is another case where the material storage is also movable, a condition which is generally found to be desirable in a tire plant. The tires are taken from these platforms and wrapped, and from there are carried on a belt conveyor to the storage building.

Aside from a considerable saving in space which was made by the installation of this material-handling system, about 10 per cent labor saving was made in this department at the time by the installation of this system. The cycle of operation is reduced considerably, so that now a tire comes out of the mold and is wrapped within an hour.

Inner-Tube Department

The preceding description was confined to the manufacture of tires. The Fisk plant also manufactures a large number of inner tubes. Here again it has been found possible to make a saving by improvement in the materials-handling methods through the installation of conveyors and the development of movable storage trucks.

The standard method of making raw tubes in the Fisk plant is by means of extruding or tubing machines. As in the case of the tread job, formerly it was the practice to extrude tubes on to a short conveyor and transport them to a unit where they were weighed and cut to length and where the valve pads were applied. They were then transported to a splicing unit, where the ends were united, and from there were trucked to molding machines.

A new layout was evolved which tied the warming mills by means of a belt conveyor direct to a tubing machine, feeding the latter a continuous strip of rubber. A conveyor system was then built from the extruding end of the tubing machine which carried the tube over an automatic weighing scale and through a cooling tank, where the tube was shrunk, and then through a drier, and then to a device for marking the tube to length, and, finally, through a machine for pressing the valve pad into position. The tube is then cut to length and the valve hole punched. In other words, by the development and installation of special mechanism on this conveyor system a series of operations were tied together, eliminating the trucking and considerable manual handling that previously existed. The inventory of the tubes was reduced, and a considerable labor saving amounting to about 15 per cent on the afore mentioned operations was made.

In the past the means of storing tubes after leaving the tubing-machine line were unsatisfactory in that the ordinary types of trucks lacked adequate capacity. Recently, however, a truck[1] has been developed which is somewhat like the tread truck, but it has metal leaves or shelves instead of fabric leaves. As many as 200 tubes can be stored on such a truck, and on ac-

[1] A product of Economic Steel Rack Company, Everett, Mass.

count of the fact that each shelf is self-supporting and does not rest on the tubes underneath it, there is no danger that the tubes will be deformed or their walls pressed together. These trucks are also movable and can be easily hauled from the tubing machine lines to the splicing-machine units.

Handling the tubes between the raw splicing units and the tube molds has not been worked out to complete satisfaction. At present, however, tubes are transported between these two units on light steel trucks which look somewhat like book racks, the spliced tubes being placed upon the shelves.

After the tubes are cured they are placed by the operator directly on a chain conveyor which runs to the floor above, where the tubes are tested for leaks and finally inspected. This conveyor also passes by a group of operators who apply the washer and nut to the valve stem and tighten it down. On account of the fact that it is necessary to make this conveyor pass by three lines of tube molds and also the line of valving operations, just mentioned, it is a conveyor of considerable length, amounting to about 900 ft.

The finished tubes are taken from this conveyor and are carried through a water tank to be tested for leaks. From there they are put on a chain conveyor which carries them through a drying oven, and then by a group of operators who deflate the tubes and inspect them. The tubes that pass this inspection are placed upon a belt conveyor which carries them to the boxing operators. A break in the system is made here in order to sort the tubes according to style and size for the boxers. The boxing is done alongside a belt conveyor where can be located trucks of tubes and boxes. This conveyor carries the boxed tubes directly to a point where they can be placed in carbons and sealed. In other words, all trucking between the tube curers and the assembly of the tubes in cartons is eliminated except for a brief sorting period at the end of the tube-inspection conveyor. The saving made by the installation of these conveyors has amounted to a fairly large portion of the manufacturing cost of the tube. Previous to their installation several operators were required to transport the tubes on trucks between the various operations. By the use of conveyors the efficiency of the various producing units was improved. This was especially noticeable in the case of the tube inspectors. Approximately 14 per cent reduction in labor cost was effected on this operation. In addition to these savings there was a considerable reduction in punctured tubes, which ran high under the old method, where a great deal of handling was required. At the same time the inventory of tubes in process was reduced.

The preceding description covers briefly the main course of production of tires and tubes in the Fisk plant. There are other articles manufactured, such as flaps, sectional airbags, and rubber articles associated with tires. Improvements in their handling methods will not be discussed in this paper.

General Observations

One might judge from the preceding descriptions of handling methods that practically all the improvement in this direction was made by the installation of various types of conveyors. It is true that conveyors were responsible for most of the improvement in the handling methods, but there were many instances in the last few years where improved types of trucking equipment have accounted for a reduction in cost.

In spite of the fact that every effort is made to carry the product through the plant without interruption, there are many occasions where it is necessary to remove it from the regular producing lines either for some operation or for inspection. Then again there are places in the tire processes where it is necessary to maintain banks of materials in order to eliminate delays between operations. In such cases it has been necessary to develop trucks to store these materials and at the same time to transport them. As was mentioned in the case of the Tube Department and the Tread Department, there are many other instances where a mobile storage truck was invariably found to be better than stationary storage equipment.

Records show that there are 115 various types of trucks used in the plant. A great many of these trucks are built upon a standard type of platform. Standardization, in so far as parts and casters are concerned, is carried as far as possible. Maintenance alone on these trucks is an item to which it has been found necessary to give considerable attention. The same maintenance men are always assigned to work on repairing them; consequently, their increasing skill and knowledge of the equipment results in decreasingly lower costs.

The manager of the Trucking Department makes a special point of constantly calling the attention of the production supervisors to the damage that is being done to trucking equipment by their operators. Obvious abuse is traced down, if possible, to its source. Efforts along these lines have resulted in a reduction of 33 per cent in trucking-maintenance cost since 1925.

It was rather difficult to obtain figures in regard to the amount of money that actually was saved by improvements in materials-handling methods in the last few years. In many instances improved materials-handling methods not only made a reduction in trucking costs, but also reduced the direct labor cost of building the tire and improved its quality.

In some instances certain conveyors acted as pacemakers for the operators who were fed by them. A constant regular flow of material apparently tended to iron out the previous tendency to produce at an irregular rate of speed. Higher production figures resulted. Then again it was found possible, as in the case of the Tread Department and the Tube-Extrusion Department, to make the conveyor an integral part of the process, thus contributing to a lower labor cost over and above that which would normally be made by merely eliminating the trucking.

There are some figures, however, that can be used for comparison, such as the trucking cost for the years 1924–25 with the present cost. It was found that the present trucking cost per 1000 lb. of mill-room production is one-half of what it was during the years of 1924–25. This is a fact of considerable importance because during the period of 1924–25 the trucking cost, including the labor cost of operating elevators, amounted to approximately $200,000 per year. It is probably safe to estimate that this saving of approximately $100,000 per year would actually be doubled if all the other items of saving that were brought about by improved handling methods could be segregated.

In spite of the tremendous savings that were made in the Fisk plant in materials-handling methods during the last four or five years, the possibilities of further improvement are far from being exhausted. Tremendous strides have been made, but it is safe to assume that in the next five years improvements quite comparable to the last five years will be accomplished.

MH-53-4

Handling Materials at the Postum Company

By P. P. PRATT,[1] BATTLE CREEK, MICH.

This paper offers a discussion of various types of conveyors in general use, together with descriptions of a few specialized conveyors. Practically every phase of manufacturing at the Postum plant embraces operations performed on the material in course of handling—especially is this true of modern automatic packaging machinery, referred to in this article. An enlarged section treats with the operation and mechanical details of the "intercommunicating pilot-car package-conveying system," used as a connecting link between shipping points. This system automatically handles a wide variety of cased products and offers selective loading at each of four shipping points. It has effected quite considerable economies and has made it possible to utilize widely separated storage spaces, more or less as a unit.



THE production of food was man's earliest occupation, but the handling of materials in quantity was not a very serious problem. Today it is a problem of major importance in practically every industry. The food industry is no exception.

To meet the demands of the consuming public, a food must be clean and made under conditions of scrupulous cleanliness—even more so than exist in the average home. It must have an appeal, both to the taste and to the eye; it must be uniform in quality and packaged in an attractive container, with every safeguard taken to prevent deterioration either by absorption of moisture or contamination with insect pests; and, last of all, it must be sold at a popular price.

The development and growth of the cereal-food industry would not have been possible without automatic packaging machinery of various kinds, which made possible the rapid handling and packaging of the manufactured product, with a minimum of labor and handling of the food itself.

Similarly, the development of special machinery, made necessary by the growth of the business, has insured the rapid handling of materials and manufactured product under increasingly clean and uniform conditions, resulting in the high-quality product that exists today.

Proper handling of materials governs three very necessary factors of food manufacturing:

(*a*) Quality, which is accomplished by mechanical handling that regulates the flow of materials to produce the uniformity necessary for rigid quality standards.

(*b*) Cleanliness, which means the absence of human handling, that is replaced by mechanical handling, with special alloy equipment wherever food in process is contacted.

(*c*) Continuity of production, which insures that the manufacturer can produce a maximum quantity of the product with a minimum of material in process.

Before considering the details of equipment for materials handling, the five products made at the Battle Creek plant of the Postum Company, Inc., should be enumerated. They are: (1) Postum Cereal and Instant Postum; (2) Grape-Nuts; (3) Post Toasties; (4) Post's Bran Flakes; (5) Post's Whole Bran.

There are seven distinct classes of materials to be handled in the processes. These are:

(1) Grains—corn, wheat, and malted barley in bulk.

(2) Sacked materials—wheat, bran and dairy feed ingredients such as oil meal, cottonseed meal, and gluten meal.

(3) Flavoring materials—sugar and salt in bags, and syrups in barrels and tank cars.



FIG. 1 ORIGINAL PRODUCTION UNIT, THE LITTLE WHITE BARN, WITH NO CONVEYING PROBLEMS

(4) Products in process of manufacture and packaging.

(5) Products packed in fiber and wood cases for domestic and foreign shipment.

(6) Paper and boxboard materials for the manufacture of cartons and cases, together with handling to packing departments.

(7) Paper and wax stocks for manufacturing the protective wrappings characteristic of Post products packages.

MATERIALS HANDLING

The handling of grain is the foremost consideration in maintaining a proper standard of quality. To the uninitiated, corn is either white or yellow, and all wheat is either red or white. It is a fact, however, that there are several varieties of corn, together with some 60 varieties of wheat—only certain varieties and grades of these are suitable for the manufacture of Post products. By means of a number of storage tanks for grain, it is possible to segregate the various grains and blend these properly. Postum's grain-handling and storage equipment,

---

[1] Engineer, The Postum Co., Inc. Mr. Pratt is a graduate of Massachusetts Institute of Technology, 1923. Basic experience was gained in a two-year factory apprenticeship working through the departments of the Postum Company plant. For the last five years he has had charge of experimental work, especially with reference to changes and improvements in manufacturing processes and equipment.

Contributed by the Materials Handling Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.



FIG. 2 PRESENT PLANT OF THE POSTUM COMPANY, WITH MILES OF CONVEYING SYSTEM

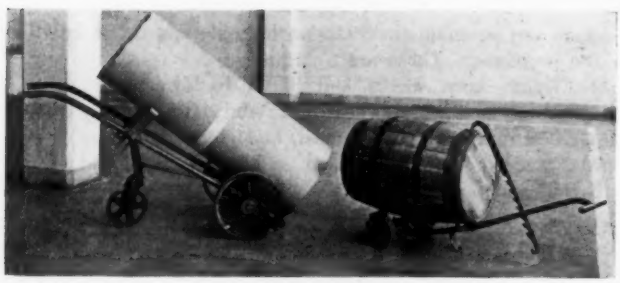


FIG. 3 TYPICAL BARREL AND ROLL-PAPER TRUCKS

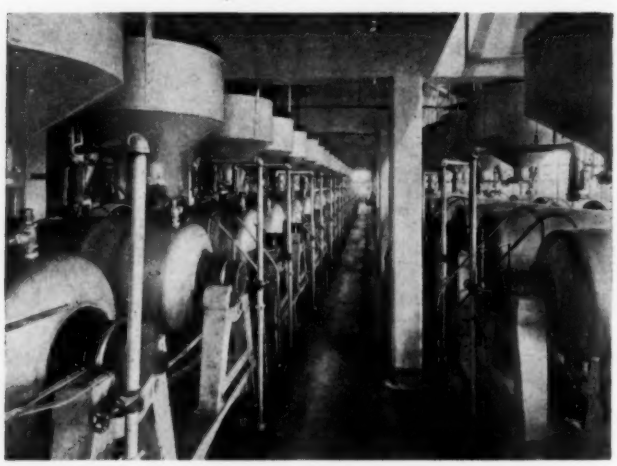


FIG. 4 ROTARY STEAM COOKERS FOR POST TOASTIES
(Hominy is loaded to the cookers by gravity, the cooked product being removed by belt conveyors.)

about a million-bushel capacity, is sufficient to provide raw material for 30 days' production. Sometimes grains contain relatively high moisture and develop "hot spots." The tanks are equipped with electrical temperature recorders that indicate the temperature every 5 ft. for the full 90 or 100 ft. depth of the tanks. To save heated grain, it is necessary to draw out the hot portion of the grain, aerate, and rehandle to a cool tank. Equipment is available to perform this operation with elevators and screw or belt conveyor at a rate of 6000 bu. per hour, in addition to unloading many cars of grain each day.

Sacked materials such as wheat bran present a very simple handling problem. For this raw material Postum depends upon controlling the supply to maintain a small but adequate storage and insure that it is fresh and sweet from the milling processes. Sacked material is stocked either on portable skids or in low piles and arranged to be used in stock rotation, i.e., oldest stock first. Since blending is performed at the flour mill, there is little necessity for blending at the plant. Dairy-feed materials are purchased in 100-lb. bags, combined with by-products of the food processes to govern the proteins, fats, and carbohydrates, and serve as an outlet for waste foods.

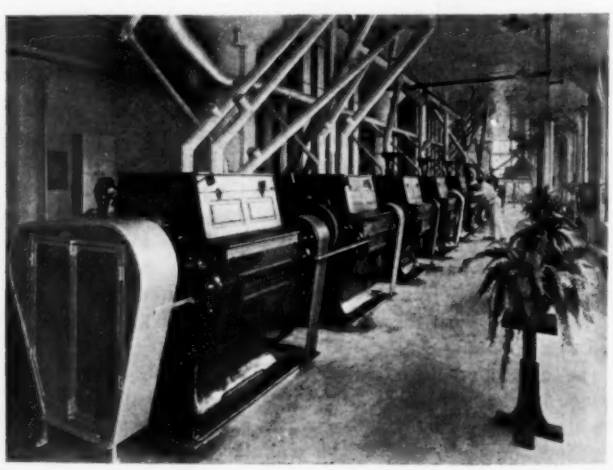


FIG. 5 MILLING WHEAT AND MALTED BARLEY FOR GRAPE-NUTS
(The material flows by gravity to and from the mill.)

Flavoring materials are a very important adjunct to the cereal-food industry. Sugar and salt are bagged products, of course; these are handled from the car by lift trucks and stored in the buildings on skids. Malt syrup is received in barrels and stored in cool areas, thence handled into process by barrel trucks. (See Fig. 3.) Sugar-cane molasses is received in tank cars, piped to specially constructed concrete tanks, thence pumped to process mixers for flavoring wheat bran before roasting for Postum Cereal.

Practically every step of the several processes involves the handling of materials. The essential operations of cooking (see Fig. 4), cooling, drying, flaking, toasting, and packing are performed on the materials in transit; in fact, only a few hours elapse between the time when corn hominy or bran enters the process and when it is passing into freight cars, as flakes. In milling corn to produce flaking hominy, it is continually moving through degerminators, reels, driers, coolers, rolls, aspirators, oil expellers, attrition mills, and sifters until all grades of flaking hominy, table hominy, cones, cornmeal, flour, corn oil, and

hominy feed have been derived from the original kernel of corn.

The processes of producing Grape-Nuts, including milling of flours (see Fig. 5), mixing, baking, drying, grinding, sifting, and packing, are operations performed upon the material in transit. Probably the most complete system of mechanical handling is evidenced at the Grape-Nuts bakery. Blended flour is conveyed from storage bins, weighed in hopper scales, and dropped into mixers; similarly, salt, yeast, and water are added in measured quantities. After mixing (see Fig. 6) for a predetermined period, the mixer is automatically stopped,

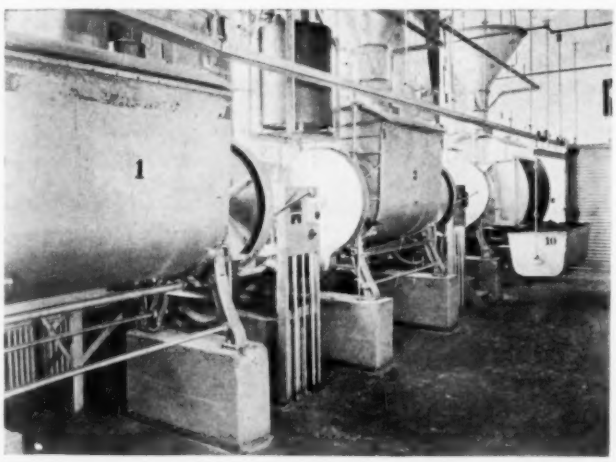


FIG. 6 DOUGH MIXERS, PREPARING GRAPE-NUTS FOR THE BAKING PROCESS

(The material is fed to the mixers by gravity; the finished dough is conveyed from the mixers in tubs on a monorail track.)

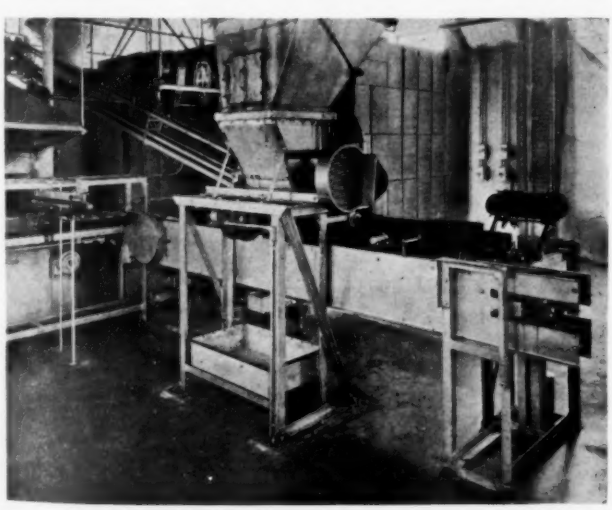


FIG. 7 AUTOMATICALLY FEEDING LOAVES OF GRAPE-NUTS DOUGH INTO THE OVEN

(Pans are sprayed automatically with grease while in transit, are turned over, and then filled with dough by an automatic feeder, and conveyed to the oven.)

electrically operated doors open from the raising room, and a dough tub traveling on a monorail track is placed in position to receive the dough. (See Fig. 6.) Pressing a button tips the mixer and discharges nearly a ton of dough into the tub, which is then guided to a position in the rising room to remain for a definite period of time under regulated conditions of temperature and humidity. After a proper interval for yeast action, the batch is positioned over a hopper, and dough is dropped into molding machines. Here it is automatically divided into loaves and dropped into baking pans at a rate of eight loaves per minute for each of three baking units. (See Fig. 7.)



FIG. 8 DELIVERY END OF CONVEYOR-TYPE OVENS BAKING GRAPE-NUTS LOAVES

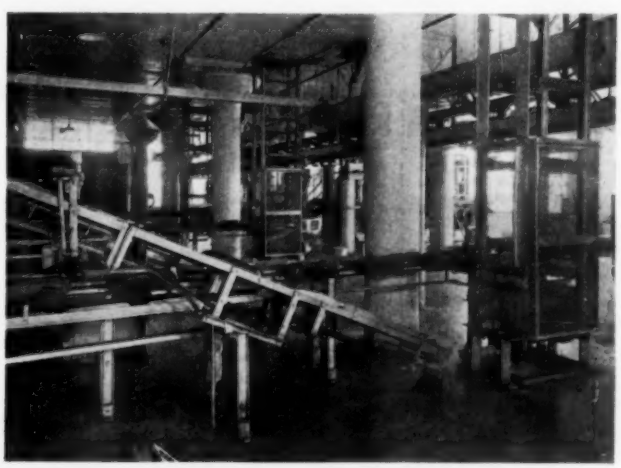


FIG. 9 GRAPE-NUTS LOAVES ON DROP-SHELF CONVEYOR, TRANSFERRING TO ELEVATOR

(Note the pan-conveyor timing and detecting devices.)

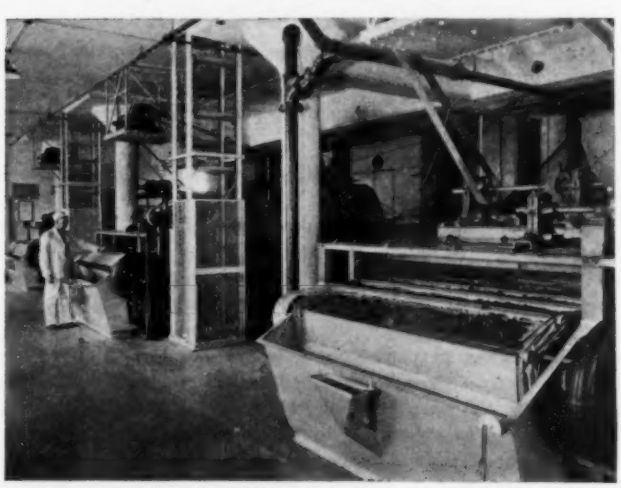


FIG. 10 CONTINUOUS CONVEYOR SECONDARY OVENS, PRODUCING GRAPE-NUTS TO BE GROUND AND PACKED

Baking pans are conveyed from the discharge end of the oven by a drag conveyor, to be passed through a spray greaser that covers the inside of the pan with a thin film of vegetable oil. Pans are timed to receive a loaf of dough from the divider, then pass to the oven inlet, where they are automatically arranged and fed into the oven in rows 17 pans wide. These ovens are of a conveyor type of traveling hearth with temperature and humidity automatically controlled to produce uniformly baked, golden-brown loaves.

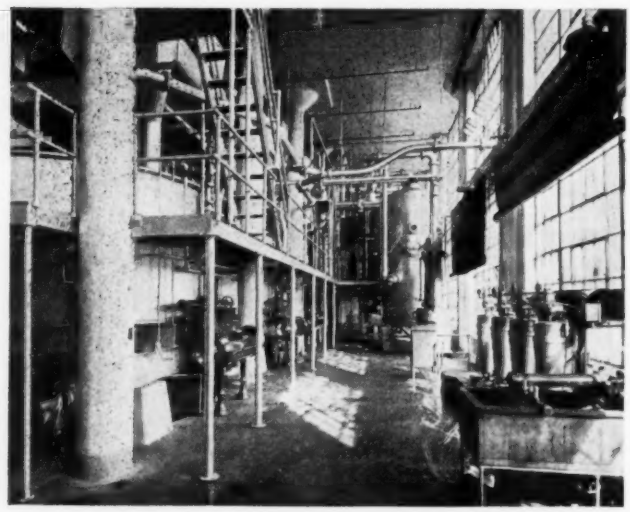


FIG. 11 PERCOLATORS, CLARIFIERS, AND EVAPORATORS PREPARING INSTANT POSTUM

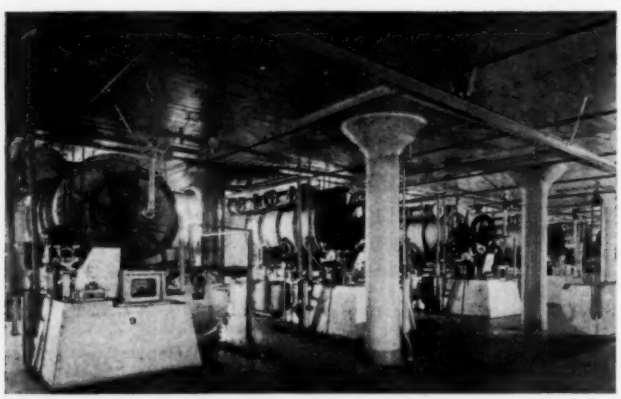


FIG. 12 VACUUM-TYPE DRUM DRIERS PRODUCING INSTANT POSTUM TO BE GROUND AND PACKED

(Liquid is supplied to the driers through pipe lines. The finished product is transferred from the driers to the grinding department on monorail tracks.)

Grape-Nuts, in the freshly baked state, are quite soft; hence it is necessary to handle the product manually and to inspect each loaf—overbaked portions, of which there is a very small amount, must be set aside to be trimmed. (See Fig. 8.) A loaf is removed from the pan and inspected, then placed on a drop-shelf type of conveyor, passed through a counter and transferred to a specially designed bucket elevator, where it is conveyed to the sixth floor for breaking and secondary baking. (See Fig. 9.) The shelf loaf conveyor is arranged to signal the inspector should a pan happen to lodge on the loaf conveyor—this is a protection to prevent a metal pan getting into the loaf breakers. Baking pans are inverted and placed in a magazine-type conveyor that automatically synchronizes these pans to be conveyed to the inlet end of the oven for receiving the unbaked loaves. An electrical device signals the inspector if two pans are together or if a pan is missing from the conveyor.

Returning to the loaf elevator, an automatic trip is arranged to distribute loaves alternately between two breakers that spread the broken particles of bread over a wide belt and deliver into a conveyor secondary oven. Food passes over eight conveyor flights designed to handle relatively soft, moist particles within the first half and smaller dry particles within the other half. The last conveyor section moves the food before an inspector, who is charged with responsibility of color and quality; this inspector can control the flow of food to reject all or a portion, if not up to standard. (See Fig. 10.)

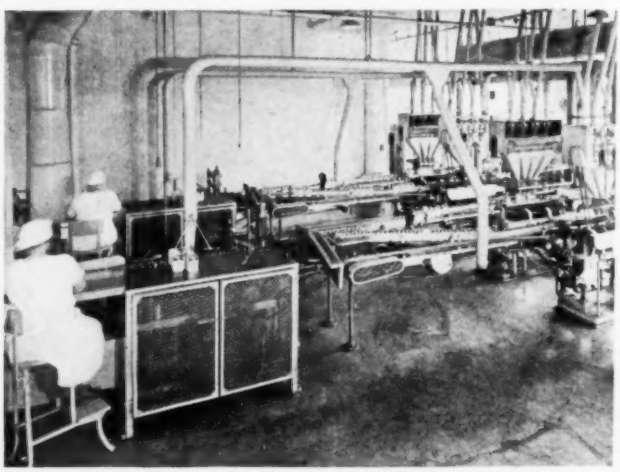


FIG. 13 FILLING AND INSPECTING CANS OF INSTANT POSTUM

(Food is supplied to scales and then to cans by gravity. The cans are filled in transit, the caps are applied automatically, and then the cans are transferred to packing tables.)

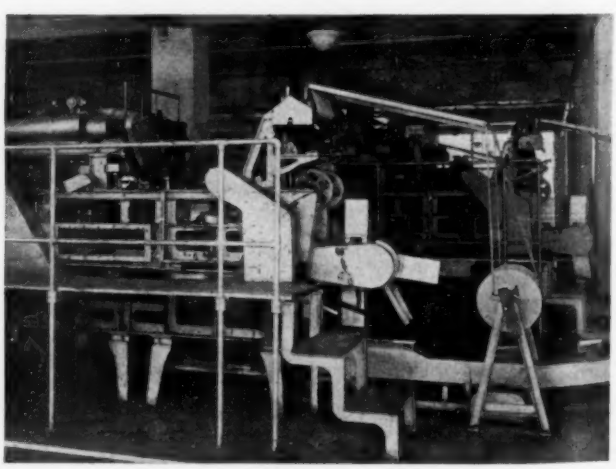


FIG. 14 CARTON-LINING MACHINE FOR POST TOASTIES

(All materials are assembled and the cartons are formed in transit.)

A troughed belt conveyor gathers finished baked cereal from three ovens and distributes this to reserve bins for grinding, sifting, and packaging.

Postum and Instant Postum are handled by machinery at all points of the roasting, blending, percolating, clarifying, evaporating, grinding, sifting, and packing operations; at no point in the process is it necessary to handle the food manually. (See Figs. 11, 12, and 13.)

In handling a vast number of identical units for 24 hours

each day, there are wonderful opportunities to develop special handling equipment. Post Toasties involves a production of more than five hundred thousand packages per day. Previously, it was necessary to place the carton on a block, glue the bottom, place paper sacks within the box, and hold these sacks beneath the filler spouts. Modern lining machines cut the proper size of sack paper from a large roll, form and glue this sheet into a sack on a moving block conveyor, remove one unglued flat carton blank from a magazine, convey this to the proper location, synchronized to meet the block carrying the made-up sack, then form this carton around the sack-covered block, glue the side seam and bottom of the box, as well as the liner to the inside of box. (See Fig. 14.) The finished box ready to receive food is dropped to a short conveyor, filled with flakes, tested for full measure, the sack is folded, closed, and top sealed. (See Fig. 15.) At this point it leaves the machine to be conveyed to an automatic wax wrapper, where a protective covering of wax paper, characteristic of Post Products, is placed around the package and sealed moisture-tight. This operation is an interesting example of handling, for a proper sized sheet of waxed paper is cut off a roll and conveyed to a position over the package "well." Paper is timed to arrive just as a package

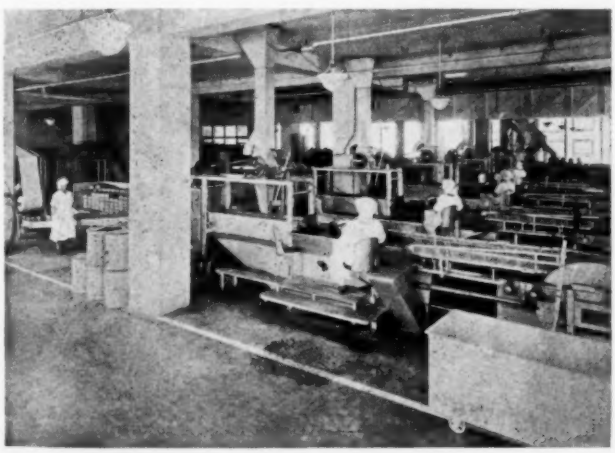


FIG. 15 FILLING AND INSPECTING PACKAGES OF POST TOASTIES WHILE BEING CARRIED ON BELT CONVEYOR

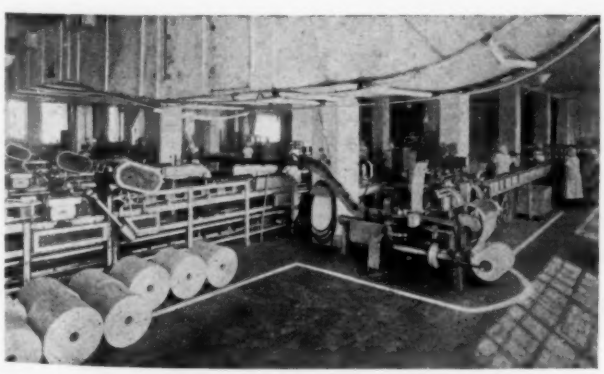


FIG. 16 SEALING, WAX-WRAPPING, AND PACKING POST TOASTIES WHILE ON BELT CONVEYORS

is being elevated through this "well," and it is literally enveloped by wax paper. While the package and wrapper are in motion, the trailing edge of paper is tucked beneath the leading edge at the side of the well, the ends are folded by a mechanical thumb and finger, and the paper is sealed by passing first over electrically heated hot plates to melt the wax contained on the paper, then over cold plates to set the seal. (See Fig. 16.)

The finished wrapped package is conveyed to packers who place the required number in cases and push them on a conveyor to be sealed and conveyed to the shipping floor. (See Fig. 17.) Within one of the departments automatic packing has been used successfully; a girl places the case in the packing



FIG. 17 CASES OF POST TOASTIES CONVEYED FROM PACKING LINES TO SHIPPING FLOOR BY BELT AND GRAVITY

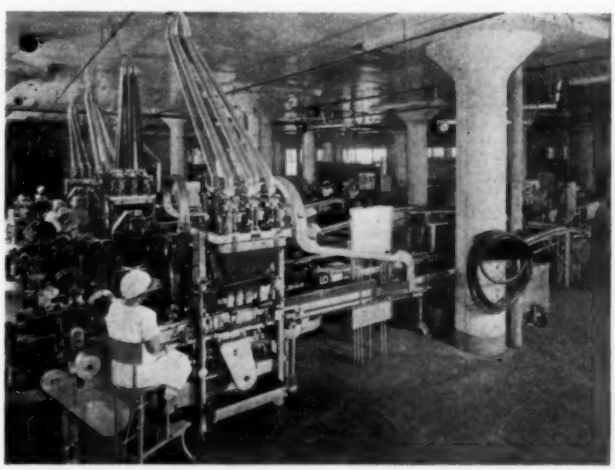


FIG. 18 FULLY AUTOMATIC HANDLING OF CARTONS TO PACK GRAPE-NUTS

machine, the packages are automatically inserted, and the case is transferred to the sealing conveyor.

Other types of packaging that do not require sack liners employ a small and compact machine to produce these units. Folded cartons are fed from a magazine by a mechanical finger and held by vacuum cups to a turret-type cam conveyor that transfers the folded cartons to a second turret conveyor, where cam-operated fingers grasp the cartons, force them open, and place them upon moving forms. The cartons pass beneath glue rolls and pressure belts to form boxes ready to receive food. Arriving at a point directly beneath a chute, a jet of air sweeps the box in an arc to the other side of the machine, where it lands open end upward to be conveyed beneath scales and receive the correct amount of cereal. Each box is checked for quantity and automatically rejected if not properly filled. Tested packages are conveyed beneath the glue rolls and pressure belts once more, where tops are sealed, then conveyed to

wax wrappers and packers. Each one of these machines will handle nearly one hundred thousand packages in 24 hours. (See Fig. 18.)

Products packed in fiber shipping cases involve a specialized system of handling equipment that is elaborated upon at the end of this paper.

Paper and boxboard material for cartons and cases are received at the west end of the plant; this moves steadily east-

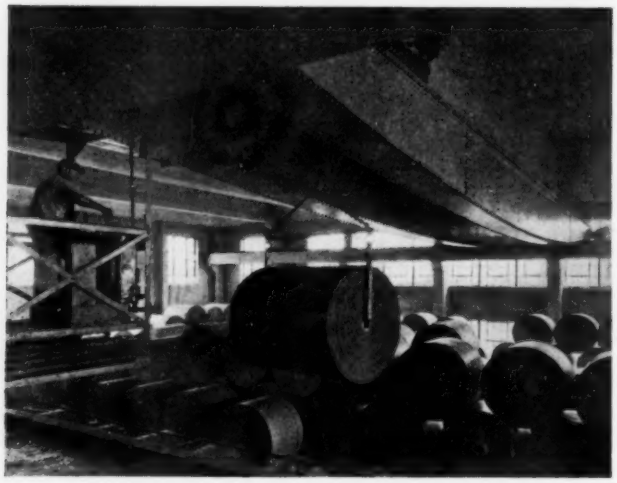


FIG. 19 HANDLING ROLL PAPER READY TO BE MADE INTO STOCK FOR SHIPPING CASES



FIG. 20 PRESSES PRINTING CARTONS AND DISCHARGING TO PORTABLE HANDLING TRUCKS

ward during the processes of board making, printing, cutting, and stitching or gluing; thence to receive finished products in the packing departments. In addition to packing materials for Post Products, this department furnishes cartons and cases for many other divisions of General Foods. There are many interesting examples of materials handling in this department, including traveling cranes to handle heavy roll stock (see Fig. 19); machines that print two colors, (see Fig. 20) as well as cut and crease shipping containers (see Fig. 21); a machine that glues the side seams of Jell-O cartons at a rate of one million per day (see Fig. 22); and cutting machines that cut and crease as many as twenty million cartons with one set of dies (see Fig. 21).

Handling materials for waxed wrapping paper involves the movement of heavy loads by special trucks and traveling chain-hoist cranes; similarly these materials and the finished product must be handled under exact conditions of heating and cooling; in fact, this situation is generally confronted in all departments of the plant. Reliance is placed almost entirely

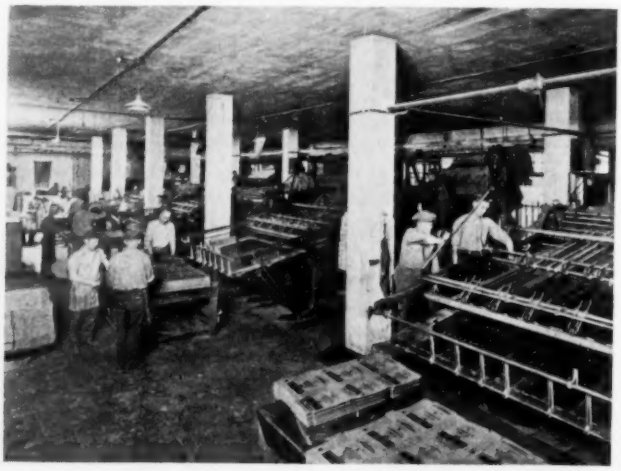


FIG. 21 CUTTING AND STRIPPING CARTONS, MANUAL HANDLING TO PORTABLE PLATFORMS



FIG. 22 GLUING CARTONS, WHICH ARE HANDLED ON BELT CONVEYORS



FIG. 23 LIFT TRUCKS FOR HANDLING CARTONS AND CASES

upon automatic control of temperature and humidity to govern manufacturing conditions. (See Fig. 24.)

HANDLING CASED PRODUCTS

When Mr. Post started to make Postum in the little white

barn on the site of the present plant, he little realized the extent to which the business would develop. As business grew, buildings were constructed, generally hurriedly, to take care of the phenomenal demand. Thus the plant expanded along the hillside orchard, and its location became quite firmly established. Although the site was far from ideal and the general contour of the land made it practically impossible to develop a plant with several distinct operating units grouped about a central warehouse, it is fortunate in that two large railroads are adjacent to company property and good railroad facilities are provided. Moreover, the products are generally bulky

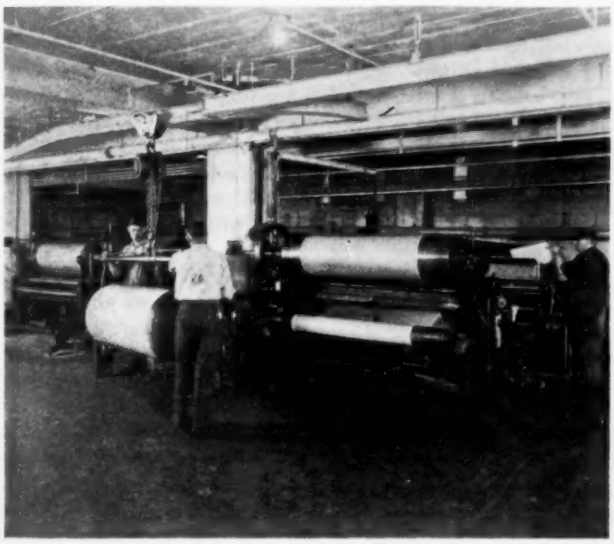


FIG. 24 MANUFACTURING WAX PAPER FOR WRAPPING CARTONS, THE ROLLS OF PAPER BEING HANDLED ON MONORAIL TRACK



FIG. 25 RECEIVING CASED PRODUCTS AT SHIPPING FLOOR TO BE LOADED INTO CARS

and produced in tremendous quantities—over a million packages of all products are produced daily.

The next best arrangement was to provide warehouse space for each product within the producing unit and to provide each unit with ample trackage; thus the loading of complete cars of a single product is not complicated. (See Fig. 25.)

It is the aim of the Postum Company to have its products delivered to the consumer as fresh as possible. For this reason it does not depend on maintaining a stock of all products in

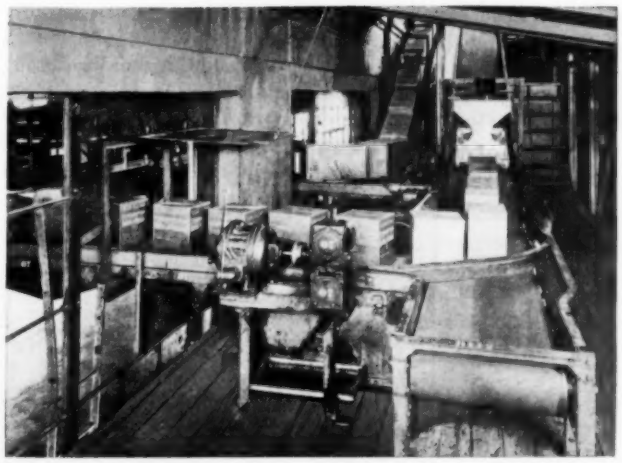


FIG. 26 CASES LEAVING GRAPE-NUTS BUILDING AT SAME TIME AS A LOAD OF POST TOASTIES IS BEING RECEIVED

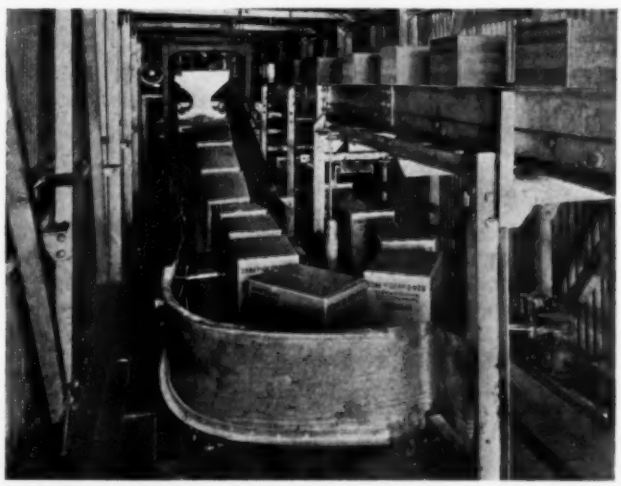


FIG. 27 SHIPMENT OF GRAPE-NUTS PASSING EAST JUNCTION INTO POST TOASTIES BUILDING, PILOT WITH LOAD OF BRAN FLAKES READY TO FOLLOW
(Note cases changing direction at 180-deg. turn.)

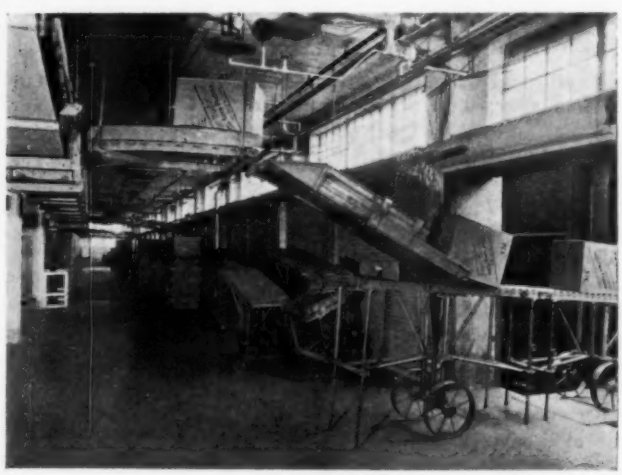


FIG. 28 CASED PRODUCTS LOADED INTO CARS BY OVERHEAD BELT CONVEYORS AND PORTABLE SECTIONS

various warehouses to supply local markets, but in practically all cases ships direct from the factory to the distributor. This means that each car leaving the Postum Company generally contains some of all of the products, and frequently these are in small lots to several customers who have combined their orders for carload shipping rates.

To assemble the various products at one or more points for

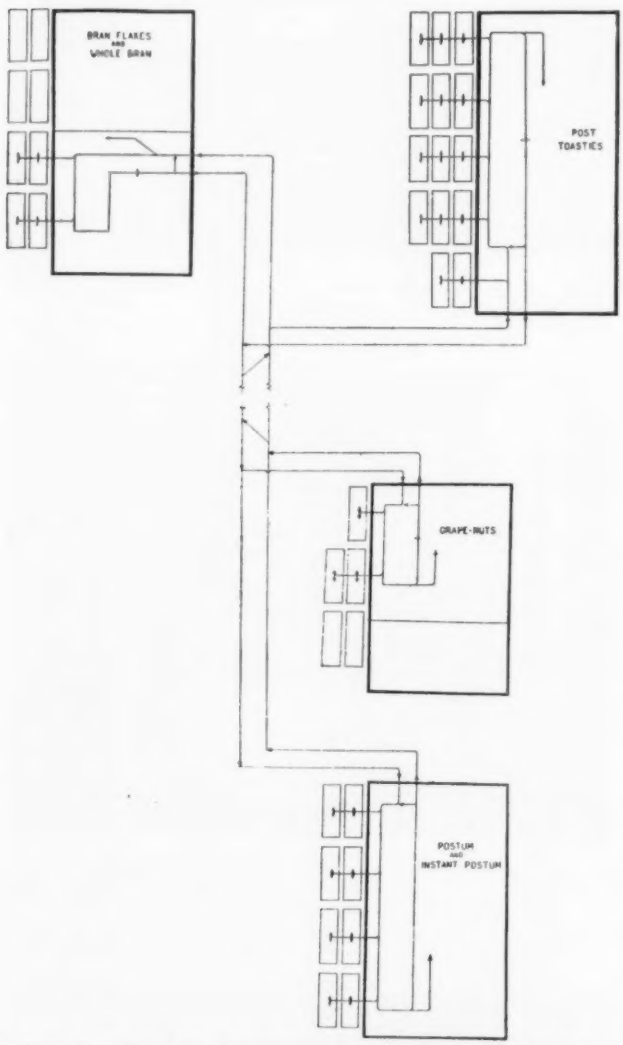


FIG. 29. PLAN OF MAIN LOADING CONVEYORS CONNECTING MAIN PRODUCTION UNITS

loading into cars has been a perplexing problem. Numerous conveying-system studies were made and installations investigated with the result that the present intercommunicating pilot-car package-loading conveyor system was installed.

These products are produced in four different sections of the plant, and although there are four group loading points, it is necessary to convey a portion of each carload from one or more of the three remaining sections. For example, a car may contain 60 per cent Post Toasties, 20 per cent Post's Bran Flakes, 5 per cent Post's Whole Bran, 10 per cent Grape-Nuts, and 5 per cent Postum. Naturally, of the four loading points, the one most convenient to Post Toasties would be selected, and the remaining products conveyed to the loading point. Before the conveyor was installed these miscellaneous cases were handled by auto truck or freight cars switched between the different buildings. Not only was this costly, but "stock" cars occupied valuable loading space and double handling was necessary. At present it is possible to dispatch the necessary number of cases to a designated car, and the so-called pilot car traveling ahead of the cases will automatically operate the system of conveyors leading into the proper car. Providing that certain portions of the system are in use, the train of cases will automatically stop until the system is clear, then proceed to the designated car.

A brief description of "pilots" and "cabooses," as used with this system, would serve to explain the almost human performance that one is able to obtain by electrical and mechanical means. Pilots and cabooses are identically the same and fully interchangeable. A pilot is so termed when it is leading a train of cases and preparing the various interferences and deflectors to "lock out" other trains of cases. A caboose is so termed when it is used to follow a consignment of cases. Both consist of a heavy base with vertical guide rolls at the four corners; these enable the car to move around curves easily. An upright section, mounted transversely in the center of the base, is arranged with 16 pinholes.

Each dispatching point is supplied with long and short pins to be placed in the pinholes of the pilot or caboose. A dispatcher at any point, wishing to send a consignment of cases to any car or building, consults a chart for proper pin settings, and places his pins; for example, short No. 2, long No. 6, short No. 7. This pilot, passing through the system with cases following, will contact all electrical lock-out interferences with the high pin and prevent other trains on branch conveyors from entering the system and colliding with the cases that follow. (See Fig. 26.) If the train-load is about to approach another section of the conveyor that is already conveying cases, the belt conveying the train stops at the instant the high pin contacts the interference switch; the passing of the caboose of that train instantly starts the waiting train. (See Fig. 27.) At a certain point in the system, short pin No. 2 contacts a tab that no other pin setting could operate, and electrically moves the proper deflector to divert the train to the proper route. Short pin No. 7 may operate to throw another deflector and further direct the train properly. Arriving at the destined building, long pin No. 6 contacts a tab and throws a deflector between the pilot and the cases following to divert these into the proper car. Meanwhile the pilot continues straight ahead and automatically comes to rest on a reserve section of conveyor, ready to be used on another mission. (See Fig. 28.) The caboose, with a long pin at No. 16, follows the train of cases and operates tabs to replace all interferences and deflectors at neutral; it operates the car-loading deflector before arriving at the deflector and after the last case had passed to the car, so that the caboose may pass to the reserve conveyor.

Simultaneous loading can be carried on at the four main units of the plant, where a total of 29 cars may receive cases via conveyor, during half of the 24-hour day, then to be hauled away and replaced with a similar number of cars for the last half of the day. When 58 conveyor-loaded cars cannot handle the day's shipping requirements, there are additional car-loading points that can be made available to meet the emergency.

The loading procedure by conveyor (see Fig. 29) is as follows:

Cars are cleaned and lined with paper.

Heavier products, Grape-Nuts and Postum, are always loaded at the bottom or ends of the cars; lighter foods, Post Toasties and Bran Flakes, are loaded on top of the heavier products. For example, a typical car may contain 500 cases Post Toasties, 50 cases Post's Whole Bran, 100 cases Post's Bran Flakes, 200 cases Grape-Nuts, 150 cases Instant Postum.

It is readily seen that this car will be placed for loading at the Post Toasties department. The loading foreman tele-

phones his orders to the respective departments for delivery to car at No. 59 door. Grape-Nuts are given the right of way for they must be placed in the bottom of the car. These are loaded to the conveyor at the Grape-Nuts department and dispatched to No. 29 building (Post Toasties department). The load of cases may be temporarily held to await a consignment of Bran Flakes that is finishing a car at No. 58 door. Cases are handled at rates as high as 2000 per hour; hence the delay is comparatively brief. As the "caboose" of the Bran Flakes cases passes the junction point the "pilot" for Grape-Nuts cases automatically follows, and leads into the Post Toasties department. At No. 59 door the "pilot" operates a deflector to send cases over portable conveyor sections into any one of three cars at this door.

Meanwhile, Instant Postum cases have been loaded to the conveyor; these arrive at the junction to find Grape-Nuts already passing. Postum cases automatically halt until the Grape-Nuts have passed, then automatically start up and follow into the Post Toasties department, where they are diverted into the proper car. Bran products, having been dispatched from a third section of the plant, arrive outside of the Post Toasties department to find Grape-Nuts and Postum occupying the right of way. Likewise this train of cases automatically awaits the passing of the "caboose," then follows into the Post Toasties building. Thus there are assembled all required cases with a count both at the dispatching point and at the car. Our consignment of Postum cases may have been a mixture of several different sized cases; similarly with Grape-Nuts and Bran Flakes or Whole Bran; but a double check is always forthcoming and no confusion results. During the time Grape-Nuts and Instant Postum cases are flowing to the Post Toasties shipping point, Post Toasties may also be moving to cars at other buildings.

Portable conveyors are changed and miscellaneous loading is carried on at one of the other cars at No. 59 door, or portables are quickly rolled to cars at one of the five other loading doors. Post Toasties cases are then loaded into the typical car, from an adjacent area, with platform trucks. There is no economy in handling cases from the immediate shipping floor into nearby cars by conveyor, except in a very few instances; moreover, the conveyor is usually occupied in bringing miscellaneous cases into the loading department or in conveying Post Toasties to other loading points.

In event that the bulk of the load is Grape-Nuts, Postum, or Bran products, the car is located at the department representing the least handling, and other cases are conveyed to the department, or loaded direct from reserve stocks if any are available within the department.

Within recent years a considerable amount of autotruck hauling for local distribution within a hundred-mile radius has been built up. These trucks often move in trains of two or more trailers. The conveyor has made it possible to assign these truck trains to a convenient loading point within the plant to which all products may be conveyed; thus no confusion arises from operating these truck trains through the busiest parts of the works. Without the conveyor it is questionable if these trucks could be loaded economically.

Mechanically, the conveyor loading system comprises some two miles of belting. High-friction rubber belting is used for inclines, none of which exceeds 22½ deg. pitch. For level runs a heavy sewed canvas, oil-impregnated belt is used. All supporting structure is guarded to prevent injury to cases in transit.

Operation is accomplished by means of 46 alternating-current motors, varying in size from 1 to 10 horsepower. These may be manually operated, or if the system is occupied by a train of cases, control is automatic in accordance with the pin setting of the "pilot" car.

Both mechanically and electrically, the automatic conveyors operate with comparatively little trouble; an experienced maintenance man is assigned to the system for purposes of inspection, adjustment, and lubrication.

Discussion

R. K. Merrill.[2] One who has gone through this plant, as has the writer, could observe the orderly and cleanly appearance of the layout. Not only does this result in a reduction of cost under that which would result with other methods of handling, but the hand trucking of materials would certainly be the cause of an enormous amount of dirt and dust, which in turn would increase the cleaning expense in proportion. The handling of foods by mechanical methods is almost a necessity in this day when dietitians and medical authorities are insisting on careful attention to these details to prevent contamination and spoilage. In the recent past there have been numerous outcries against the cost of packaged foods as against the cheapness which seemed to obtain in the old cracker-barrel days when the grocer lifted out of the barrel a scoopful of crackers, assisting the scoop with hands which had probably been engaged in sorting potatoes or pouring out a gallon of kerosene. It is probably true that the Postum Company had no choice in the competitive situation which exists today but to completely conveyorize their operations.

Where the shipping containers of a product are fairly uniform in size and shape, as in the case of the Postum Company, the system described seems to fit very well. However, it should be pointed out that mixed carloads of other classes of material, such as furniture composed of wood and metal parts, are best handled by tractor and trailer system or by means of electric lift trucks handling skids, since the odd-shaped crates and multiplicity of sizes do not lend themselves so readily to handling by belts, elevating conveyors, and transfers. The quantity of boxed material handled also has some effect on the type of handling equipment to be selected for this work.

The mechanical engineering fraternity is somewhat familiar with the enormous amount of experiment and thought necessary to produce such a layout as the author has perfected and which is described in this paper. It is unfortunate that the average layman can have no conception of this phase of the matter.

George H. Dettling.[3] The writer would ask these questions: What saving has been effected by the use of the selective conveying system for handling the cartons to various loading points? Has the author made any saving study, and is the system producing definite results as regards to saving? How much lost time in conveyor operation is caused by electrical trouble due to switches, pilot cars, and cabooses getting out of order? What provision is made for taking care of the loading of cars when the conveyor system is out of order? Does the Postum Company maintain a trucking system to replace the conveyor system when, due to electrical trouble or other causes, the conveyor system is out of order? Are there any cost figures on operation of the conveyor system. The author mentions that there is no economy in handling cases from immediate shipping floor into cars by conveyor except in a very few instances. The writer would like to learn something more in connection with this. It is noted that he specifies the use of platform trucks for this type of conveying. The paper indicates that the Postum Company has

[2] Mechanical Engineer, American Seating Company, Grand Rapids, Mich. Mem. A.S.M.E.

[3] Material Handling Engineer, Grand Rapids, Mich. Assoc-Mem. A.S.M.E.

made a very thorough study of their conveying problems, and it should be complimented on the system worked out.

C. M. Bonnell, Jr.[4] The paper indicates that, by the application of proper materials-handling equipment, costs of production can be very materially reduced in an industry such as the Postum Company, where the receiving of materials, packing of same, and the arrangement for shipping constitute the principal part of production and marketing.

Exception is taken to that statement which might be construed to classify such machines as two-color presses, cutters and creasers for shipping containers, gluing machines, etc., under the classification of materials handling.

It is interesting to note the statement that when cars are loaded such heavier products as Grape-Nuts and Postum are always placed at the bottom or ends of the cars. This is interesting, for the reason that a few shippers give consideration to this point, but the majority, it is believed, do not. Some contend that, if the fiber container is certified by the railroads to carry a certain weight, then it should be satisfactory if placed in any part of the car. Perhaps this position is so taken because some shippers do not wish to be bothered by what they consider a trivial matter. Others claim that it would be difficult always to load the heavier containers in the bottom.

Another statement is made in reference to distribution to the effect that in recent years a considerable amount of motor-truck hauling for local distribution within a 100-mile radius has been built up. The use of motor trucks for short hauls undoubtedly is increasing, and it is presumed that the Postum Company has adopted this method because it is cheaper and quicker than by the use of l.c.l. freight service of the railroads. It is not stated whether the company owns its own trucks or whether it employs the services of local trucking companies.

[4] President, *Packing and Shipping*, Bonnell Publication, Inc., New York. Assoc-Mem. A.S.M.E.

Author's Closure

The regularity and uniformity of established shipping cases lend every advantage to the possibility of mechanical handling. As for conveyor operation, the loading costs have been lowered and the loading speed increased. While the conveyor is not free of interruptions, it is reasonably in line with other automatic equipment with which we are experienced. It has not been necessary to resort to previous methods of intra-plant hauling and transferring. The conveyor as now installed is primarily to transfer cases between unit buildings; thus it is usually occupied with these duties and should not be interrupted by lesser handling costs.

MH-53-5

Mold-Handling Methods in Foundries

By WILLIAM L. HARTLEY,[1] CHICAGO, ILL.

This paper analyzes the mold-handling problem in foundries, first listing the important features to be considered in a study of the application of mold-handling devices. Present-day practice in intermittent and semi-continuous foundries is reviewed. The numerous advantages and reasons for lower cost production in continuous mold-handling operations as compared with floor methods are given. The design and application of continuous power-operated mold conveyors are discussed, with illustrations of several types.



THE handling of molds in the foundry covers a very broad field when consideration is taken of grey-iron, malleable, steel, bronze, and aluminum foundries, each having a great variety of products and range of production. The importance of the mold-handling phase of foundry production will be appreciated when it is realized that it generally represents between 20 and 40 per cent of the total materials-handling effort in the average foundry production work. As foundrymen and foundry managers have become more and more mechanically minded, the mold-handling operations have been given a great deal of consideration, along with handling equipment for sand, castings, cores, metal, etc.

In all types of foundries certain fundamental factors apply to their problem of mold handling as regards method, arrangement, and type of equipment, all of which have a direct influence on production costs.

The following is a tabulation of these important fundamentals:

1 *Molding Method and Arrangement of Machines.* It is of major importance to obtain maximum production from molding machines, patterns, and molding operators. Every thought should be given to relieving molders from all duties but making molds. To illustrate this point, numerous cases can be cited in high-production repetitive work where relieving the molder even of setting cores, a helper doing this work instead, has brought about a marked increase in production per man and machine, and thereby lower production costs.

2 *Size and Variety of Molds.* Any means of mold handling must provide sufficient flexibility and be suited to production requirements. Where a wide variety of work must be handled or when a shop does not have full control of its own products and is likely to encounter frequent changes in classes of work, flexibility is of major importance in the successful application of mechanical devices.

3 *Quantity of Production.* Careful planning must guard against too elaborate or expensive equipment for material or tonnage to be handled. Coordination of plans with production requirements is of vital importance, as equipment mu t justify itself from a cost standpoint and tonnage basis, as well as for its operating features. An unusual illustration of the point in question was a case where a complete mold-handling unit was installed for a particular job to handle a two-year production contract and, because of circumstances, was of no further use on completion of it, yet the unit well justified itself from a cost standpoint, for the one contract alone.

4 *Setting Cores, Finishing, Drying, and Closing.* Each of these operations must be given its proper consideration in every mold-handling layout. On many installations efficient handling and delivery of cores is a vital factor to the successful operation of the molding unit.

5 *Foundry Layout and Available Floor Space.* Quite often volume of production can be more than doubled from existing floor space by the addition of mechanical equipment costing less than half of what new buildings would cost to obtain an equivalent increase in production.

Because of the tremendous saving in floor space by the use of mold-handling equipment, as compared with floor methods, it is nearly always found possible to provide an effective arrangement of mechanical devices for old foundries, with minor building alterations.

6 *Flow of Material.* Proper relation of pouring zones to melting furnaces, shakeout and delivery of castings to cleaning room, and delivery of cores to molders are of prime importance. To establish the most economical and effective arrangement it is sometimes necessary to make a most thorough analysis of comparative material-handling costs. For example, it is quite generally a fact that hot metal can be conveyed in ladles some distance to molds at a lower cost than molds can be conveyed to the hot metal.

7 *Intermittent or Continuous Molding, Pouring, and Shakeout.* There is no doubt that continuous operations most nearly approach the ultimate in efficient and low-cost production when continuous power-operated mold conveyors are utilized. For intermittent operation more careful planning is necessary in estimating cost of equipment and returns on investment, for, strange as it may seem, the cost of proper mold-handling equipment for batch operation will often be more than the cost of continuous conveyors.

8 *Handling Jackets, Weight Shifting, and Clamping.* These operations in their proper order, with provision for reclaiming and convenient storage of the parts, are most essential. On several jobs handling molds on gravity rolls where it was possible to standardize flask sizes, weights are suspended above and are placed on the molds merely by shifting a lever; thus a considerable saving in time and labor is realized.

9 *Storage of Molds and Proper Cooling Time.* An even balance between mold production and melting capacity is most essential to avoid waste, yet it must be kept in mind that excess mold storage, beyond that necessary for flexibility of operations, is also a waste that will reflect itself in production costs. With regard to capacity of continuous mold conveyors, 25 per cent over the estimated mold production, to take care of peak periods and equalize for missed trays, has proved in practice as quite generally a conservative basis for figuring.

The pouring operation deserves careful attention in regard to handling of metal and method of pouring, and particularly

---

[1] Engineer, Foundry Division, Link-Belt Company. Mr. Hartley has been connected with the Link-Belt Company in engineering and sales since 1916. He has specialized on foundry-equipment sales for the past five years.

Contributed by the Materials Handling Division and presented at the Semi-Annual Meeting of The American Society of Mechanical Engineers, Birmingham, Ala., April 20 to 23, 1931.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

as to the size of molds and length of time necessary to pour. The mold-handling problem is often complicated where numerous molds are poured through two or more gates.

Because of its effect on quality of castings and the size and capacity of the mold-handling unit, proper length of cooling time deserves very careful study, particularly in the continuous grey-iron and malleable foundries.

10 *Method and Arrangement of Shakeout.* Every effort should be made to reduce manual labor at the shakeout, particular attention being given to speed and ease of operation of mold-handling devices. A very marked improvement in mold-handling and shakeout arrangement has been made on several continuous snap-mold units where simple and effective devices have been developed that automatically dump the mold at the shakeout without manual control, thus eliminating one of the most laborious tasks in the foundry.

11 *Flask and Bottom-Board Return.* The importance of this item should not be overlooked, particularly where high-speed production of molds is to be considered. This feature accounted for a difference in production of 15 molds per hour on a green-sand mold made on jolt-squeeze machines. This was not an unusual job except in the high production obtained, but it is cases of this kind that bring such facts to light.

Mold-Handling Method in Semi-Continuous Foundries

Under this classification would come practically all steel foundries, non-ferrous (with a few exceptions), and small production grey-iron and malleable foundries.

Manual. Where the molder carries out the molds he makes or has a helper to do this work. It would seem unnecessary to mention a method so well known and common in practice, yet it is just to the foundries still using this method that knowledge on the subject may be of most interest and value. For reasons apparent to all, it is difficult to establish hard and fast rules that will show with accuracy at just what point in production volume the carry-out method becomes uneconomical. However, for light and medium work, that is, molds weighing up to 250 lb., it can be stated with confidence that mechanical mold-handling equipment will be well justified where a production of 3000 or more molds per day is involved. For heavier work, mechanical devices become a necessity, and it is a question of the speediest and most economical.

Crank, Jib, or Hoist. Operated manually, electrically, or by air; where the crane operator takes the mold from the closing station and places it on the floor or where the jib and hoist assist the molder or helper in the carrying out; also, where a number of molds are placed on a long frame or skids, being taken from the molding station by the crane. When a single crane is depended upon to serve a number of molding stations, frequently delays occur that penalize operations to the extent that individual hoists for each molding station prove a profitable investment.

Skids and Trucks. Electric or gasoline. Where the molds are loaded on skids by the molders, with a lift truck for moving the skids to the pouring zone, and then to the shakeout station, returning the skids and flasks to the molders.

Gravity Roll Conveyor. Where the molder places the molds on gravity roll lines, which conveys them by gravity or hand power to the pouring zone, then to the cooling zone, and then to the shakeout station. Empty flasks and bottom boards are returned to the molder on a second run of conveyor. In some cases air cylinders have been used to push the molds along the conveyor, particularly for large flasks on production jobs, such as bathtubs, truck side frames, etc.

The roller conveyors provide a mobile storage of molds, and can be arranged to enable the molding operator to load his finished section or mold directly at the molding machine. Many effective layouts handling molds on roller conveyor are so arranged as to concentrate the pouring in comparatively small areas and to deliver the poured molds to a central shakeout point.

To move the great weight with ease, the roller conveyor must provide a minimum of friction, and this under the most adverse conditions. As the greater amount of molding is done in green sands, and demands are being made for closer specifications and finer detail and finish in castings, mold-carrying conveyors must operate smoothly, avoid destructive vibration, and be continuously maintained in this condition. They must withstand sand and metal spillage and provide an effective method of bearing preservation by preventing entrance of abrasive matter and moisture to the bearing elements. Development of roller-conveyor equipment specifically for foundry use has now reached a stage where any green-sand mold that will stand pouring can be rolled safely.

Mold-carrying systems using floor gravity-roll conveyors have the advantage of flexibility, selective directional movement to suit the day's work, and provide an easy means of access to a central shakeout point. They may be fitted to premises of peculiar shape and may be operated either longitudinally or latitudinally, or both, with molding points arranged in line for economy in sand-handling and other equipment.

Such systems may be altered easily in arrangement from time to time to suit changing conditions. This is readily accomplished, as no power applications are concerned, and the equipment, being above the floor level, is quickly shifted.

There are of course numerous modifications and combinations of the several methods mentioned in successful practice in many of the intermittent and semi-continuous foundries. Also, for certain conditions and requirements, very special devices offer the only mechanical means for effective handling of the molds, such as cast-iron pipe production, dry-sand molding, and stack molding, but a description of these is of little value in general.

Much has been written of the advantage that continuous molding and continuous pouring offer over intermittent operation, and there is no longer any question that continuous operation does bring about startling economies. There are many foundries today operating on a continuous basis that considered the idea impossible and foolhardy a few years ago. In numerous cases these foundries modified their production schedule, and even adjusted the design of their products so that they would lend themselves to continuous operation.

A national authority on foundry practice stated, "Any foundry manager who is interested in reducing labor costs and decreasing his foundry floor space at least 50 per cent, thereby substantially lowering the cost of castings, should fully investigate continuous melting and pouring operations."

Present-day melting methods in steel, and in most non-ferrous foundries, preclude the use of continuous mold conveyors in those fields, but it seems very likely that as steel-foundry production increases, metallurgists in that field may solve the melting phase of the problem in time not far distant, as was done in the malleable field where metal-control methods centuries old were changed to bring about a supply of hot metal for continuous pouring.

Mold-Handling Methods in Continuous-Process Foundries

The most general application of continuous-power type mold conveyors has been in the grey-iron and malleable foundries. As it is necessary to have a sand-handling system in conjunction with the mold conveyors to get their full benefit, it is almost im-

possible to give the exact savings chargeable to the mold conveyors alone. Therefore, to some extent, the following items cover the combined advantages of sand- and mold-handling devices, and apply most generally to continuous grey-iron and malleable foundries:

1 *Production Methods in the Foundry.* Continuous operation results in a steady flow of materials and products. Cleaning rooms, machine shops, and all finishing departments benefit by a systematic, orderly foundry. Continuous inspection is made possible, thereby reducing losses. It is possible to work two or three shifts during rush periods, which results in more than double saving, as it is accomplished on original investment. High-speed production lowers inventories of work in process.

2 *Floor Space.* Value of floor space is multiplied in accordance with increase in production, and thereby a saving is realized by lower taxes, insurance, maintenance, and depreciation of buildings per ton of castings produced.

3 *Saving in Flask and Molding Equipment.* A reduction of at least 75 per cent in flask equipment is generally obtained on a continuous system as compared with the old floor method; at the same time production of molds is increased. In shops having a large production of tight flask work, the saving in flask equipment alone will generally more than cover the cost of power-operated mold conveyors. In one foundry producing cylinder molds on a continuous basis, a production of 1200 molds a day was obtained with 100 flasks, the one set of flasks going through the operation of molding, pouring, and shaking out twelve times a day.

4 *Increase in Efficiency of Molders.* Conveying molds directly from the molding machine eliminates walking and bending to place molds on the floor, thus relieving the molders of their most laborious task. There is less danger of damage to molds in placing them on a mold conveyor than when the molders carry out and set them on the floor. This is true particularly when it is necessary to stack snap-flask molds on account of lack of floor space. The molder's time previously used for pouring and shakeout is used for molding, which results in continuous molding all day, with no less of molder's time waiting for iron when his floor is filled. This method gives the good and speedy molders a chance for high production, their energy not being used up in shoveling and long carry-outs; also as a result of continually performing one operation, the molders find short-cuts and become experts in their part of the system. The benefit of all-round better working conditions should not be overlooked.

5 *Systematic Pouring Reduces Costs.* Continuous pouring requires the minimum number of men working the entire day on the pouring gang, as it does away with the extra labor required when a large tonnage of metal is poured in a comparatively short time. It is generally possible to locate the pouring zone of the conveyors adjacent to or near the melting furnaces, thereby obtaining the shortest possible carry for the hot metal. Pouring at one central point simplifies pouring equipment, supervision, and metal control.

6 *Centralized Shakeout.* After pouring, molds are cooled on the conveyor and delivered to a central shakeout point, thus eliminating collecting and carrying the castings around the foundry area. Likewise the shakeout sand is delivered to one point for rehandling, establishing ideal conditions for economical continuous sand-handling equipment. The speed of the mold conveyors can be adjusted to suit production requirements, and regulated so that the molds are shaken out in the shortest safe cooling time, thereby reducing burning out of the molding sand.

7 *Pace Setter.* Continuous operation tends to speed up production and set a pace that can be maintained, because of the elimination of fatigue and back-breaking operations. With careful design, equipment can be arranged for waist-high operations all along the line.

Early Design and Development of Continuous Power Mold Conveyors

Fig. 1 is an illustration of what was perhaps the first continuous power mold conveyor installation in the country, having been in successful operation more than 40 years ago. The general design of this early mold conveyor consisted of successive carriages about 2 ft. above the floor, which formed the propelling medium

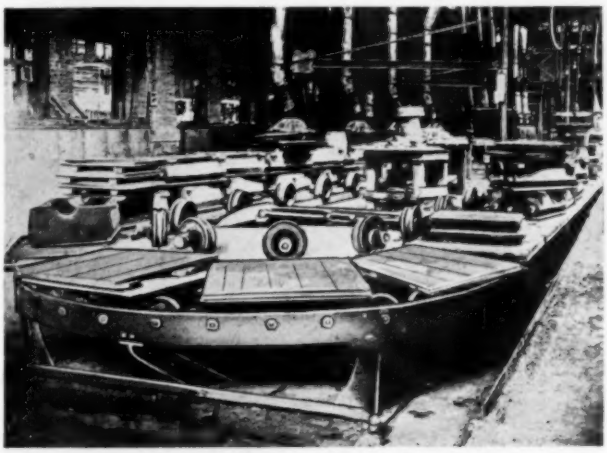


Fig. 1 An Early Power Mold Conveyor

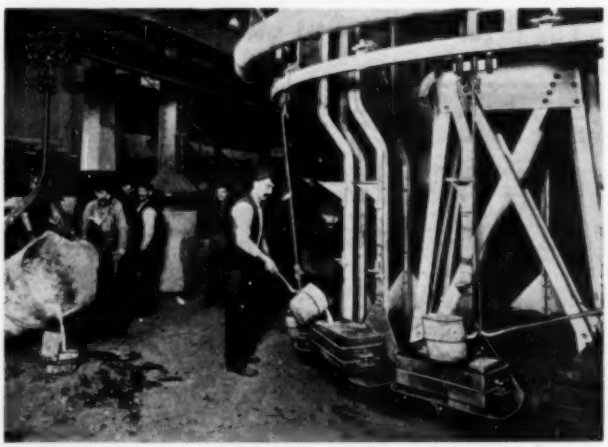


Fig. 2 Suspended Trolley-Type Mold Conveyor

and the molding, pouring, and shakeout operations were performed in much the same manner as at present.

Trolley-Type Mold Conveyors

Some years later the pendulum or trolley-type mold conveyor was developed and used quite extensively, and with much success. This design, one of which is illustrated in Fig. 2, consisted of long hangers made of pipes or structural steel, to the bottom of which were attached trays for carrying the molds. Flanged rollers are attached to the hangers at the upper end, which in turn travel on a flat steel-bar track. The chain is attached adjacent to the rollers to provide motion to the trolleys and to keep them properly spaced.

An electric motor drive, with the necessary gear reductions

and end sprockets, is used most generally on this type of mold conveyor, although a few have been built that are propelled by a wire cable and an air cylinder. The advantage of the trolley type was that the rollers, track, and chain, together with all driving machinery, were overhead and safe from falling sand or hot metal, which resulted in smooth travel and minimum wear, but required rather expensive construction, because of the necessary heavy overhead supporting structure.

Although the trolley-type mold conveyor still remains in favor with some foundry operators, it seems principally because of the impression that the trolley conveyors operate more smoothly than the floor types. However, close observation of the various designs of mold conveyors in operation does not indicate any particular advantage for the trolley conveyors in so far as smooth action is concerned.



FIG. 3 SLIDING-TYPE MOLD CONVEYOR

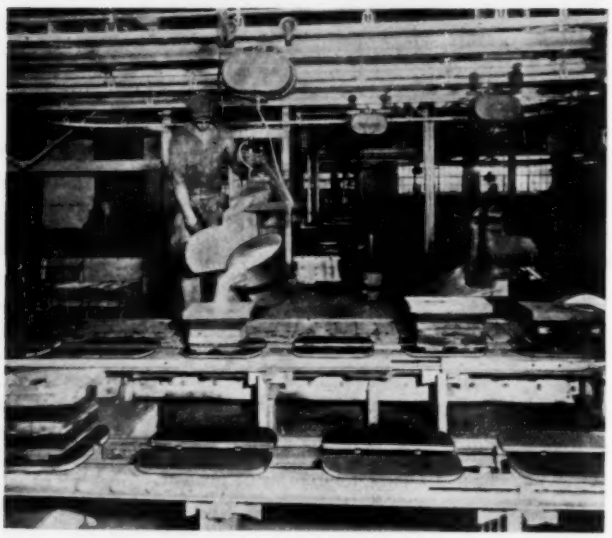


FIG. 4 POURING OPERATION ON A SLIDE-TYPE MOLD CONVEYOR

FLOOR-TYPE MOLD CONVEYOR

In the course of time a heavy, rugged chain was developed that would stand the wear and tear of sand and hot metal and which therefore did not need to be placed overhead. This did away with the costly overhead steel work, and also made it possible to locate the motors and driving machinery at the floor level.

What is commonly referred to as the slide type, illustrated in Figs. 3 and 4, is perhaps the most widely used of all the different designs. It has been in general use for a number of years and is still popular, being economical in first cost and very effective for handling light molds not weighing more than 150 lb.

Various designs of this machine have been made, all of which use the two T-rail tracks for supporting the mold tray. On the early slide-type conveyors the chain was carried on a separate pair of rails and was equipped with dogs or attachments for pushing trays along the tracks. On later designs the chain is attached directly to the trays and is carried by them, eliminating the extra rails. While in most cases the tray slides directly on the rails, a few have been made in which the trays are fitted with rollers. The smoothest travel is obtained with the sliding trays, as these keep the tracks cleared, while the rollers on the truck trays tend to ride over any obstructions of metal or sand, causing jolts, or will slide, due to shot on the rails, resulting in flat spots on the rollers.

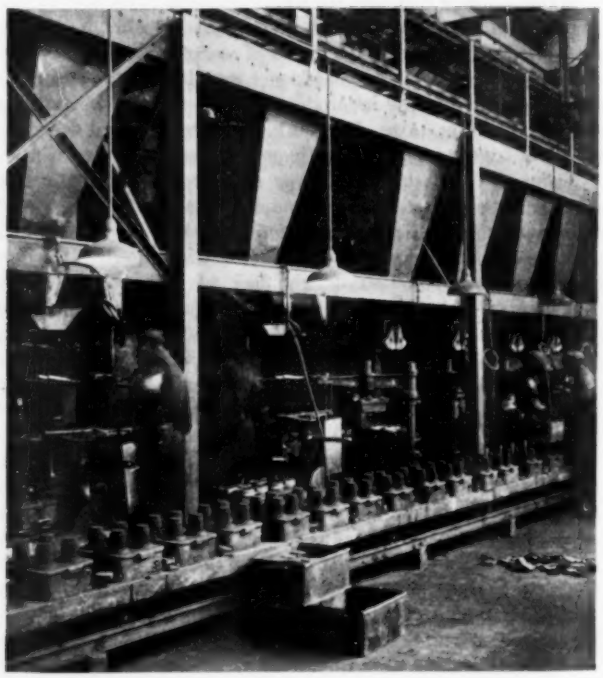


FIG. 5 CONTINUOUS TRAY-TOP "ANTI-FRICTION" CAR-TYPE POWER MOLD CONVEYOR

The most general application of the sliding-type conveyor is for the handling of snap-flask and the smaller tight-flask molds, although quite a few are handling molds weighing 300 to 400 lb.

To eliminate a crane or hoist handling from the molding station to the conveyor for these heavier molds, the cope and drag are built on a stationary section of gravity conveyor placed at right angles to the power-driven mold unit. The tray tops of the power mold conveyor also consist of gravity rolls, which permit rolling the molds on to the conveyor tray from the stationary gravity conveyor as it passes the molding station.

For long conveyors handling molds weighing more than 200 lb., experience and study have shown the advisability and economy of utilizing "anti-friction" roller design.

MODERN ROLLER AND CAR-TYPE DESIGNS

In the last few years several durable and effective designs

of "anti-friction" roller and car-type mold conveyors have come into general use. Figs. 5, 6, and 7 illustrate two such designs used extensively, both of which present a continuous series of tray tops that fully protect the conveyor chain and supporting structure. Fig. 5 is a development of the earlier car type, which can be equipped with "anti-friction" rollers and is so designed

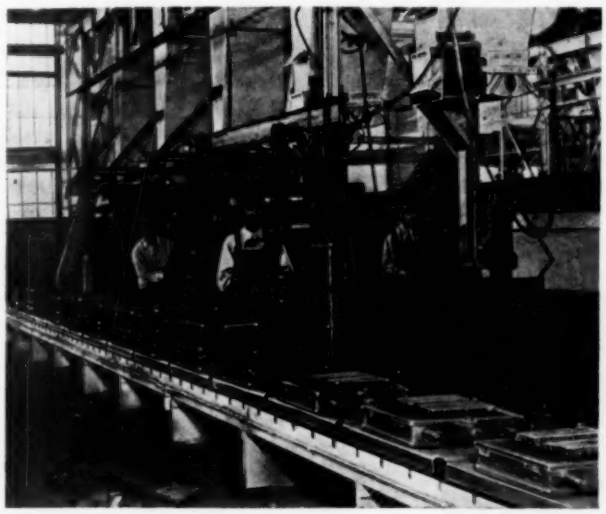


FIG. 6 CONTINUOUS TRAY-TOP "ANTI-FRICTION" ROLLER-TYPE POWER MOLD CONVEYOR

that the trucks will operate around short turns without excessive roller or track wear.

The conveyor illustrated in Fig. 6 is known as the "anti-friction" type because of the liberal use of high-grade tapered roller bearings. The tracks of the conveyor are attached to the under side of the trays, making it impossible for metal or sand to fall on them, as, besides being inverted, they are always under the tray. The chain is likewise protected in its position under the inside edge of the tray flange. The inverted tracks ride on large diameter rollers, having pressure grease-lubricating attachments for the roller bearings on which they revolve. The continuous overlap trays completely cover the rollers, making a continuous moving platform, and thus protecting the rollers from sand and metal.

A number of modified designs of car-type mold conveyors are in use to suit special conditions. One of these consists of trays, fitted with swivel casters and traveling through a rectangular path. Propulsion is provided by air cylinders mounted at the corners giving motion to each horizontal line of trays, the corner turn being accomplished by the nature of the joint between adjacent trays.

Standardization of mold-handling devices has been carried on in so far as it is possible to do so, and for most problems it

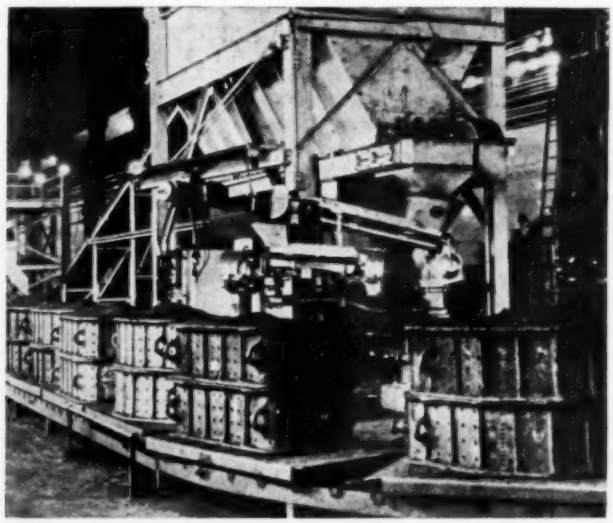


FIG. 7 CONTINUOUS POWER MOLD CONVEYOR HANDLING LARGE MOLDS WEIGHING 5500 LB.

will be found possible to purchase suitable standard conveyors, cranes, hoists, or trucks at reasonable cost.

All too often tremendous expenditures are made for special and expensive equipment when actually standardized designs would prove much more effective and economical.

Discussion

H. L. MCKINNON.[2] The author makes the statement that gravity conveyors have been developed to such an extent that molds of all characters may be handled safely thereon.

It is the writer's judgment that this is scarcely a correct statement inasmuch as we have found that short molds, and especially when stacked, unless they are put on long bottom boards, cannot be safely handled on gravity conveyors. It seems to be almost impossible to get gravity conveyors to handle the molds without high and low rollers, and if the bottom boards are less than covering four rollers, difficulty is usually experienced.

[2] Secretary, C. O. Bartlett & Snow Company, Cleveland, Ohio

MH-53-6

Synchronization of Production Control With Conveyorization

By E. M. OLIN[1] and W. C. BEATTIE,[2] MANSFIELD, OHIO

Mass-production methods are applied to a plant making a variety of things by concentrating for a definite period on one thing, making a supply for distribution over a certain length of time, and then turning the plant's efforts to producing another type of product. This paper deals with the production control of a partly conveyorized plant of the Westinghouse Company where 1500 operators produce certain domestic and commercial heating devices.



E. M. Olin



W. C. Beattie

MASS-production methods, first utilized in a large way in the automotive industry toward the end of the second decade of this century, have revolutionized industrial practices in this country and indeed throughout the world.

These methods have been introduced and are being employed in mining, transportation, the oil and gas industries, agriculture—in fact, in every walk of life.

They have resulted in tremendous economic changes affecting the living conditions and social status of the peoples of the entire civilized world.

The higher standards of living which we have been enjoying during the past ten years must in large measure be due to the reduced costs brought about by mass-production methods. These reduced costs mean that the average output of each individual worker per unit of time has been greatly increased. In some cases this increase is as high as 200 per cent; that is, the same number of workers, with modern facilities, will turn out three times what they did before the introduction of mass-production methods. To the uninitiated this seems almost unbelievable, but it is true nevertheless.

To some of us, at least, these methods have brought a change in our thinking as it applies to almost every problem of our daily lives.

No doubt there has been a certain loss in cultural and esthetic values because of our so-called mechanized existence, but, on the other hand, tremendous wastes have been eliminated which in the long run must result in great benefit to the human race.

Hundreds of magazine and newspaper articles have been written about mass production and its influence on our economic and social life. The authors of these articles for the most part have little or no knowledge of the technique of mass production. They form their opinions without full knowledge of the facts, and in many instances draw erroneous conclusions. For example, nearly all of them lament the disappearance of the old-time, all-around mechanic, the jack-of-all-trades, the artizan who took such joy in his craftsmanship. They deplore the robot who is simply a cog in the machine, doing the same monotonous task hour after hour, day after day on a production assembly line, let us say.

There is plenty of evidence that there is very little foundation for this feeling of sorrow on the part of the uplifters, and that the American workman in a modern factory using mass-production methods is today better housed, better paid, better satisfied than ever before.

While the average output per man in a modern factory may be from two to three times what it was under the former method of operation, this does not mean that the worker himself does in physical labor what two or three men formerly did. On the contrary, due to labor-saving devices and better handling facilities in general, he has it easier now than ever before. The huge savings made are owing to the use of automatic machinery, the elimination of effort due to better handling by means of conveyors, and other mechanical aids to better coordination all along the line.

This paper deals with the production control of a partially conveyorized plant of the Westinghouse Electric and Manufacturing Company at Mansfield, Ohio, where 1500 operators produce the domestic and commercial heating devices marketed by that company, such as electric ranges, flatirons, toasters, grills, bread ovens, and the like. There are other items made, such as industrial heating equipment, icing units for refrigerators, safety switches, etc.—in all some 300 different devices. One of these, the flatiron, runs into hundreds of thousands annually. Others have a yearly production of but 200 to 300 units, or even less. Obviously where the volume is small there would be no economy in using conveyors. Their first cost and installation expense could not be justified. On the other hand, flatirons and electric ranges are made in volume sufficiently great to justify mass-production methods. In fact, any other plan of manufacture would be uneconomical. In a great many instances items of relatively small volume are partially conveyorized as it is often found that considerable savings can be effected by a moderate use of this method.

Several years ago when we first began to study modern mass-production methods in the automotive plants with a view to using this principle or method in our own works, we believed that the conveyor must be the heart of the matter. Of course, none of the equipment which we studied was designed for flatirons or ranges. We therefore had to ask ourselves, "What is the function of the conveyor? What does it do?" Our simplest definition was that a conveyor transports material at a definite

---

[1] Works Manager, Westinghouse Elec. & Mfg. Co. Following his graduation from the electrical engineering course at Purdue University in 1894, Mr. Olin became a student apprentice with the Westinghouse Electric & Mfg. Co. He served as superintendent of commercial testing at the East Pittsburgh Works from 1900 to 1905, and then became design engineer on turbo-generators and rotary converters. He has held his present position since 1917.

[2] Director of Production, Westinghouse Elec. & Mfg. Co. For six years after being graduated from the Mansfield, Ohio, High School, Mr. Beattie was estimator and later manager of a general contracting company. For the past thirteen years he has been with Westinghouse Electric & Mfg. Co., in its works engineering, production-planning, and stores-control departments.

Presented at the "Three-M" (Management, Materials, Maintenance) Congress held in Cleveland, Ohio, April 13 to 17, 1931, under the auspices of the A.S.M.E. and the S.I.E.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

speed over a definite route. Therefore, it provides for three things:

1 Transportation
2 Definite speed or time
3 Definite routing.

The value of any conveyor may depend on any one of these things, or on a combination of any two or all three of these things.

It was to perform the first function that conveyors were first used—mainly for handling bulk materials. Many conveyors are still installed for this purpose.

The bulk-material conveyor has been followed by package conveyors of various kinds. Many of these are designed with very intricate deflecting arrangements which permit selective routing of packaged materials.

About four years ago we installed a complete conveyor system of this nature in a new warehouse. This is primarily a transportation unit, although we have gained a number of benefits from the definite routing feature also.

One of the first conveyors which we installed at our works was a cord-assembly conveyor. We had seen a conveyor of this kind in one of our competitors' plants. We adopted the basic idea and improved on its application. All are familiar with the common appliance cord with attachment plug on each end. For assembling this cord we constructed a conveyor 36 ft. long, having in addition a cutting and stripping table about 10 ft. long, the overall width being 30 in.

The conveyor is equipped with a number of lugs for carrying the cord along and is driven by a variable-speed motor.

Various tools are used at intervals along the length of the conveyor, such as cutting and stripping fixtures, eyelet machines, drill presses, arranged with friction screwdrivers, testing devices, etc.

Six girls work nine hours a day turning out 1500 flatiron cords. Each operator has a particular operation to perform—one cutting and stripping the wires, another putting on the eyelets, still another the attachment plugs, etc. It is in reality a Ford assembly line in miniature. By means of this conveyor we reduced our assembly time from 13 hours per 100 cords to $7^1/_2$ hours, a reduction of 45 per cent. In money, this would amount to a 2-cent saving per cord. We make at Mansfield well over 500,000 cords per year, so here we have a clear saving of $10,000 or better.

This unit was the first of a number of pace-setting conveyors. Now let me say something about this pace-setting feature of a conveyor.

I know of several shops where they are using conveyors for transportation only, ignoring or neglecting the pace-making or time feature. The managers of these plants are overlooking one of the most important features of the conveyor, and one which can be made productive of large returns.

Another type of conveyor in which the definite time element is all-important is the process conveyor, used for painting, baking, drying, vitreous enameling, heat treating, etc. The human element is almost entirely eliminated, and the same results are produced day after day. This results in lower rejects and lower costs.

Even if no direct labor savings are made, the reduction in rejects alone often returns the whole investment in a very short time.

We have at the Mansfield works one conveyor application which is unique and which it is believed is original with us.

A study of a certain department which assembles a considerable variety of the smaller appliances, such as toasters, waffle irons, hot plates, bowl-type radiant heaters, and the like, showed that there was a great similarity in the fundamental assembly operations in each of them. For example, all had wire-wound heating elements of similar design. All of them had frameworks or bodies consisting of sheet-metal parts held together by hollow rivets. All of the devices were electrically tested in about the same way and were afterward packed in cartons of about the same size and design.

Now, while no one of these devices had enough volume to justify a conveyor set-up, we conceived the idea of using one conveyor for all of them and running the different devices consecutively at different times during each monthly production period. For example, we would run toasters at the rate of 1000 a day for ten days, then waffle irons at 500 a day for ten days, then hot plates or bowl-type radiant heaters at 500 a day for ten days, thus gaining the advantage of conveyor economies although we would not have been justified in conveyorizing for any one of the devices because of lack of volume.

Of course this is an entirely special condition with us, and it is mentioned merely to indicate that conveyors can be employed advantageously in many cases which at first glance do not appear to be susceptible to conveyor treatment. We were able in this case to reduce our assembly labor about 35 per cent on each of these devices.

In the foregoing paragraphs we have outlined briefly the manufacturing set-up in our Mansfield plant and have touched upon mass-production methods in a rather superficial way in order to get a background for an understanding of the methods used for production.

Any engineer who has been thrown in intimate contact with manufacturing operations in a factory operating under old-fashioned methods knows that there is a tremendous loss of time due to lack of coordination. Outwardly this may not be apparent. The operators seem busy. There is the whir of machinery. Loads are being trucked to and fro in an endless stream. Clerks are flying about checking materials and operations. There is no loafing, and yet 50 per cent of this activity may be superfluous. One of the greatest advantages of the conveyor treatment in manufacturing operations is that it brings this superfluous activity to light, and forces the correction of the myriad abuses of the former method of haphazard manufacture.

This means that with the conveyor there must be production control of a very high order. Work must flow from operation to operation or from process to process in an orderly way, without delay and in a predetermined order.

When layout engineers plan a conveyorized system of manufacture, each separate operation must be studied minutely and scrutinized as never before.

Not only must each individual operation be time-studied, but the different operations all along the line must be tied together or balanced out in such a way that there will be absolutely no lost motion, and this requires expert planning.

When we installed the place-setting conveyors and began to manufacture according to modern methods, we found immediately that we had to have better production control. We soon found that if we started material at the right time, it would be at the proper place on the assembly line at the proper time. We therefore centered our production control on getting started at the right time. If parts are started too soon, congestion and extra handling result; if started too late, the assembly line is held up. So a production-control system of the highest order must go hand in hand with every conveyor layout.

We use a graphic control plan that starts with a definite graphic plan for each assembly conveyor. From this chart a plan is made for each component part back through the finishing departments, fabricating departments, etc., to the raw-material requirements. Every part is checked monthly, and each day's

performance is indicated on the feeder and assembly charts so that we know at all times exactly where we stand.

The definite routing plan also results in less supervision, as the foreman does not have to spend a large part of his time chasing jobs through the department. This has tended to raise the standard of our foremen, and they have been very active in cost-reduction work, which was not possible when their time was taken up with the daily routine of following work to see that schedules were met.

The reduction in work in process inventories is one of the greatest advantages of a conveyor system, and usually it can be shown that conveyorizing is justified from the money saving thus made, if for no other reason.

Under the old scheme of manufacture it was customary to build parts for assembly anywhere from one to three months ahead of the time of actual assembly. These parts were stored until used. Many extra handlings were required, not to mention the additional space used for stocking. Then, there was always a loss due to bookkeeping errors, and many times, due to changes in schedules and design, the stored parts became obsolete and would eventually go into the scrap pile.

Now, with the conveyor plan, the idea is to keep the material moving all the time. The stocking of parts is reduced to as near zero as possible. "A rapid assembly and a quick turnover" is the slogan of a conveyorized factory.

Since conveyorizing we have been able to reduce our inventories of raw materials and work in process more than 25 per cent, and we would be able to make far greater reductions if we had not so many articles in the line on which the volume is very small, but which must be carried for commercial reasons.

We shall now describe the graphic control plan in a more detailed way.

The first step is an order or schedule to make a specified number of certain complete devices. We shall use electric ranges for the sake of illustration.

Some one must determine the quantities and types to be built and the time deliveries are required. In our company a department known as the stock-control section sets up these schedules. This department works between the sales and manufacturing branches of the organization and has accurate data at all times relating to sales, stocks, and manufacturing schedules.

The delivery date is determined by the time required to secure raw materials, fabricate and finish those materials, and finally assemble them.

It is our practice to provide just enough raw materials to cover this period of time.

Sales are forecast four months in advance, and materials are provided against that forecast depending upon the time required to secure them.

Production planning starts when the stock-control schedule is received.

Graphic load charts are built up for the entire schedule for all types of ranges. These load charts are built up two weeks in advance of the time final assembly is to start and cover a four weeks' assembly period, the first two weeks repeated, the second two weeks new.

Three major factors determine the load-chart values:

1 The commercial requirements
2 The facilities and materials available as reflected by individual tooling for each type of range
3 Operator efficiency.

We find the dates the completed ranges are wanted, the possibility of meeting these dates, and the gross loads on the different manufacturing sections.

These loads are then plotted graphically, using figures and "blue bar" for load and "red bar" for performance against load. Opposite each range under the working day or portion of working day in the month, the bars and figures indicate the time for assembly and later the performance against the load.

Requirements for the complete schedule of ranges are estimated with respect to parts, raw material, expense material, and labor from permanent manufacturing information. These are nothing more or less than bills of material showing parts and multiples thereof, to make one range of each style.

DESCRIPTION	L OR R	QUANTITY	1 2 3 4 5	6	7	8	9 10 11 12	13	14	15	16 17 18 19	20	21	22	23 24 25 26	27	28	29
C 431 – GREY	L	200	105	95														
	R	300		10	105	105	80											
C 631 – GREY	L	250					25	105	105	15								
	R	350								90	105	105	50					
C 831 – GREY	L	100											55	45				
	R	200												60	105	35		
C 841 – GREY	L	100														70	30	
	R	200															75	125
CR 842 – GREY	L	125	20	20	20	20	20	20	5									
	R	175							15	20	20	20	20	20	20	20	20	
CH 631 – GREY		100	DELIVER 13 PER DAY (5TH THROUGH 18TH) BENCH															
CL 621 – GREY		100	DELIVER 13 PER DAY (19TH THROUGH 29TH) BENCH															
TOTAL		2200																

FIG. 1 RANGE CONVEYOR ASSEMBLY FOR JANUARY, 1931
(Quota is 125 per day on conveyor.)

These figures are transposed to another graphic load chart (assembly chart, Fig. 2), which becomes the medium through which replenishing supplies of parts are ordered and followed.

As on the load chart for complete ranges, we express the time each part will be needed for assembly by vertical bars blocking off the days the part is required for assembly. These charts are

STYLE	DESCRIPTION	ROUTE	STORES STOCK	WORK STK PLUS FINISHED PARTS	REQ'D NEXT 4 WEEKS	1 2 3 4 5	6	7	8	9 10 11 12	13	14	15	16 17 18 19	20	21	22	23 24 25 26	27	28	29
513518	DOOR FASTENER	E12-E32		50	600																
590065	YOKE	E23-E32		100	600																
590369	RH SIDE PANEL	E12-LM-E32		25	350																
		LM																			
590369	LH SIDE PANEL	E12-LM-E32		100	280																
		LM	200																		
669456	BODY BACK LINER	E12-E11-E13-E21		0	600																
		E11	30																		
669780	OVEN BRACE	E12-E11-E13-E32		10	600																
		E11	28																		
230647	SPRING	E13-E32-E21		60	600																
		E21	700																		
327797	HANDLE ROD	E13-E21-E31		20	600																
		E31	625																		
522106	TERMINAL	E23-E13-E22		1300	1200																
		E13	0																		
590010	COIL BRKT.	E12-E23-E13-E21		100	600																
		E23-E13	0																		
		E13	600																		

FIG. 2 ASSEMBLY CHART FOR JANUARY, 1931

written on the 14th day of each month and on the last day of each month. Each chart shows the load for four weeks, the first two weeks being a repetition of the previous chart, the second two weeks added.

Space is provided on these charts to tabulate the following values on one line for each part:

Identifying number of part
Description of part
Routing through departments

Available quantities in process
Required quantity
Working days of ensuing assembly period.

A comparison of these values with the vertical bars which show the time the parts are required for assembly indicates the different quantities of parts to be provided and when needed.

Let us emphasize that with fabrication starting only two weeks before assembly, for those parts on which production is small and for specialized equipment, the longest investment turnover is two weeks. Later we shall explain the much more rapid turnover accomplished on that portion of the range schedule where quantities are large, which justifies the specialized facilities which have been provided.

Part orders are written up on still another graphic chart (departmental chart, Fig. 3), which becomes the order to the department to do the work. This chart shows the following values:

Identifying number of the part
Identifying description of the part
Routing through departments
Quantity to be made.

A red dot under the date indicates when delivery must start. This is graphed by extension of the red bar showing performance against requirement.

Perhaps the greatest obstacle to successful conveyorized control is a sudden change-over from one type of device to another to meet some commercial demand.

We can revise our control set-up for the entire plant in 12

DEPARTMENTAL CHART— JANUARY 1931

STYLE	DESCRIPTION	ROUTE	MAKE	1 2 3 5	6	7	8	9 10 11 12	13	14	15	16 17 18 19	20	21	22	23 24 25 26	27	28	29
E 12																			
513518	DOOR FASTNER	E12–E32	600	●															
590369	SIDE PANEL	E12–LH	400			●													
590369	SIDE PANEL	E12–LH	0																
669454	BODY BACK LINER	E12–E11	600		●														
669750	OVEN BRACE	E12–E11	600		●														
590010	COIL BRACKET	E12–E23	0																
E 23																			
590005	YOKE	E23–E32	600	●															
522106	TERMINAL	E23–B13	0																
590010	COIL BRACKET	E23–B13	0																
B 13																			
522106	TERMINAL	B13–E32	0																
590010	COIL BRACKET	B13–E32	600	●															
B.O.																			
250647	SPRING	PURC. ORDER 81950	500					●											
327797	HANDLE ROD	PURC. ORDER 81953	500					●											

FIG. 3 DEPARTMENTAL CHART FOR JANUARY, 1931

hours by means of our chart system, and can produce any type of range wanted in two weeks.

No part production is started more than two weeks before assembly begins, and many parts are made and finished as used.

With part fabricating and finishing synchronized with the assembly requirements, the entire manufacture is in tune and rapid revisions of schedules are simple; also obsolescence of material is avoided.

We shall now explain our method of providing purchased parts and material. Again we have developed a plan which permits rapid turnover and guards against obsolescence.

We advise our suppliers monthly of our expected requirements during the third and fourth months from that date. At the same time we give them an order for the second month's requirements.

Since we have indicated our February requirements on two previous estimates—namely, those of November 1 and December 1—our suppliers have their facilities and material available to furnish February requirements during January just a short time before assembly starts.

Certain classes of material, such as porcelains, castings, and cartons, are handled on weekly, semi-monthly, and monthly schedules.

Cartons, for example, are particularly adaptable to schedule

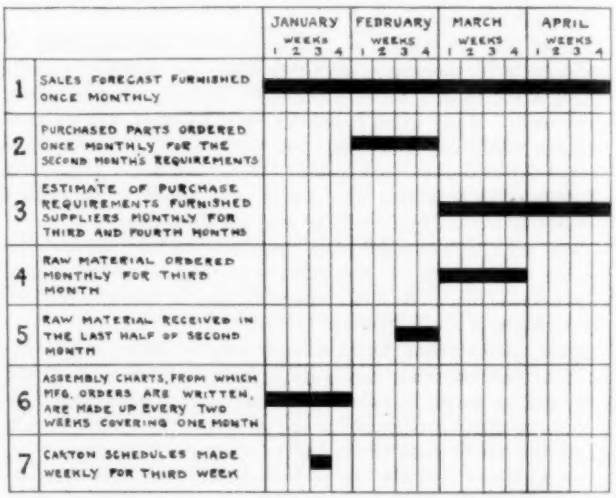


FIG. 4 CHART SHOWING REQUIREMENTS OVER A LONG PERIOD

purchases. However, their variety and changing requirements, together with storage and handling problems, present considerable difficulties.

We have worked out a plan whereby we secure a four days' turnover, and we purchase every carton we use.

On Monday morning of each week we provide our suppliers with a schedule of our third week's requirements. Once monthly we furnish our suppliers an estimate of our next three months' requirements, with which they fortify themselves with board and manufacturing facilities.

The suppliers consume the first and most of the second week to make our cartons and ship to us via motor trucks, so that we receive on Friday or Saturday of each week all requirements for assembly in the next week.

Other purchase schedules are similar.

Raw-material control presents a different and more difficult problem because it frequently happens with us that one kind and size of material is used to make many different items. This is particularly true of steel sheets.

From the breakdown of the complete schedule previously described, we post the four months' advance requirement to a "raw material card" (see Fig. 5), which we use to accumulate requirements and availables of raw material from which replenishing supplies are ordered.

These cards are serially numbered and are headed with the name of the material, purchasing specifications, and ordering size, one for each item of raw material which we purchase.

Each part requiring material on any card is shown with a cross-reference of the card serial number back to the "permanent manufacturing information." Requirements are posted opposite each part for each of the following four months. The available amount in process is also posted on the card.

The number of pieces of each part that can be obtained from one ordering unit of raw material is divided into the requirement, and totals for each month are added and multiplied by the weight or measurement of the ordering unit. The quantity thus obtained is reduced by the available number or amount in process and in raw stock. A replenishing supply is then ordered.

The turnover of raw material accomplished with this method never exceeds four weeks and averages three weeks. When we consider that extra supplies must be carried to permit flexibility of schedules, this turnover is extremely good.

We have talked about the paper control with no mention of the conveyor control. However, the paper control is essential, otherwise the conveyors cannot function. We shall now trace the fabricating and finishing operations.

Our limited schedules do not permit of punch presses and dies specialized for each part we use.

The departmental charts indicate the sequence of operations for the different parts to be made on each press in order that

CARD NO. 68 P.D.S. 3908

DESCRIPTION OF MATERIAL VITREOUS ENAMEL SHEET STEEL

DIMENSIONS .0375" x 27¾" x 81" WEIGHT 23.4 LB.

STYLE NO.	PIECES PER SHEET	WKM'S REQ'M'TS PCS.	WKM'S SHEETS	JANUARY PCS. REQ'D	JANUARY SHEETS REQ'D	FEBRUARY PCS. REQ'D	FEBRUARY SHEETS REQ'D	MARCH PCS. REQ'D	MARCH SHEETS REQ'D	APRIL PCS. REQ'D	APRIL SHEETS REQ'D
521723	7	800	114	1100	157	—	—	800	115	700	100
668073	4	50	12	—	—	175	44	—	—	—	—
668074	4	50	12	—	—	225	56	—	—	—	—
691932	12	500	41	700	59	—	—	1100	92	1000	84
691933	12	700	58	900	75	—	—	1100	92	1000	84
TOTAL SHEETS REQ'D			237		291		100		299		268
WEIGHT PER SHEET			234LB		234LB		234LB		234LB		234LB
TOTAL WEIGHT REQ'D			5435LB		6800LB		2340LB		7000LB		6270LB

DEC. 1780 LB.
JAN. 6800 "
FEB. 1750 "
MAR. 6840 "
17000 "
716 " WPH
16284 LB. 4 MO.

JAN 6800 LB.
FEB. 2340 "
MAR. 7000 "
APR. 6270 "
22410 "
5450 " WPH
16960 LB. 4 MO.

STOCK 7700 LB.
PURCH. ORDER 103985-6000#2/25
PURCH. ORDER 104099-6000#3/15

Fig. 5 Raw-Material Card

delivery dates may be met and assembly continued without interruption, also so that no surplus stock will be accumulated.

Control is accomplished by a master punch-press board which has three pockets for each press (see Fig. 6): first, one for parts to be made (called the load-ahead pocket); second, one for parts in work, and third, one for parts which are delayed. This board indicates the sequence of operations for each part.

A work ticket for each operation for each part to be made in a two weeks' period is filed in the "load ahead" pocket of the planning board. It is a simple matter to file all of these work tickets for each press in the sequence required, as indicated by the delivery dates on the "departmental charts."

Delivery is made via gravity conveyors to stores in standard containers, which have no definite individual locations in the store rooms and are controlled by balance tags. These containers are handled in stores with automatic elevators.

Storage of fabricated parts is ordinarily not necessary, but sometimes becomes so for one of the following four reasons:

1 When the volume required is insufficient to justify specialized equipment to make as used
2 When the volume required is insufficient to justify suppliers of purchased material furnishing as required
3 When impossible to finish as fast as fabricated, because of volume not justifying more finishing equipment; it is impossible also to run the small quantities to match finish facilities, because of set-up costs to manufacture
4 When it becomes impossible to assemble the part or parts as soon as finished; again volume is the determining factor.

When storage of material is necessary, we have eliminated the usual bookkeeping and its attendant evils by means of the following coupon system.

A ticket serially numbered, consisting of three coupons, is attached to each container as the parts are received in stores, showing the identifying style and quantity on each of the three coupons. The bottom coupon is detached and returned to the delivering department and serves as a memorandum for paying the operator.

The second coupon is detached after the section and bin number (one of 2000 for sheet-metal parts) are noted thereon and is filed in identifying number order, the oldest container first in the file. The third coupon remains attached to a piece of material in the container. To find the total quantity of any part available it is only necessary to add all of the quantities shown on the coupons for any one style.

Because of the varying requirements for each part, it is impossible to assign definite bins in which the same kind of part will always be stored.

The location of the parts by the serially numbered coupon-control method is indispensable for conveyorized manufacture.

To obtain the best turnover for parts painted and vitreous enameled, a daily schedule is provided. A one-day bank of such parts is stored on conveyors on the assembly floors.

Each day, parts for the second day's assembly are sent out of stores to the painting or enameling departments by the gravity and mechanical conveyors.

When during finishing or assembling operations defective parts are found which cannot be salvaged at once, additional pieces are started immediately to balance out the run with no shortage or surplus.

Through the burn-off (cleaning) operation, paint, inspection,

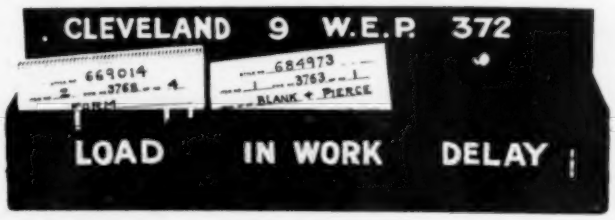


Fig. 6 Section of Planning Board, Machine-Tool Control

and assembly, the parts are handled entirely by conveyors placed by assembly requirements.

Through the vitreous-enamel department the parts are progressively handled by conveyors, delivering from stores through pickling, ground-coat dip or spray, ground-coat burn, finish-coat spray, dry, rim, burn, inspection, second coat (if necessary), and delivery to the assembly department.

Lack of volume and multiplicity of parts require closely coordinated control of nickel, chromium-plated, and lacquered parts. A schedule made daily with conveyorized transportation as in the case of the painted and enameled parts provides the right quantity required for assembly.

A coupon system of control for the stores or handling conveyors which feed the assembly conveyor again eliminates bookkeeping.

As in the case of unfinished-part stores, a coupon is detached upon delivery of the part to the conveyor servicing the assemblers. This is filed in identifying-number sequence. As the container is sent to the assemblers, another coupon is detached and clears the corresponding coupon in the file.

Part assemblies are never stored, but are synchronized on the feeder lines to the final-assembly conveyors, thus eliminating stores. Here again the time element is paramount. These part assemblies are serviced by conveyors to the final-assembly conveyor.

In the case of a range, the bottom of the crate is started first on the assembly conveyor and the proper parts added successively at various stations along the conveyor until the range is completed and ready for final inspection, after which it is crated on the conveyor and conveyed to the warehouse, where it is stored under cranes serviced by conveyors. It is then available for miscellaneous shipment.

In a modern conveyorized factory, the old-time production chaser has been replaced by a much higher type of man.

Production control has become a fine art, one which is worthy of any man's talents. In no other branch of industry is there greater need of resourcefulness and imagination. A mistake or oversight on the part of the production-planning department may wreck a season's output.

Once under way, a production line must be kept going. No price is too great to pay to accomplish this. Truly an efficient production department is one of the most indispensable factors in this great modern development which we call mass production.

MH-53-7

Material Handling in Mass Production at the General Electric Plant

By H. C. RUNDLE,[1] SCHENECTADY, N. Y.

In this paper the author describes the conveyor system used throughout the various buildings where refrigerators are made at the General Electric Company. Before this installation material was handled by tractor-and-trailer trains, lift trucks, and gas and electric trucks. The new system has eliminated congestion, retracing of material routes, delays in schedule, spoilage of material, and maintenance and replacement of the industrial trucks. The conveyor serves as a traveling stockroom, thus releasing floor space for use in manufacturing operations. The conveyor system has linked together four different sections located in 12 separate buildings and maintains a uniform flow of material.



THE General Electric Company started to build electric refrigerators at the Schenectady plant in 1926, at which time it was decided to use the existing buildings that were available. As in most cases, these buildings were not adapted to this type of manufacturing, so it was necessary to fit the equipment to the buildings in order to meet the requirements of refrigerator manufacture. In 1927, the refrigerator department occupied a single three-story building. During the next two years, electric-refrigerator manufacture increased at such a rapid rate that the processing of parts spread into nearby buildings formerly occupied by the switch-gear department. In 1930, the refrigerator-unit department, not including the manufacture of cabinets, occupied 12 buildings spread over a ground area of approximately 10 acres.

During this time the material distribution from building to building throughout the department was handled by means of tractor-and-trailer trains, lift trucks and skids, gas and electric trucks. This method of material handling resulted in congestion on the avenues, courtyards, and in buildings, the retracing of material routes, delays in schedule, spoilage of material due to rough handling and exposure to weather, and maintenance and replacement of equipment. Material-handling costs were excessive. This condition brought up the question of a better method of handling material, and resulted in the installation of an overhead chain conveyor.

To justify the installation of such a system, advantages have been derived which otherwise could never have been accomplished. They include (1) decreased handling cost, (2) reduced spoilage, (3) released floor space, and (4) reduced inventories.

Handling costs have been decreased by a saving in direct labor, the elimination of 624 hr. of porter labor per week, which was formerly necessary for the handling of material throughout the department. In conjunction with this, approximately 90 per cent of the spoilage and waste from the former handling method, due to rough handling and weather exposure, has been eliminated. Also 140 trailers, one tractor, and one electric lift truck were released for use elsewhere in the plant, and the necessary maintenance cost of keeping this equipment in working condition was done away with. By eliminating the use of this equipment, congestion from trucking within the buildings, on the works avenue, and in courtyards has been relieved, all of which helps to improve working conditions and outside appearance.

Because of the increased speed and uniform flow of material and its storage on the conveyor which serves as a traveling stockroom, the buffer storages before and after the lines of processing have been reduced. This releases floor space that can be used for manufacturing purposes and at the same time reduces inventory.

In addition to these, the production department in charge of the material handled by this conveyor can control the work in process by regulating the speed of the chain. By governing the inflow of material to the assembly line, the contributing section can produce no more than what the conveyor can absorb. Thus, the production department can automatically regulate the output of the contributing section to conform with the requirements of the assembly line, maintaining the same production schedule throughout the whole department, with a uniform flow of material, which reduces the inventory, and at the same time eliminate any change of over-production in any one section.

Course and Type of Material Handled

The department is divided into four sections, namely: (1) DR-1 section located in building No. 28 (see Fig. 1), where the DR-1 unit, used on the 4-cu. ft. cabinet, is assembled; (2) DR-2 section located in buildings Nos. 24 and 26, where the D-2 unit, used on the 6-cu. ft. cabinet, is assembled; (3) the water-cooler section located in building No. 12; and (4) the contributing section located in buildings Nos. 8-A and 10, where all punch-pressed parts are processed for the assembly lines of the other three sections. Fig. 1 shows the outline of the conveyor path as it winds its way through these four sections.

Throughout this system there are 12 loading and unloading stations where the various material parts are received from or loaded for other points on the system. To become familiar with the course of the conveyor and the type of material handled, for further reference, the path will be followed through which it travels in one complete cycle. Assuming that the endless chain has a starting point at the drives in building No. 8-A, one can follow the path of the conveyor, in the direction indicated by the arrow, as it travels toward building No. 26. In building No. 8-A, small parts such as check valves and float bulbs, handled by means of small tote boxes, are loaded on the tote-box carriers, routed for building No. 28. Also evaporators are hooked on at this point which is within 3 ft. of the last operation in the line

---

[1] Planning Engineer, General Electric Company. The author, as student engineer to the Chain Belt Company, worked as assistant to the plant engineer during the summer of 1928. He was graduated from Washington State College in 1929 with the Bachelor of Science degree in Mechanical Engineering, and entered the General Electric Company in July, 1929. He is now engaged as planning engineer in the General Planning Department.

Presented at the "Three-M" (Management, Materials, Maintenance) Congress, held in Cleveland, Ohio, April 13 to 17, 1931, under the auspices of the A.S.M.E. and the S.I.E.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

of processing. The style of carriers, hooks, and type of material handled is shown in Fig. 2. After leaving building No. 8-A, the conveyor enters a steam-heated, corrugated-steel covered housing, lined with $^1/_2$-in. celotex, where it rises to an elevation of 30 ft. along the side of building No. 8-A. (See Fig. 2.) From this point it passes over the tops of buildings Nos. 20, 27, and 25 to where it enters the second floor of building No. 26.

In building No. 26 the conveyor travels in a straight line near the ceiling (see Fig. 4) to building No. 24, where the D-2 compressor cases are unloaded during the night shift, and stored

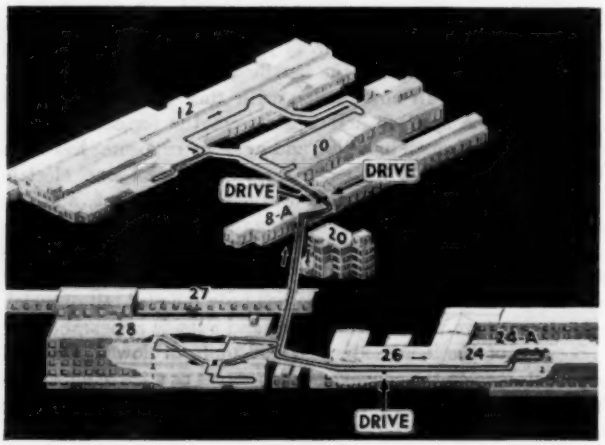


FIG. 1 PATH OF CONVEYOR SYSTEM THROUGH THE DIFFERENT BUILDINGS

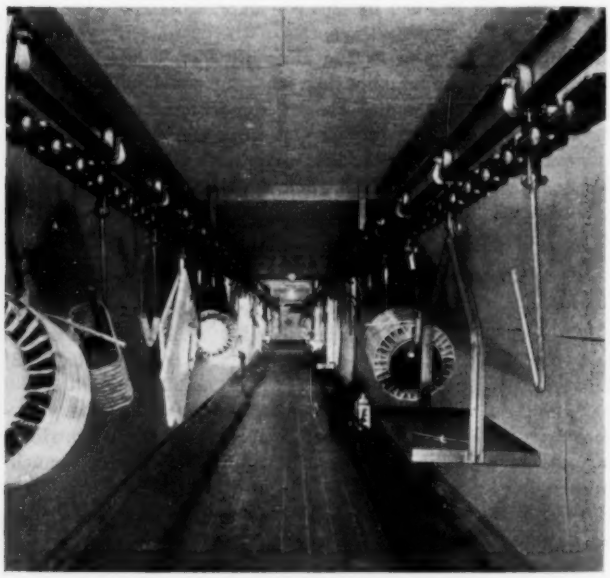


FIG. 2 INTERIOR OF BUILDING 8-A SHOWING INTRADEPARTMENTAL-CONVEYOR HOUSING

for steam cleaning in building No. 24-A during the day shift. Also miscellaneous stock parts from the general stock room, located in building No. 28, first floor, are unloaded at this point and distributed to the various points on the assembly line where needed. From the second floor of building No. 24 the conveyor drops to the first floor of building No. 24-A, where the D-2 compressor cases are unloaded for steam cleaning, after which they are carried to the assembly line by means of a secondary conveyor. From this point the conveyor rises again to the second floor of building No. 24, where it parallels the incoming path to building No. 26, where a second drive is encountered.

In building No. 26, cabinet tops, float valves, and miscellaneous-stock parts are unloaded at the operation on the assembly line where they are used. Leaving the second floor of building No. 26, the conveyor rises to the third floor, where it enters building No. 28. In this building the conveyor dips to a convenient workable height for a length of approximately 30 ft., where DR-1 cabinet tops, float bulbs, and stock parts are unloaded at the operations where they are used on the assembly line. Also the completed DR-1 unit is loaded on the unit carrier (see Fig. 2), routed for the water-cooler section, where it is assembled to the water cooler. Leaving the third floor the conveyor descends in the inclosed housing along the outside of the building, to the first floor, the general stockroom for all four sections, where there are two stations. One is for receiving and loading for distribution all miscellaneous stock parts, and the other is for the transferring

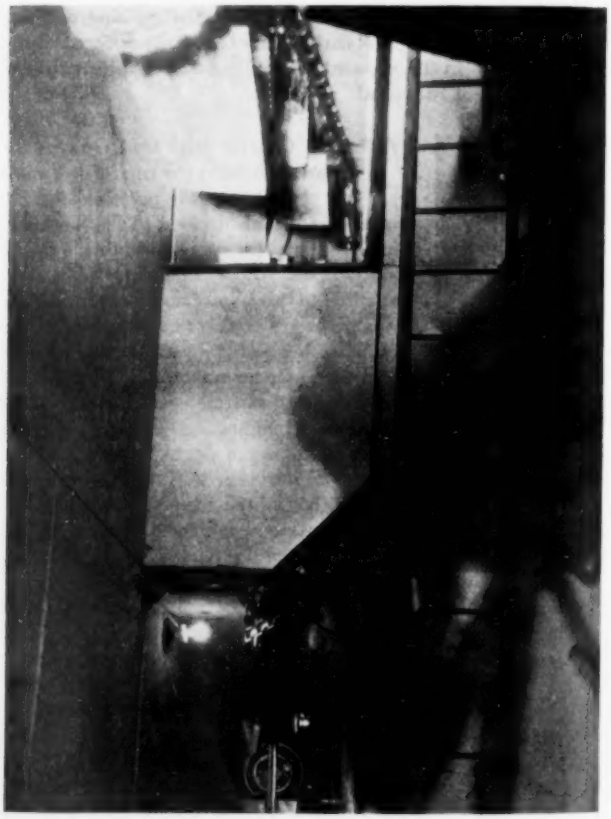


FIG. 3 CONVEYOR AS IT ENTERS THE THIRD FLOOR FROM THE FIRST FLOOR OF BUILDING 28

of evaporators, to a secondary conveyor on which they are carried directly to the sandblast machines in building No. 25. Also the DR-1 compressor cases are unloaded, at this station, for steam cleaning, after which they are carried to the assembly line, on the third floor, by means of a vertical elevator. After serving these two stations, the conveyor leaves building No. 28, where it again enters an enclosed housing, outside of the building, which rises to the third floor, where it connects with the housing of the incoming line. From this point the return line parallels the incoming line (see Fig. 3) to building No. 8-A, where a third drive is encountered. As the conveyor leaves building No. 8-A it enters another housing which it follows along the rear of building No. 10 to where it enters the rear of building No. 12. Here

the DR-1 and D-2 cabinet tops are unhooked from a secondary conveyor, used in the paint line, inspected and then transferred to the main conveyor, for distribution to buildings Nos. 26 and 28. From here the conveyor travels toward the front of the building, where it passes through a subsidiary stockroom, where the stock parts for the assembly of the water cooler are received from the general stock room in building No. 28 and redistributed to the assembly line. In the front end of building No. 12 the DR-1 compressor unit is unloaded at the operation on the assembly line, where it is assembled to the water-cooler cabinet. From building No. 12 the conveyor passes through another housing to where it enters the front end of building No. 10, at which point the DR-1 compressor case is loaded for distribution to building No. 28. From here the conveyor follows the wall, inside the building near the roof, to the rear of the building, where the D-2 compressor case is loaded for distribution to building No. 24. From this point the conveyor leaves building No. 10 and enters building No. 8-A, where it encounters the first drive at the starting point, thus completing one cycle through which the conveyor travels. At no point is this system a part of the assembly lines. Its function is purely one of transportation among the four sections within the department, no operations being performed on the conveyor.

Throughout the course of the entire system just described, approximately 50 different parts are handled daily. These parts range in size from a small wood screw to the compressor unit which is mounted on top of the refrigerator, and in weight from one ounce to 75 lb. On the present production schedule the chain is traveling at a speed of 26.6 ft. per min., completing one cycle in 2 hr. 18 min., carrying approximately 85,000 pieces per day, a total load of approximately 85 tons.

The care with which this material must be handled, involving temperature changes, humidity, dust, grit, and marring and scratching, was carefully considered in the design of this system. Proper clearances and spacing of hooks were maintained to insure against any possibility of marring and scratching of material. The hooks used for carrying the compressor cases are rubber covered to prevent the burring of machined edges when loading or unloading. To prevent the collecting of dust and grit on the material as it passes through the housing located directly above the sand blast building, between buildings Nos. 26 and 28, a dust-proof enclosure was necessary. One of the largest spoilage losses which had to be overcome by this system was that from rust due to weather exposure. In order to maintain the same temperature in the housings as that in the buildings, all of the outside housings, totaling 1000 linear feet, were lined with $\frac{1}{2}$-in. thick celotex and equipped with a steam-heating system. Approximately 90 per cent of the losses which occurred in the former handling method have been eliminated; all of which cuts down the cost of rehandling, reworking, and loss of material.

The outstanding features in the design of this system are the multiple drives and the signal system, which are as follows:

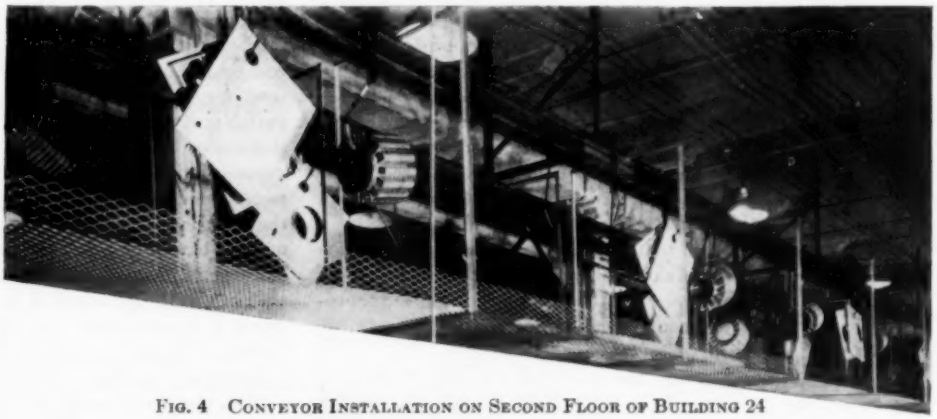


FIG. 4 CONVEYOR INSTALLATION ON SECOND FLOOR OF BUILDING 24

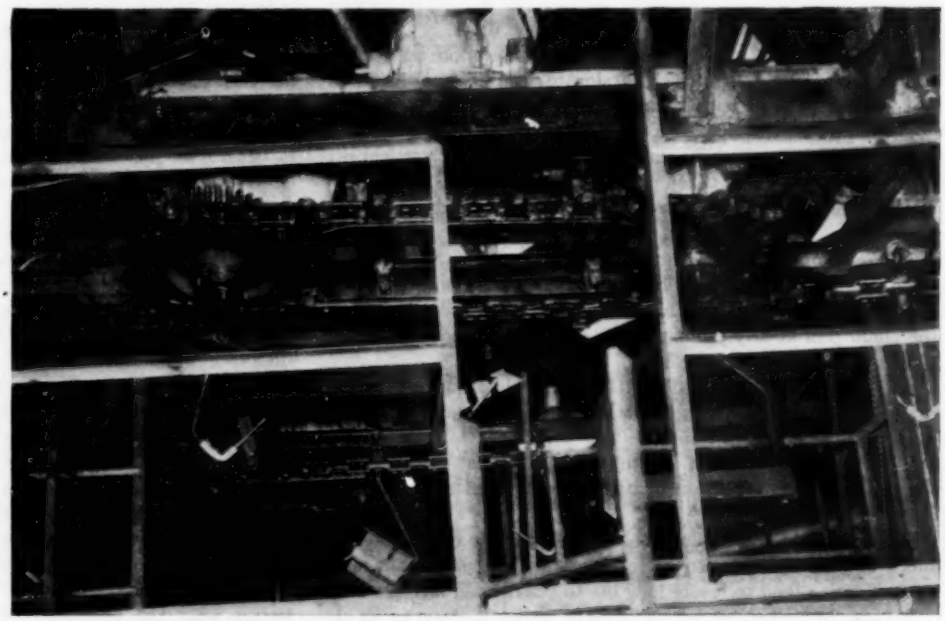


FIG. 5 FULL-FLOATING DUAL CATERPILLAR DRIVES WITH REEVES AUTOMATIC ADJUSTMENT

MULTIPLE DRIVES

The automatically synchronized caterpillar drives used on this overhead chain conveyor installation consist of:

A standard caterpillar drive mounted on a cast-iron base with Reeves speed variator. All are mounted on a structural frame floating on four heavy-duty ball-bearing rollers, with vertical ball-bearing rollers to take the side thrust. A special shaft extension is provided for the Reeves adjustment shaft. A pinion on the end of this shaft engages a rack attached to a fixed support, so that when the drive floats in either direction a resultant turning of the Reeves adjustment shaft occurs. When

the floating drive moves ahead or in the direction of travel of the conveyor chain, the Reeves adjustment shaft turns so as to speed up the drive. Conversely when the floating drive moves back or in a direction opposite to the direction of travel of the conveyor chain, the Reeves adjustment shaft turns so as to slow down the drive.

A limit switch is placed behind the floating drive frame. Should a sudden overload be put on the drive due to fouling of the conveyor chain, the floating frame moves back until contact is made with a limit switch which stops all drives.

The maximum allowable travel of the floating frame is 6 in., but in actual practice the travel is about $1^1/_2$ in. The amount of pull the drive is to take is predetermined and compression springs are placed behind the floating drive frame calibrated to withstand a reaction equal to this pull.

In application of the above principles, let a conveyor be assumed with three drives driven by independent motors of the same nominal speed. (No special winding of motors is required.) Suppose each drive is intended to take $^1/_3$ of a total chain pull of 3000 lb. The three drives are started instantaneously by a common switch, but due to difference in motor windings one of the drives runs slow and does not pick up its share of the load. This drive will move ahead with the conveyor chain, automatically turning the adjustment screw on the Reeves adjustment shaft to speed up the drive.

On the other hand should the drive run too fast, tending to take more than its share of the load (1000 lb. pull), the springs behind the floating drive frame will compress, allowing the drive to move backward, automatically turning the adjustment screw of the Reeves adjustment to slow down the drive.

In using two or more (the conveyor in question has three) drives of this type it is advisable to locate drives so that an equal distribution of load will occur. Elements to be considered in determining distribution of load are (1) length of loaded conveyor pulled by each drive, and (2) uncompensated vertical lift of the loaded conveyor taken care of by each drive.

Advantages of the automatically synchronized drive are as follows:

1 Conveyors of extreme length may be constructed using the same standard chain and trolley equipment as is commonly used for conveyors of short length. This is accomplished by keeping the chain pull at any point within the limits of good design for conveyors of ordinary length.

2 Drives may be placed at the most advantageous points, it not being necessary to group them around a countershaft or other medium of power.

3 Variation in pitch of conveyor chain is not important as the floating type drive automatically overcomes this variation by variation in speed.

4 The compression springs behind the floating drive take the shock of starting and also act as a cushion in case of a wreck, both of which tend to insure long life to the driving mechanism.

5 Limit switches behind each drive make it easy to locate trouble in case of a wreck on the conveyor. In connection with this feature, it is possible to install signal lights at a central location which will tell the operator in charge when a limit switch kicks out, and also in what zone the trouble is to be looked for. This will save considerable delay in getting the conveyor into operation again.

Safety

Safety plays an important role in the design of all conveyor installations, as well as other equipment in the plant of the General Electric Company. All proposed layouts must be approved by the safety engineers before the proposals are submitted to outside companies for propositions. After the conveyor has been installed, a final inspection is made to see that the safety appliances conform with the safety rules of the General Electric Company.

Throughout this entire system, guarding for safety has been carried out to the most minute detail; no expense has been spared. Control switches, guards, and signal horns have been supplied to insure safety to those working in the vicinity through which it travels.

At each loading and unloading station, a safety switch is located directly above the conveyor rail within easy reaching distance from the floor. The switches are of the push-button type, equipped with a locking device, so that when the button is pushed the locking device must be unlocked before the switch makes contact. The switches are so connected that they can only be used for stopping the conveyor. The one and only starting switch is at the control station, located in building No. 8-A, which is operated by an officially appointed operator. In case the conveyor is stopped by a safety switch, the control-station operator must receive word, from the station where the switch was used, that the line is cleared before starting. Also a switch is located on the drive frame of each drive, so that the conveyor can be stopped in case of accident while making speed adjustments, the only work permitted to be done on the drives while the conveyor is in operation. At all places on the system where the material on the conveyor is below 7 ft. from the floor, safety switches are installed within reaching distance.

To prevent any accident which might occur when starting the conveyor, a horn-signaling system was installed whose function is to give warning to those near the conveyor. Each station is equipped with a siren horn, connected to the starting switch, so that for approximately 10 sec. before the conveyor starts the horns throughout the system sound warning that the conveyor is going to start. Also a horn is located near each drive. From a safety standpoint, this announcing system has proved very satisfactory. To eliminate any possibility of a hazard while oiling, each drive has been equipped with a one-shot lubricating system.

One of the difficulties which arose in having such a large number of stop switches, located at points in eight different buildings, was the lost time in locating a locked switch, after the conveyor had been stopped. In order to overcome this a tell-tale lighting system was installed which has reduced lost time to a minimum. It is very important that the system be put into operation as soon as possible after a shutdown, as all three assembly lines depend upon this installation for their source of incoming material. Each stop switch is connected to two lights, hooked in multiple, placed on a panel board, located near the starting station. The stations are numbered in rotation, corresponding to those stations numbered on the panel. When the conveyor is stopped, the operator at the control panel is able to tell instantly which stop button was pushed. He can then communicate, finding out whether or not the line is cleared, so that he can start the conveyor. For example; when the switch at station No. 1 is opened, the two lights No. 1, on the panel board, which are connected to station No. 1, will be switched off, while the remaining lights designated by the numbers corresponding with the station number to which they are connected will remain lighted. The control operator then knows that the switch at station No. 1 stopped the conveyor. This light system is also very effective in stopping practical jokers from meddling with the stop switches.

Metal guarding has been used extensively; the entire length of the conveyor through buildings is guarded with expanded metal and sheet iron. Wherever the conveyor runs overhead, an expanded-metal guard is fastened from the rail so that it hangs below the material being conveyed. (See Fig. 4.) The

guards are made with the metal turned up on the sides to prevent the material from falling in case it becomes disengaged from a hook or carrier. The only points at which the conveyor runs near the floor are at the unloading and loading stations. At these points the track is guarded with an umbrella sheet-iron guard, completely inclosing all moving parts. The driving and idler sprocket and caterpillar chain on the drives are inclosed in a sheet-metal guard. Every precaution has been made to make this system safe.

Scheduling of Material

The scheduling of material plays an important part in the operation of the conveyor. The material movement of parts from the contributing section and DR-1 section must be synchronized with the operations of the assembly line. Miscellaneous stock parts must be scheduled in accordance to production requirements. Rework parts and other material must be scheduled as fast as they are repaired.

The scheduling of stock parts is routed by a numbering system. Those parts which are common to both DR-1 and D-2 units are routed to the different sections in different quantities, which makes it necessary to have some system whereby the person at the station where they are to be unloaded will know which allotment is to be unloaded. By a numbering system, for example, a tote-box container filled with stock parts scheduled for station No. 1 is numbered by means of a clip with a corresponding number, so that the material is conveyed directly to the station where it is unloaded.

The synchronizing of material movement between two sections is maintained by painting the hooks for one section a different color than those used for another section. These colored hooks are spaced so that the inflow of material at the assembly line is at the same rate as the outgoing. To take care of any fluctuation in the assembly line or conveyor delivery, a space has been allowed at each station for a small buffer stock.

The installation of this conveyor system has resulted in linking together four different sections located in 12 separate buildings and in maintaining a uniform flow of material.

Discussion

Jervis B. Webb.[2] For a number of years the problem of driving long conveyors with multiple drives had not been entirely satisfactorily worked out. The Webb Company had worked out problems of this nature from many angles, but a considerable amount of care and attention and adjustment had always been necessary on the methods heretofore used.

One of the first types of multiple drive was used at the Ford Motor Company in Toronto, where three individual drives using the alternating-current type of motors, having as nearly as possible the same characteristics, were connected to one conveyor chain at convenient points, so located as to balance the load as nearly as possible. The system was worked out on the basis that an ordinary a.-c. squirrel-cage motor has a slight variation in full-load and no-load speed—that is, no-load speed would be around 1200 r.p.m. and full-load speed around 1140 r.p.m. As the motors were selected to have practically identical characteristics and the load was fairly evenly balanced, any discrepancy in speed would be taken care of by this slight slippage.

This installation proved to be very satisfactory, but when adopted, in a number of instances, due to the inability to get equal loading and accurate similar characteristics of motors, much difficulty was experienced. One drive was found to take all the load, or even to try to pull through the other drive; and of course this type of system was of constant speed. This system also had its limitations in that, on a slow-speed conveyor, where the reduction in speed from the motor involved a considerable amount of gearing or a large ratio, it did take a tremendous pull on the driven chain to have any appreciable effect upon the lag in the motor, so that on slow-moving jobs the difficulty was even greater. On fast-moving jobs it was easier to get proper distribution of load per drives.

Another form of multiple-drive application is one where all drives are driven from a single motor with possibly a Reeves for varying the speed and the power being transmitted by means of a shaft to several drive units. The first obvious limitation of this is that the drives must be reasonably close together, in order to be able to connect them mechanically. The second objection to it is that in a long conveyor any 1000 pitches of chain will vary in its total length quite a little from any other 1000 ft. of chain in the conveyor. This is due to wear, strain, or general conditions, and it inevitably works out that certain lengths of the chain, which correspond to the same number of pitches in another part of the chain, will be shorter or longer. Inasmuch as the individual drives turn the chain, pitch for pitch, being mechanically connected, when a certain part of the conveyor chain is between two certain drives, it may have a very slack or loose condition of the chain, while this same section of the conveyor, when it has passed by one drive and is now between another set of two drives, may be tight. In long conveyors this slack frequently amounts to several feet. If the slack is taken out at the point where it occurs between two certain drives, when the conveyor is started up and that conveyor chain passes beyond one of those drives, there being so many less pitches in that section than there was before, the system tightens up and breaks the chain.

The only safe solution when such a condition occurs in a conveyor is to break up the main conveyor chain into small units, and redistribute them, so that any 500 or 1000 ft. of it equals any other 500 or 1000 ft. in length. This is a difficult job and requires considerable expense, and has to be done from time to time. Also careful adjustment of the drives has to be made frequently.

Another type of drive for handling this problem is the synchronous-motor type of drive which electrically holds the drives together, much in the same manner as when they are mechanically connected, and much of the same problem is presented. The difference in pitches of chain frequently causes one drive to pull against the other or introduces very heavy initial tensions between drives, due to shortness in pitch. Actual tests with a dynamometer chain-pull measuring device have been made on such drive constructions and have proved this point.

With the floating type of drive, which has been described in the paper, and which we designed to meet this particular problem, when the strain occurs on an individual drive, due to any condition, the drive pulls back against the springs, which slows up its speed and relieves its tension, and as it were "passes the buck" on to the next drive that otherwise would try to take it off. It thus operates to even up the pull on the chain at any one point. One of its most important factors, however, is that the pull is always against springs, which give under shock, particularly in connection with long systems where possibilities of foreign obstacles may occur. Sudden stoppages of a conveyor while in motion put great strains on drives, even though shear pins are present. While coming up against the spring the shock is cushioned. All of this makes for longer life of the system.

Then there is the additional factor of safety of the limit switch which operates when the spring is fully compressed, shutting off the power on all drives. The system described also has the advantage of making it possible to vary the speed of the whole

[2] President, Jervis B. Webb Company, Detroit, Mich. Mem. A.S.M.E.

system. Also, the drive, being of the caterpillar type, permits of placement at practically any advantageous spot on the conveyor system.

In other words, a boost or a pull can be introduced at any desirable spot on a conveyor by the use of such a system. One interesting use of this is on long inclines. One of these floating-booster type of drives is inserted part way up the incline, and the springs are adjusted to take a certain amount of the pull, thus operating to reduce the total strain of the chain by one-half, or whatever amount is desired at this point.

Handling Bulk Materials Mechanically

By WILLIAM W. SAYERS,[1] CHICAGO, ILL.

The paper outlines the general characteristics and fields of applicability of the numerous types of mechanical handling and conveying equipment. It also covers the types commonly used for handling such bulk materials as coal, coke, sand and gravel, crushed stone, lime, cement, phosphate and fertilizer, grain, cottonseed, sugar, earth, foundry sand, etc. No attempt has been made to cover overhead cranes and hoists, crawler and locomotive cranes, and such dock handling equipment as bridge tramways, unloading towers, lift-type car dumpers, etc., all of which are also used for handling bulk materials mechanically.



HOW can the average purchaser intelligently select, from the many types and sizes of conveyors, the one type that is the best and the most economical under the worst conditions of service? The answer is that no two plants are alike and neither are the handling conditions, and that the buyer should seek the expert counsel of the conveyor manufacturer or of a consulting engineer who has had a wide experience in the application of equipment of this character.

The manufacture and satisfactory application of conveyors is a business in which the owners of the business gain valuable engineering experience with the years. The longer they have been in the business and the more applications they have made and the more conditions they have met, with perhaps resulting grief, that much better should they be fitted to engineer and install a conveying system that will be a credit to all concerned, with everything considered, such as efficiency, dependability, and ultimate economy.

Seemingly expensive engineering usually means work by more experienced engineers and a good general design based on a real study of the requirements. This is important because there is often a question at first as to whether, for a specific installation, a bucket elevator is better than a skip hoist or whether the horizontal movement should be by drag or by apron, by belt, or by a screw conveyor, or whether one conveyor can be made to serve the purpose better, within the space available, than separate vertical and horizontal units.

Good engineering will insure rigid and dependable supports to carry the machinery so that it will stay in line and so that it will be safe and accessible for painting and for cleaning and oiling, all of which means less ultimate cost and longer life. There should be paths of travel that are economical, chutes that will not clog, and clearances that are safe.

Good engineering extends to every detail of a conveying system, and especially considers the nature and size of the material to be handled, whether it is sticky or abrasive, and the manner in which the material is to be fed to the system.

The selection of the proper types and combinations of conveyors is an engineering problem that cannot be solved by the use of any set formulas; but data are given farther on which it is hoped will be of some help and guidance to those who are seeking to obtain reliable information on the subject or who have occasion to design, operate, supervise, or maintain equipment of this kind.

The commonly known bulk materials and other materials that some may not realize are mechanically conveyed to the extent indicated are treated under individual sections of the text, such as grain, sugar, etc., in an attempt to give specific information or concrete examples of handling, along with the necessarily more or less general statements made in the section immediately following.

Types of Conveyors and Their Application

Under this heading will be covered (1) belt conveyors, (2) skip hoists, (3) trolley bucket conveyors, (4) screw conveyors, (5) apron conveyors, (6) flight conveyors, (7) drag conveyors, (8) chain and bucket conveyors, and (9) power hoes.

Belt Conveyor. The belt conveyor is especially well suited to the conveying of large quantities of bulk materials, such as coal, sand, ores, clay, coke, chemicals, pulpwood chips, etc., at a low cost per ton, due to its small power consumption and to the large capacity that results from its continuous delivery of material at a relatively high belt speed.

The principal elements are the rubber-covered conveying belt, the troughed multi-roll idlers over which the belt moves in the carrying run, the straight rolls over which the belt returns in the bottom run, the end pulleys, and the driving machinery.

When the material is not to be discharged over the end or when it is to be delivered into bins or bunkers at various points along the conveyor's travel, a traveling tripper is used. This tripper supports the belt and raises it sufficiently for the material to be discharged through the tripper's one-way or two-way chute to the storage or to floor below.

Belt conveyors are installed horizontally or at inclinations up to 18 or 20 deg. therefrom, or both horizontally and inclined, but it should be borne in mind that steep inclines are to be avoided, as the material will start to roll back and the belt-carrying capacity is reduced. Therefore, when the material must be elevated to a considerable height, the belt conveyor may be out of the question due to insufficient space or length being available within which to make the grade, and because of the first cost resulting from the additional length and height of framework needed. Trippers are used only on horizontal sections.

The conveying belt may be from 12 in. up to 60 in. wide, may operate at 300 ft. to 600 ft. per min. (or faster when handling grain), and have its carrying-run supporting idlers spaced 3 ft. to 5 ft. apart, depending upon the capacity required, the size of lumps, and the weight of the material per cubic foot. The return idlers may be 9 to 10 ft. apart.

---

[1] Chief Engineer, Link-Belt Company. Mem. A.S.M.E. Mr. Sayers is a graduate of the University of Illinois, with the degree of B.S. in Mechanical Engineering. His experience includes the apprenticeship course in the shop drafting room of the Chicago & Alton Railroad, Bloomington, Ill.; heating and ventilating draftsman, Treasury Department, Washington, D. C.; heating and ventilating draftsman, D. H. Burnham & Co., architects, Chicago, Ill.; Link-Belt Company from 1903 to date, in the following capacities of draftsman, one year; superintendent of construction, erecting, and operating the various types of equipment manufactured by this company, six years; sales engineer, twelve years; chief engineer, Philadelphia, three years; chief engineer, all Link-Belt plants and subsidiaries for the past five years.

Presented at the "Three-M" (Management, Materials, Maintenance) Congress held in Cleveland, Ohio, April 13 to 17, 1931, under the auspices of the A.S.M.E. and the S.I.E.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

The use of the belt conveyor has advanced very rapidly during the past five years due to the perfection of anti-friction idler rolls, thus prolonging the life of both the rolls and the belt and further reducing the amount of power required per ton of material handled.

Today's idler is equipped with tapered roller bearings and with grease reservoirs that need to be recharged at but very infrequent intervals by a grease gun. Attention to maintenance is reduced to a minimum. Accurately made, well-balanced, well-lubricated, free-turning rolls result in long life of the whole equipment.



FIG. 1 ANTI-FRICTION BELT CONVEYOR HANDLING SAND OR GRAVEL

Grease seals are important, to keep the grease in the reservoir and the dirt and grit out and away from moving parts.

Skip Hoists. In its simplest form, the skip hoist consists of a rectangular bucket running on vertical or inclined tracks, a hoisting cable, and a hoisting machine. It has few moving parts, is easy to operate, and is very dependable.

The skip hoist plays an important part in the handling of materials, and has been used extensively in mining operations for generations. It has always been the standard means for feeding blast furnaces, a service for which it is peculiarly adapted, not only because it delivers accurately measured charges at definite intervals of time, but by reason of its reliability.

The skip hoist is well adapted in general for high lifts and to such uses as these:

(1) Storage of coal in high bins
(2) Elevation of ashes in boiler plants where large clinkers are produced
(3) Handling any large or coarsely broken material, as well as fines
(4) Lifting abrasive or corrosive materials, since only the loader and bucket are in contact with the material handled.

The skip hoist may be classified in three ways: (1) Single bucket with no traveling counterweight to offset the dead weight of bucket; (2) single bucket, counterweighted; and (3) two buckets, one balancing the other and ascending while the other is descending.

The electrical control is either entirely automatic, semi-automatic, or non-automatic, depending on the requirements.

The loading of the bucket or skip may be accomplished by hand shoveling or by dumping material into a hopper fitted with a hand-operated discharge-regulating gate or through an entirely automatic loader attached to the bottom of a storage hopper.

The automatic loader may be of (1) the damming type, the bucket continuing to be hoisted and lowered whether or not there is material in the hopper; (2) the full-bucket-control type, causing the bucket to be hoisted when filled, and to remain inoperative when empty or only partially filled; (3) the weight-of-load-in-bucket type, permitting a certain predetermined weight of material to flow into the bucket; or (4) the measuring type, loading the bucket with a fixed volume. In all cases the ascent of the skip closes the discharge gate of the automatic loader, the loader's counterweights holding it in the closed position until contact of the bucket in its descent has again moved the loader into the automatically discharging position.

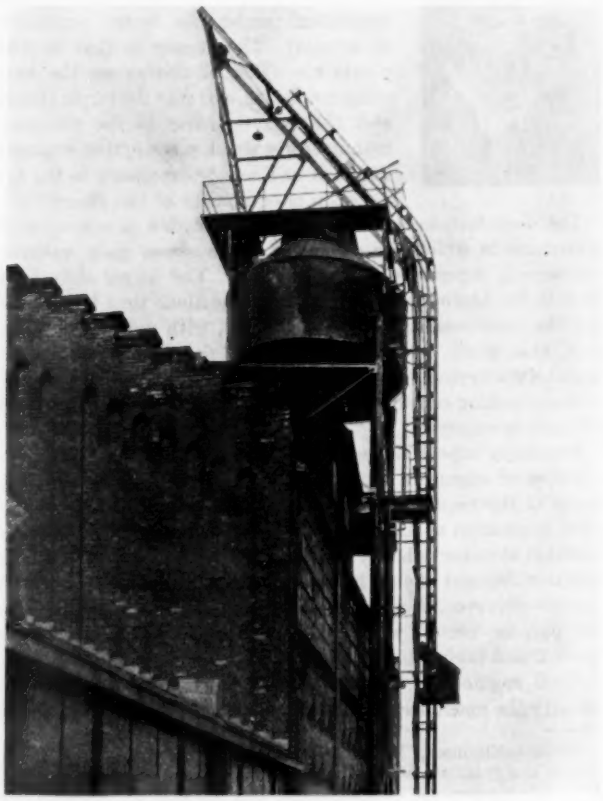


FIG. 2 SKIP HOIST

The machine's electrical control prevents overtravel of the bucket and is arranged to shut off the power when excessive slack has accumulated in the hoisting cable. The bucket may have a holding capacity of 20 cu. ft. up to 150 cu. ft. or more. The normal cable speed may run from 40 ft. to 450 ft. per min. or more, in the handling of such materials as coal, coke, ashes, limestone, gravel, etc.

Trolley Bucket Conveyor. The trolley bucket conveyor is a form of skip hoist, and since it has thus far been used primarily in retail coal yards, it is further discussed under the section, "Retail Coal Yards."

Screw Conveyor. The screw conveyor is used to excellent advantage in the conveying of such small-sized material as grain, cement, cottonseed, and its by-products, pulverized coal, sand, salt, starch, rice, sawdust, sugar, etc.

Screw conveyors are low in first cost, being exceedingly simple in construction and requiring very little head room. The conveyor consists of a spiral mounted on a central shaft or pipe and serves to "screw" or push the material ahead in the all-steel or wooden steel-lined trough in which the spiral is rotated by suitable driving machinery, and from the top of which trough the ends of the 8-, 10-, or 12-ft. sections of standard conveyor are supported at intermediate points by means of hanger bearings.

The diameter of the conveyor or spiral may be as small as 3 in. and as large as 24 in. in diameter. The material is discharged either over the end of the conveyor trough or through gates fitted in the trough's bottom where desired. Sometimes there are simply discharge holes, with no gates, with the result that the coal, for example, discharges through the first opening until the height of pile reaches and closes up the hole. Then the

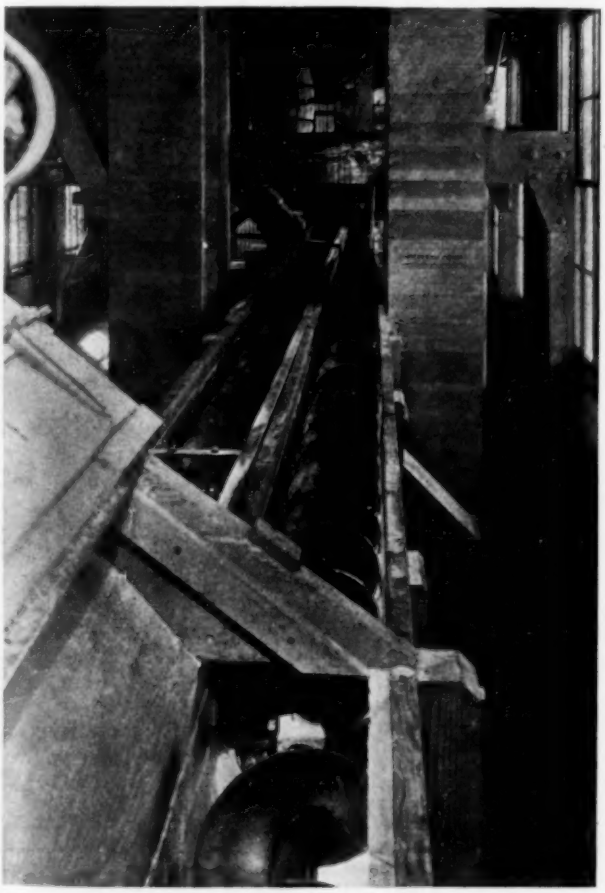


FIG. 3 SCREW CONVEYOR

discharge will be through the next hole in the trough, and so on until the storage is full.

Screw conveyors are used for moving materials horizontally or at slight inclines not exceeding about 10 deg. As a rule their use is confined to the lighter classes of service; and when employed for conveying ashes or other abrasive materials tending to cause rapid wear, the screw and the trough ought to be constructed of cast iron, manganese steel, or heat-treated steel.

The screw conveyor is used quite often at sand- and gravel-washing plants as a dewatering device. The screw has a scrubbing action on the sand and keeps the loam and other foreign matter in suspension, to be carried away in the overflow. Thus a clean and dry sand is secured for use where the specifications and inspection are rigid.

The screw conveyor can often be placed in close quarters where other forms of conveyors could not work.

Apron Conveyors. Apron conveyors for handling bulk materials, such as coal, sand, crushed stone, etc., are usually con-

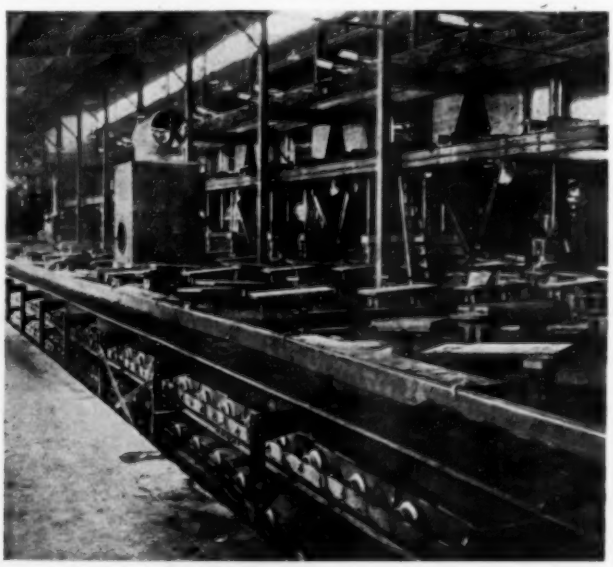


FIG. 4 STEEL APRON CONVEYOR IN THE FOUNDRY

structed with two strands of roller chain traveling on suitable tracks in both the top and bottom runs—overlapping steel pans tying the two chains together and serving as the apron on which the materials are conveyed. Stationary retaining sides or skirt boards are used to permit a greater depth of material to be carried.

Apron conveyors make excellent feeders from coal-track dump hoppers to other conveying media or to a crusher if it is a case of receiving run-of-mine coal and reducing the large lumps in size before the coal is delivered to the main boiler-house conveyor.

Apron conveyors may be installed either horizontally or at inclines up to 26 deg. Owing to their substantial steel construction, the pans will withstand the shock of dumping heavy masses of material on them. As the conveyor operates at a slow speed, usually from 20 to 100 ft. per min., the maintenance cost is low.

In the metal-mining field especially, a very rugged and different form of apron feeder has, of necessity, come into use, as lumps up to 6 ft. in dimension and weighing a ton are sometimes dropped to the pans from a relatively considerable height. To meet such conditions, the pans, the chains, the supporting rollers, and the sprocket wheels, all are made of manganese steel and of a very rugged design. This heavy-duty ore feeder is further discussed under the section, "Ores," in the subdivisions, "Iron Ore" and "Copper Ore."

Flight Conveyors. The flight or pusher type of conveyor was developed as a means for distributing coal and other non-abrasive materials horizontally or elevating at angles from 0 deg. to approximately 45 deg. The material is received in a trough, pushed along by flights attached to the chain at intervals, and usually discharged through openings in the bottom of the trough, being provided with gates to be opened or closed as the conditions

require. In the early days the flight conveyor was made only with a single strand of chain, the flights being fastened to it centrally, and usually made of malleable iron with a thickened scraping edge, as well as with wearing pads for sliding the flights on steel tracks on the return run. In the flight conveyor, the bottom run does the conveying.

The need for greater capacity was met by providing two strands of chain attached to the ends of the flights. A further improvement is the use of rollers suspending the chains and flights, instead of permitting the latter to drag on the bottom of the

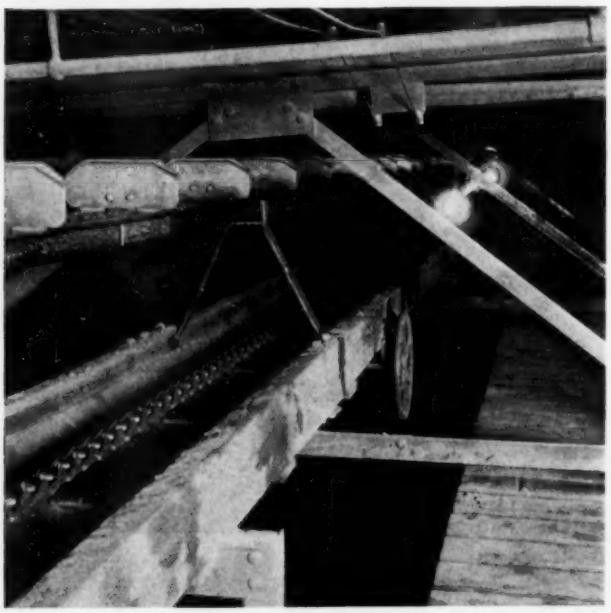


FIG. 5 SINGLE-STRAND SCRAPER FLIGHT CONVEYOR

trough; also the use of double strands of chain with rollers at the articulation points, to carry the chains and flights.

The flight conveyor is a useful type that has been much used in retail coal pockets over the bins; and at the smaller power plants for receiving coal from track dump hopper and stocking it out alongside, and for distributing coal throughout the length of overhead bunkers.

It is used a great deal where the conveying must be done at greater inclinations than 20 deg., the maximum inclination at which a belt conveyor ordinarily can be used. The height of a belt tripper might also preclude the use of a belt conveyor and make the flight conveyor the best suited where the available height is limited and where it is desired to discharge the material at various points.

The flight conveyor is not expensive in first cost, but should be avoided for handling abrasive materials, such as ashes and sand, as the sliding or scraping action rapidly wears the trough lining. The conveyor operates at speeds up to 150 ft. per min., and is suitable for either lumpy or small materials.

Drag Conveyors. Drag chain conveyors are nothing more than one or more strands of chain of suitable type, sliding along in a trough and carrying the material along. Such conveyors are used a great deal for conveying sawdust.

The cast-steel drag chain often used in conveying hot or cold cement clinker, a very abrasive product, has broad wearing surfaces on upper and lower sides. When worn on one side, the chain's useful life can be renewed by turning it over in the trough and using the other side. The head of each link acts as a pusher for conveying the material.

Drag-chain conveyors should be operated slowly to keep the wear to a minimum.

Chain and Bucket Conveyors. Under this caption will be considered the centrifugal-discharge, perfect-discharge, continuous bucket, gravity-discharge, and pivoted bucket carrier types.

Centrifugal-Discharge Bucket Elevator. This usually consists of malleable-iron buckets spaced at intervals on a strand of chain, and operates at a speed of 200 to 300 ft. per min. The material (coal, ashes, etc.) is delivered into a boot, from which it is scooped up by the buckets, elevated, and discharged or thrown by centrifugal force, while passing over the head sprocket, into a chute usually leading to a bin or a distributing conveyor. Although one of the earliest types, it is still an ideal bucket elevator for nominal capacity, for handling material that does not have very large lumps, and where the operation is intermittent.

Perfect-Discharge Bucket Elevator. This uses two strands of chain and has its buckets attached thereto at intervals. This type operates more slowly than the centrifugal type and has a pair of idler sprockets on the return side just below the head wheels, to deflect the chains and completely invert the buckets, thereby effecting a perfect discharge. Because of its slower



FIG. 6 DRAG-CHAIN CONVEYOR CARRYING SAWDUST

pick-up and discharge, the perfect-discharge elevator results in less breakage of friable lumps in the material elevated. When used on an incline, the deflector sprockets may be omitted.

Continuous Bucket Elevator. This is made up with steel buckets fastened continuously and very close together, on one or two strands of chain or on a rubber or canvas belt.

The flanged front of each bucket forms a chute for the discharge from the succeeding bucket, thus effecting a clean discharge at slow speed. This slow speed, with the use of a feeding leg, permits feeding the material directly into the buckets, thus avoid-

ing the wear and strain of digging through an accumulation of crushed stone, ore, coal, or other material of a bulky or gritty character, in an elevator boot, besides reducing breakage of the material elevated and prolonging the elevator's life.

The bucket speed is usually about 100 ft. per min. and up to 200 ft. per min. The continuous bucket elevator is more expensive, but has greater capacity at slower speed than either the centrifugal or the perfect discharge types. It is more expensive because more buckets are required, and the chain and driving machinery must be heavier to carry the greater load and to handle the same quantity in a given time.

The continuous bucket elevator was developed to fill the need for a better elevating unit than the centrifugal-discharge type, and is popularly used in stone quarries for elevating crushed stone into bins.

As the discharge is by gravity and not by centrifugal force, the efficiency or effectiveness of discharge is not particularly dependent upon the speed of the elevator and the speed of discharge.

Gravity-Discharge Bucket Elevator. As the conveying and elevating art progressed to the point where the foregoing types of bucket elevators had been developed for elevating and the flight conveyor for distributing materials horizontally or at inclinations up to 45 deg., engineers appreciated the desirability of a single unit for accomplishing both the elevating and distributing operations. The gravity-discharge elevator was designed and built to accomplish this purpose and has come into wide use.

It consists of steel buckets, approximately V-shaped in cross-section, attached at intervals between two strands of chain. Coal or other material is received in a boot, from which it is scooped up by the buckets in passing around the foot wheels, and is elevated to the desired height, where the buckets turn around wheels from a vertical to a horizontal position. The buckets then push the coal along a steel trough and discharge it through gates or openings to an overhead bunker, storage, or other conveyor, as conditions require.

Frequently, coal is discharged or withdrawn from bunkers or storage into the lower horizontal run of the conveyor, pushed along and picked up by the buckets as their travel changes from horizontal to the vertical direction, then elevated, diverted, and then shoved along the upper horizontal run for distribution to desired points.

An advantage of this type of equipment is that the material can be carried at any inclination, varying from horizontal to vertical, as the buckets will carry and convey the material at angles too steep for flight or belt conveyors.

The buckets may be spaced from 18 to 48 in. apart and vary in size from 12 in. by 12 in. to 48 in. long by 24 in. wide, a normal conveyor speed being 100 ft. per min. Malleable chains are used for the smaller sizes of buckets and heavy steel chains for large capacity installations.

This type is not adapted for handling ashes or abrasive materials, which cause rapid wear of the steel trough linings, but is admirably suited for the handling of hard coal as the pick-up in the boot is slow, and in discharging, the material is allowed to slide gently down a chute, instead of being thrown or dropped. This lessens breakage of the material handled. The machine is flexible in its applications, with special reference to the paths of travel that can be arranged, and it is very reliable in operation.

It is used in retail coal pockets, locomotive coaling stations, power houses, etc.

Pivoted Bucket Carrier. This is so called because each bucket is pivotally suspended between two strands of chain, and maintained in the carrying position by gravity, thus permitting a single carrier to transport material horizontally, vertically, and again horizontally, or in any desired path within a vertical plane of travel.

The advantages of the pivoted bucket carrier may be named briefly, as follows:

(1) The material is carried and the bucket line is supported by rollers of large diameter. Destructive friction and the power required for operation are reduced to a minimum.

(2) Using one machine to both elevate and convey avoids transfers which are often troublesome, take up valuable space, and necessitate deep pits. In addition, only one driving unit is required.



FIG. 7 UPPER HORIZONTAL RUNS OF TWO GRAVITY-DISCHARGE BUCKET-ELEVATOR CONVEYORS

(3) The material handled is automatically discharged on the horizontal runs at any desired point, by a mechanical dumper, and it is only at the dumper that the position of the bucket changes from its original upright position during the entire cycle of operation.

(4) The buckets are overlapping in the horizontal runs, and either right themselves or are righted by a separate device, after having been turned by the mechanical dumper and discharged of their contents. The manner of righting the buckets depends upon the make of the carrier.

(5) Slow speed, quiet operation, with practically no vibration. This results in long life of the conveyor.

Long pitch chains having links made of malleable iron for ordinary installations and of forged steel for very high lifts are used. The chain pitch may be either 18, 24, 30, or 36 in. The buckets have the same pitch as the chain and are from 15 to 36 in. wide, depending upon the capacity required. The chain joint is usually constructed with case-hardened steel pins and bushings and large-diameter enclosed-oiling rollers with chilled or flint-rim treads. The conveyor speed may be 40 ft. per min. and up to 80 ft. per min.

Portable Loaders. Conveyors may be further classified according to whether they are portable or stationary.

The portable loader is very popular for the loading of trucks with bulk materials, such as coal, sand, coke, borings, etc., and is made with either a belt, chain and flight, or chain and bucket elevating medium. Each type has its uses. The belt and flight types are also employed for unloading hopper bottom cars of coal, sand, etc., piling the material alongside the track or delivering it to trucks.

The majority of these portable loaders are hand moved on their

four wheels, while for the larger capacities the moving mechanism is a power-operated crawler of regular crane or tank type. This larger-capacity loader has a platform on one side, on which the operator stands, and from this position he controls the various functions of the machine by the convenient manipulation of hand levers.

The hand-propelled machines are pushed close to pile, and require that the material either be shoveled on the foot end or if the machine has been pushed hard enough into the material, it may merely be necessary to assist the flow of the material with the aid of a shovel. In this latter case no actual shoveling is required. The crawler loader has a self-feeder, and crawls under power as it digs and as it loads.

Power Hoe. The power hoe, or drag scraper, is a simple conveyor which has proved its ability to save time and money in the handling and reclaiming of various bulk materials into and out of storage piles.

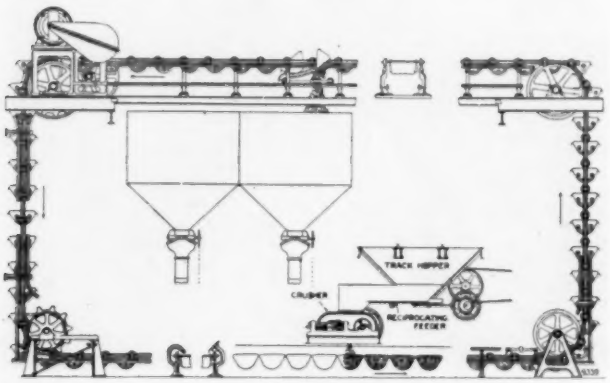


FIG. 8 PRINCIPLE OF OPERATION OF PIVOTED BUCKET CARRIER

The hoe or scoop is operated to and fro by means of a double-drum-friction or geared engine, a wire rope, and properly designed sheaves supported on posts or other form of anchorage. With it one man can store or reclaim most bulk materials at any normal capacity per hour. Used in conjunction with an elevator or loader to trucks or cars, large areas of storage can be reclaimed, and at a very small cost per ton. The system lends itself especially well to irregularly shaped property.

A man in an operator's house operates the engine by hand lever, thus working the full or empty scoop back and forth.

ROTARY RAILROAD-CAR DUMPER

Unloading a flat bottom car of coal or other bulk material by hand shoveling is slow and expensive work at any time, and much more so in winter, when laborious use of a pick is often necessary in breaking up frozen masses.

Unloading hopper-bottom cars through their doors to track hopper is less expensive than unloading flat-bottom cars, but when the car gates are not working properly, as often happens, or when the car is in bad order, and gates have been nailed up to prevent leaking, there is considerable delay, expense, and loss in the normal unloading capacity rate.

A crane with grab bucket may reduce unloading expense and delays, but will require cleaning up by hand, at the corners and sides, and where crossrods are used in hopper cars, they cause much inconvenience and delay. Of course the crane has its application, but under conditions where the capacities are great enough to justify a car dumper, it is obviously more economical to employ a dumper.

Various types of car dumpers have been used successfully for many years on the principal loading docks, both on the Great Lakes and the Atlantic Seaboard, where the rapid loading of vessels is of paramount importance. For the most part these installations have been costly. One reason for this is that in most cases vertical lifting was required. The handling of a large volume of material continuously was necessary, in order to effect economies.

With the development of the rotary dumper the economies of the lifting type of dumper have been brought within the reach of those whose unloading requirements are fairly low.

It turns the car of coal, ore, or stone practically upside down, and the discharge is orderly and immediate, winter or summer, for all types of open-top cars from the smallest to the largest.

The number of cars this dumper can handle per hour or per day depends principally on the switching and track facilities for supplying loaded cars and removing empties. The full cycle of dumping and return to normal position is accomplished in $1^1/_4$ min. The whole operation is so simple, automatic, and foolproof that unskilled labor can perform it readily; it is completely under control and observation at all times, and can be stopped instantly at any point of the cycle.

An interesting marine coal-loading plant is that operated by the Louisville & Nashville Railroad at Pensacola, Fla. While designed primarily to handle cargoes of coal, doing so at a high rate of speed from railroad cars direct to ship without provision for intermediate storage, this dumper and ship-loading elevator have also handled cargoes of other bulk material, such as gravel, stone, and bulk pitch.

The large or super power houses must unload their coal quickly and cheaply. There dare be no delay due to shortage or to freezing of coal in cars. An interesting example is the rotary dumper in use at the Cahokia Station, St. Louis. Placed in service on Nov. 22, 1923, it was taken out of use for its first major repair on July 23, 1928, after having dumped 1,822,360 tons, or approximately 36,450 cars of coal of 50 tons each.

In order to get the 1700 tons of coal per day into the plant for a period of eight days, it was necessary to use the special emergency hoppers for unloading drop-bottom cars. A crew of eight men and the car dumper can handle the 1700 tons per day in one 8-hour shift; but with the dumper out of service for eight days, it required two extra shifts of ten men each, in addition to the regular crew of eight men, working three 8-hour shifts per day to do the same work. Thus we get a good idea of the efficiency and the money- and time-saving characteristics of equipment of this type.

SELF-UNLOADING SHIPS

A number of self-unloading ships are in use on the Great Lakes for transporting and discharging large quantities of raw materials in bulk, such as sand, gravel, cement, limestone, and coal.

The economy of this handling unit lies not only in the efficiency and dependability of its conveying and discharging equipment, but also in the fact that these vessels usually carry a full cargo going and returning, and their use makes it unnecessary for the receiver of the cargo to invest in expensive dock unloading equipment, such as a grab-bucket unloading tower or bridge tramway, since the vessel itself contains the necessary unloading machinery. The cargo may be limestone from the Northwest to the steel-mill district, and coal on the return trip.

The records on one of these vessels show that in the year 1930 its average unloading rate was 1374 tons of stone per hour and 964 tons of coal, this including all the delays and being figured practically from the time the vessel tied up at the dock until it cast off.

Each side of the ship is equipped with a steel pan conveyor under the cargo holds, with hand-operated slide gates at suitable intervals. The desired gates are opened wide, and beyond seeing

that the cargo is flowing to the conveyors, the handling system requires no further attention once it has been started. The two pan conveyors operate slowly, and up an incline at one end, sufficiently steep to discharge the material sidewise to a centrally located continuous bucket elevator, which in turn delivers to a belt conveyor mounted on the vessel's hinged, power-swung unloading boom. This boom conveyor delivers the material direct to the customer's dock or yard, the boom usually being swung out over the side of the boat in time to start unloading the ship's cargo the moment the vessel gets to the shore. A maximum of only about 200 hp. is required for driving all conveyors.

The discharge rate of another of these vessels is said to be in excess of 2000 tons of stone per hour. Belt conveyors and power-operated regulating gates are used under the cargo holds. To keep the belts fully loaded and attain the large handling capacity mentioned, it is necessary to handle the gates constantly.

The apron conveyor moves slowly, carries a deep load, and serves as its own feeder. The belt conveyor operates fast, carries a thin stream, and requires regulation of the feed of material to it.

Quite a number of old boats have been converted into self-unloaders by building two or more tunnels into the bottom of the regular hold, and operating a drag scraper or power hoe back and forth in each tunnel. The scraper may deliver through a hopper, to an inclined belt conveyor delivering to the belt conveyor on the boom.

Self-unloading ships can serve economically the docks which, through lack of adequate and fast enough unloading facilities, have been unable to attract the large ships with low freight rates.

The self-unloading ship keeps its rates down by carrying cargoes going and returning, and by its ability to receive and discharge cargoes at intermediate points or points beyond the regular course that a vessel, not self-unloaded, would be obliged to follow in order to take advantage of the use of dock unloaders.

Terminal Handling of Grain

Today's modern terminal grain elevator, with holding capacities up to 11,600,000 bushels of grain, is equipped with machinery of large capacity to insure rapid handling. There are elevator legs handling as much as 25,000 bushels an hour and scale hoppers weighing 2500 bushels of grain at a time.

At the smaller grain elevators, box cars are unloaded by power shovels into a receiving hopper feeding the elevator.

The large elevators employ the automatic tilting type of grain-car unloader, which is capable of unloading any size car of wheat, oats, or corn in 6 or 7 minutes.

This machine, besides doing the unloading much faster and at a greatly reduced labor and operating cost, effects a further saving by automatically removing the grain doors without destroying them, so that they can be used over and over again, thus avoiding the usual delay for this operation and the expense of repairing damaged doors.

As one unloader will do the work of seven hand pits, there is a substantial saving, due to the use of the former requiring less property, a less expensive trackage layout, a smaller track shed, and a simpler and less costly work house.

The loaded grain car is spotted on the platform of the unloader, with the outer door of the car open on the dumping side. As the disappearing end clamps are brought into position in contact with the car couplers, the car is automatically centered, and two platform locks are disengaged to permit tipping the car sidewise.

The door opener automatically removes the grain door as the car is tipped sidewise 15 deg. The platform is then tipped endwise to a maximum angle of 40 deg. in both directions, and the car load is discharged into a suitable receiving hopper.

After the car has been emptied, the unloader is returned to its normal position, the empty car is removed, and another car is brought on for unloading.

From the track receiving hopper the grain is fed either direct or over a short belt to the boot of the elevator leg, which consists of a centrifugal-discharge belt-and-bucket elevator traveling at a high speed—from 400 to 800 ft. per min., depending upon the type of buckets and the size of pulleys used.

The elevator leg discharges into a garner, or large concrete storage hopper, equipped with a number of gates for rapid dis-

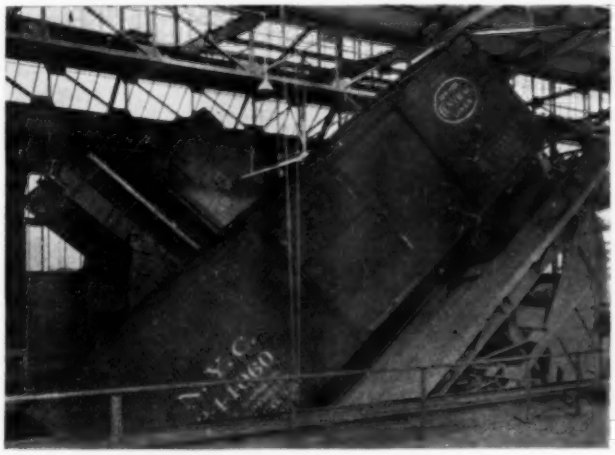


Fig. 9 Grain-Car Unloader

charge into the scale hopper. A car can be unloaded and its contents passed to the scale. Then while it is being weighed and discharged, another carload can be unloaded and elevated to the garner.

If the grain is to be cleaned, it is spouted to the headhouse bin and from there to the cleaner, or direct to cleaner from the scale hopper. After cleaning, it is elevated, weighed, and passed to storage. Grain is delivered to storage by means of a gallery belt conveyor, with a traveling tripper, which runs the length of the tanks.

Grain-handling belt conveyors operate with the belt flat or very nearly so; i.e., there is very little troughing of the belt. A flat belt has a good contact with the horizontal idler pulleys and will run straight, empty, or loaded, or even loaded to one side of the center. Grain carefully fed to a flat belt will not tend to shift toward the edges and spill, but to prevent scattering at the loading points and to increase the carrying capacity moderately, it is customary to use inclined concentrator pulleys at the loading points only, or at 10- or 12-ft. intervals along the length of the conveyor.[2]

In modern practice, wheat and corn are carried at speeds up to 800 ft. per min. on belts 36 in. and wider, but on belts less than 20 in. wide the speed should be less than 500 ft. per min.[3] The main reason for this is that in chutes suited to wide belts the flow of grain is rapid and regular, and the belt can be loaded full, but in narrow chutes the flow is not so fast and even, and the narrow belt running faster than 500 ft. per min. will not take a full load nor will the loading be uniform along the run of the belt. Oats, bran, and flour have to be conveyed more slowly to keep from being blown off the belt by air resistance.

The capacities of grain belts with concentrator pulleys at intervals are given by three manufacturers, in bushels per hour, at 100

[2] Hetzel's book, "Belt Conveyors and Belt Elevators," p. 9.
[3] Ibid., p. 155.

ft. per min., as 1.75 W^2, 2.2 W^2, and 2.5 W^2, in which W is the belt width in inches.

Grain is drawn out on a lower belt conveyor, elevated, weighed, and then spouted to cars. Generally, the large houses load in on one side and reload on the other for shipment. The car-loading capacity of most elevators today is approximately 25,000 bu. per hour per spout. Two spouts are generally used.

Often grain is received with a high moisture content. To keep this grain from getting hot or burned, it must be turned over, so the grain is drawn from bin, handled on the lower conveyor, elevated, and discharged back to the gallery conveyor for delivery to storage. Frequently an extra leg, known as the "turning leg," is installed to handle this work, so that the process of loading and unloading will not be disturbed.

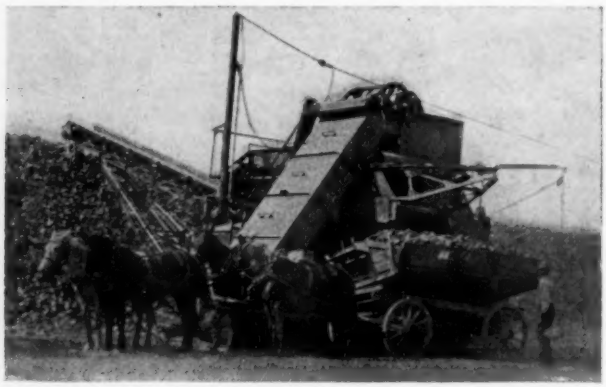


FIG. 10 PORTABLE BEET PILER

When grain is unloaded from boats to elevators, the "marine leg" is employed. This is mounted adjustably so that the foot of the elevator can always reach the grain to be unloaded.

The screw conveyor also is well adapted to the handling of grain, and finds many ideal uses in grain elevators of all sizes.

The 1,000,000-bu. capacity wooden-construction grain elevator of 25 years ago, with its maximum continuous unloading capacity of approximately two cars per hour per unloading pit, with three men and one bucket elevator per pit, required a ground area of approximately 20,000 sq. ft., as compared with only about 15,000 sq. ft. needed for today's fireproof concrete-construction elevator of similar holding capacity, unloading eight cars per hour per unloader, with two men to operate the unloader, a third man to attend to moving of the cars, and the use of only one bucket elevator.

The old house of 1,000,000-bu. capacity usually had at least three tracks, with three but sometimes four pits per track. This compares with two tracks (one for unloading and one for shipping) required for a modern elevator of similar capacity, equipped with an automatic unloader and one unloading pit.

HANDLING SUGAR BEETS AND SUGAR CANE

The chief sources of sugar are sugar cane and the sugar beet. The sugar beet is a native of various sections of the United States. Cane is grown most extensively in the West Indies, Hawaiian Islands, Philippines, State of Louisiana, and now to a limited extent in Florida. First, let us consider the beet.

Sugar beets are dug out of the ground before cold weather sets in, the tops are chopped off with a heavy knife, and the beets are thrown in rows paralleling the furrows. They are then forked into wagons or trucks and are transported to a conveniently located beet dump. The beet-slicing season generally lasts 90 to 120 days, and on account of insufficient storage space at the factory, farmers have to unload a good portion of the crop alongside or near the dump.

A caterpillar-mounted portable beet piler and reloader has been developed, consisting of a hoist for dumping trucks or wagons, a hopper for receiving the beets, a very unusual type of conveyor, a rotating-disk grizzly screen, a stacking belt conveyor for delivering the beets to the pile, and a dirt conveyor for loading the dirt back to the farmer's truck. The use of this machine has resulted in the forming of larger storage piles at the dumps than ever before. The reloading into cars is done by means of a grab bucket which operates over the boom that supports the stacking belt conveyor, this conveyor being removed when it is desired to recover the piles.

The advantages claimed for this machine are that there are no elevated structures to drive upon (in other words, a ground-level dump), large capacity, no bruising of beets, good screening and dirt return, higher piling, compactness, portability without loss of time, and minimum number of operators.

One of the latest factories receives the beets through a flume direct from railroad dump hopper. Beets grown nearby and received in trucks are stored in piles by a portable stacking unit as already described, but regardless of whether factory yards store the beets by this method, by a track dump hopper and belt conveyor system, or from parallel elevated railroad trestles, they are stored over concrete flumes covered with boards which are removed when a supply of beets is to be washed into the factory for use.

The beets have to pass through a trash catcher, consisting of two strands of chain with sawtooth bars hanging down into the flume. From there they go over a stone catcher into a beet wheel, consisting of a large-diameter vertical plate with buckets on its outer edge dipping into the flume, to pick up the beets and dump them into a screw paddle washer, called a "beet washer." Now the beets pass over a roller-type screen into the beet elevator, the newest type of which employs two strands of drop-forged rivetless chain with gravity-discharge buckets fastened thereto at intervals by swiveling attachments.

The elevator discharges to a double-strand-chain wood-slat picking-table conveyor. Here any rotten beets, leaves, or refuse remaining are picked off by hand. From the end of the picking table the beets drop through a weighing device and to the knives which slice them into cossettes, or maybe they are sliced first and then weighed.

The sliced beets are carried either by flat belt conveyors or fingered-type flight-conveyors to the diffusion tanks, where the sugar is dissolved under pressure.

The beet pulp, at a number of factories, is pumped to a big outside pond and fed to livestock. Other factories now make it into a pressed pulp. This pressed pulp is handled by screw conveyors and centrifugal-discharge bucket elevators.

The wet sugar from the centrifugals drops into a ribbon conveyor which delivers to a centrifugal-discharge belt and bucket elevator. The sugar then passes to a granulator or rotary-kiln type of drier with lifting vanes to a dry-sugar bucket elevator, to a screen for taking out lumps, to bins, and to the automatic weighing and sacking machines. Portable bag pilers and conveyors are used for piling the filled bags in the warehouse.

As much sugar has hardened in the bags, necessitating reprocessing, the newer factory handles the finished sugar from the screens by screw conveyors either to the sacking bins or to the boot of an elevator which delivers the sugar into a distributing screw conveyor system over immense concrete silos in which the loose sugar is stored, and from which it is withdrawn by screw conveyors underneath the silos to an elevator delivering to the sacking bins.

Let us now see how sugar cane is handled in Cuba, for instance.

The cane is chopped down by hand and stripped of leaves with a long, broad knife, and then loaded into ox carts, many of which are now equipped with caterpillar treads, or tractors. The cane is hauled to the mill if nearby or to a conveniently located bridge-mounted cane hoist and transfer which lifts the entire load after slings have been passed around it, trolleys it over the railroad track alongside, and dumps the cane into side-dump cane cars, usually of about 30 tons capacity, for transportation in train to the mill.

At the mill a side-tilting hydraulically operated dumper serves to discharge the cars to a large hopper over the foot end of a slow-moving feeder carrier of the apron type, which has to carry a tremendous load of criss-crossed cane stalks, sometimes 8 ft. deep. The feeder carrier delivers the cane to the inclined cane conductor, which feeds it to the initial crushing rolls. Cane kickers or levelers are used in some mills, near the head end of the feeder carrier, to help level the cane and provide a more uniform feed to the cane conductor. The leveler shaft has arms fastened to it in staggered positions, extends across the width of the carrier, and is rotated at a slow speed. Power-operated revolving knives, also used in some mills, are very effective in preparing the cane for more effective crushing. There is no appreciable loss of juice due to this cutting operation.

The cane carriers are as wide as the mills, which measure up to 7 ft. wide. The sugar juice is squeezed out by passing the cane through a series of mills, perhaps consisting of one initial two-roll crusher and six three-roll mills in one tandem, totaling 20 rolls.

The cane is carried from one mill to another on intermediate carriers, which are mostly of the steel-apron type. Sometimes a carrier is used consisting of two strands of chain with angle-iron cross flights for pushing the cane along a juice-tight trough over into the next mill.

From underneath the mills a combined juice straining and trash elevator, of specially constructed chain and flight type, scrapes any trash in the juice up an incline and disposes of it through a screw conveyor to the blanket of crushed cane on one of the intermediate carriers. The juice drains back through perforated trough linings and is delivered to the juice tanks by pipes or canals.

The bagasse, or what is left of the cane after the juice has been crushed out, is carried by double-strand chain and flight conveyor to the boiler room where, through rack-and-pinion type slide discharge gates in the conveyor trough and through specially designed feeders the bagasse is fed to the furnaces and burned as fuel. More recent developments, however, are resulting in the use of bagasse as a by-product for the manufacture of cellulose and fiber board.

The juice and sugar are handled through processes similar to those used in the production of beet sugar, excepting that another form of conveyor, called the grasshopper conveyor, named after its hopping, reciprocating motion, is frequently used in the cane-sugar field, under the centrifugals. It has a steel trough (flexibly supported on a series of hickory sticks) along which the operation of its drive serves to jiggle the sugar without crushing or grinding the crystals. It has large capacity, the trough is self-cleaning, and there is no danger of wrecking the conveyor by broken paddles falling into the trough from the centrifugals.

In the warehouse the bags of raw sugar, weighing 325 lb., are usually handled by portable conveyors and stacked to a height of not over 25 ft. by means of self-propelled portable bag pilers of the double-boom type, the discharge height of the upper boom being adjustable by power.

Much of the bagged raw sugar is refined in plants located in the United States. Here more mechanical handling equipment is used.

Methods of Handling Cottonseed

In the South and Southwest the cottonseed-crushing industry is one of the largest fields for the use of bulk handling equipment; in fact, there are far more cottonseed-oil mills than the North has cement, lime, and gypsum plants.

Bulk cottonseed is received largely in box cars from cotton gins, although all mills have facilities for receiving shipments by truck and some of the smaller plants receive all seed by truck, or wagon.

The modern mill may employ a double automatic power shovel for unloading the seed from box cars and discharging it on an inclined belt to the elevator shaft. The belt may operate at a speed of about 450 ft. per min. over anti-friction troughing-type idlers in the carrying run. Along the unloading track a collector screw receives seed which is "broken out" as the car doors are opened, but the bulk of the unloading is done by the shovel at the unloading hopper. A screw conveyor collects the seed received by truck or wagon and feeds it by one of the track screws to the track-unloading hopper.

The inclined unloading belt discharges into a large double-row bucket elevator in the elevator shaft. This elevator in the particular mill in mind has two rows of 12-in. by $6^1/_2$-in. seamless steel buckets bolted staggeredly to a canvas elevating belt, which is driven by a 40-hp. motor. This motor also drives a 20-in. wide canvas, anti-friction troughing-type distributing belt conveyor at a speed of about 500 ft. per min., through the conveyor's tail pulley. A light steel-frame self-propelled cotton-seed tripper, with three-way discharge spout, distributes the seed in the seed storehouse.

One feature of these plants is that all motors are kept out of the storage house proper. For this reason, distributing belts are driven from the foot pulley, and provided at the far end with long-travel horizontal gravity take-ups, weighted sufficiently to accommodate the belt pull and to take up any belt stretch due to wear, tear, and moisture.

The unloading and distributing equipment in the average fair-sized mill is designed to handle at least 50 tons of bulk cottonseed per hour, weight 20 lb. per cu. ft. It is essential that the unloading equipment be dependable and of adequate capacity, since practically all the seed is delivered to storage by the source of supply between the months of September and December.

The larger unloading elevators have belt speeds of around 400 to 450 ft. per min., with head pulleys 36 in. to 42 in. in diameter. The casings in the newer plants are mostly of steel and of single-leg construction. The elevating speed is slower than that of grain elevators, as cottonseed is lighter, bulkier, and not so free-flowing.

The seed unloaded contains quite a little trash and sand, and it is for this reason that belt elevators and belt conveyors are preferred, although screw conveyors are used in plants of smaller capacity or those with comparatively short distributing conveyors.

The seed storage houses as a rule have capacities of from 5000 to 15,000 tons and are built with fire walls dividing the main body of storage into two, three, or more compartments. The distributing belt conveyor and tripper operate through special firedoors which are arranged to accommodate the conveying equipment in the monitor.

For reclaiming seed from storage, an A-shaped structural-steel tunnel is usually provided down the center of the house. On the floor of this tunnel a drag belt conveyor, of 10-in. or 12-in. wide canvas, operates at a speed of around 500 ft. per min. in a 4-in. or 5-in. deep steel trough to which the seed is fed through doors in the tunnel structure, the belt dragging in the loaded run and returning usually over anti-friction flat rollers. The capacities in this operation are less, usually not more than 5 to 10 tons per hour, in line with the capacity of the mill to lint, separate, and crush the seed.

The reclaiming drag belt delivers to a second elevator provided in the tower for discharging the seed to the proper point in the seed-cleaning room or the mill. Here the trash, sand, and bolls are removed from the seed by rotary reels, and by a secondary screening operation over flat, inclined shaker screens, thus assuring longer life of the linter saws and the knives in the processing equipment through which the seed is later passed.

Some of the more modern mills preclean the seed, meaning that they clean it before storing instead of after entering the storehouse—a practice that probably will grow, for it is safer to store clean seed in large volume than wet, trashy seed, which often may generate heat and catch fire. The mill operator also protects himself by cleaning the seed on arrival, since it is possible to weigh the seed as it is unloaded and cleaned, after trash and sand have been screened out. Some seed is very dirty.

The cleaned seed passes on to the linter room in screw conveyors. Here the first-cut and second-cut linters remove practically all cotton lint from the seed, the lint being handled pneumatically to a baling room, while the seed passes through the linters to more screw conveyors, and then to elevators to the hulling and separating room.

In special processing machines the seed is now slit open, and by a series of shaking trays the hulls are separated from the meat or nuts. Here some inclined drag elevators are used.

Screw conveyors carry the hulls to a separate storage building, for use as an ingredient in cattle food. A small feed-mixing plant usually adjoins the hulls-storage building.

The cottonseed meats pass to the roll and press room. The rolls grind the meats to a form suitable for cooking and compressing. From here the meal is elevated and conveyed to the cookers, comprising a series of three or four steam-jacketed kettles, and serving to bring the meal to the temperature required for properly separating the oil from the body of the meal.

The cooked meal passes from the bottom of the lowest kettle to a hydraulic former, which shapes the material into cakes about 12 in. wide by $1^1/_4$ in. deep by 18 in. long, to fit the press boxes. When all the compartments of the presses have been filled, pressure is exerted, and as the box frames are contracted, the oil is extracted—the oil running down the edges of the inclined back side into a large storage tank. Continuous oil expellers have come into use, but the box presses still seem to be the more popular.

After the oil is extracted by the press, the residue of the meal (a hard cake) is broken up in a cake breaker and elevated to a distributing screw conveyor over storage bins. This conveyor is provided with screen discharge chutes, and a rough separation of several sizes of cracked cake is made in the bin compartments.

This cake also is used as an ingredient in the mixed feeds for cattle and as sheep food. For grinding the cake an attrition mill is usually provided in the meal room, together with an elevator and a batching and sacking unit.

For the mill operations screw conveyors and bucket elevators serve very nicely, since the hourly capacity as governed by the presses is not large as a rule.

In the lint and separating rooms vertical screw conveyors have recently been applied with considerable success. However, in the seed-unloading layout bucket elevators will undoubtedly continue to be favored because the capacities are necessarily rather large.

Phosphate and Fertilizer

Phosphate forms a large percentage of the bulk of most prepared fertilizer and most of it comes from Florida. These prehistoric animal remains exist there in vast areas, being of the variety known as pebble phosphate.

It is mined hydraulically, the formations, consisting of sand, clay, and phosphate, being broken down by powerful water jets. The material so broken down is sluiced to a sump, from which it is pumped by centrifugal pumps to an inclined sluiceway 3 or 4 ft. wide and about 40 ft. long, the bottom of which is a plate with longitudinal slots about $^1/_2$ in. long by $^3/_{64}$ in. wide throughout its area. Sand, clay, phosphate, and dirty water pass through the slots to a second sump. The material carried over goes to a revolving scalper screen provided with holes of approximately $1^1/_2$ in. diameter. The grass, lumps of clay, and rock are discharged at the end of the screen and are rejected.



Fig. 11 Portable Bucket Loader Handling Super-Phosphate to Motor-Driven Buggy

The material which has passed through the screens now goes to a log washer, consisting of two cylindrical logs side by side, with projections, and revolving upwardly in the center. The log washer is inclined similarly to a sand washer and breaks up and disintegrates the clay balls which have passed through the scalper screen, the dirty overflow water discharging to the second sump, previously mentioned.

The wet material from the log washer is discharged to a vibrating screen, where fresh water is added. The water with fine material goes to the second sump, while the material passing over the screen goes to a second log washer where the washing process is repeated. The material discharged from the second log washer goes to a finishing vibrating screen of 20 mesh. The material passing over this screen is the commercial phosphate, which is now loaded on small cars and generally run over a trestle about 20 ft. above the ground and dumped to storage below.

There are generally two tunnels under the trestle, containing belt conveyors for reclaiming the stored phosphate and carrying it to the drying plant. The material is dried, and later ground to a fineness about equal to that of cement.

Today's modern plant reclaims enough of the fine phosphate from the clay and dirty water that pass to the second sump from all washing and screening operations to justify the installation of the added equipment—a classifier and another centrifugal pump.

To make the ground phosphate rock more quickly available as a plant food, it is treated with sulphuric acid, forming what is known as super-phosphate. This treatment, at the fertilizer plant, produces a material which if kneaded or rubbed becomes the consistency of putty. Therefore conveyors for its handling are used

as little as possible, the material being preferably carried in cars or carts in bulk.

The various ingredients composing the finished fertilizer, with the exception of the super-phosphate, are stored in bins. The super-phosphate generally is stored in the center of the building in a high pile, and this, as well as the other materials used in lesser amounts, are all trucked by buggies or wheelbarrows to the mixing point. Here the proper proportions are weighed out, and the batch, consisting of the various materials, is dumped to a bucket elevator which discharges to a vibrating screen. The material going through the screen discharges through a hopper and gate to a batch mixer. The lumps in the mixture which do not go through the screen are bypassed back to the elevator and are broken down in this way.

A small-capacity, hand-propelled portable bucket loader, specially constructed to meet the conditions, is used very successfully in certain sections of the large fertilizer plants, as well as in the smaller dry-mixing plants, for loading super-phosphate and other fertilizer materials into buggies which are either hand-pushed or motor-propelled. Its use saves from five to seven laborers; besides, the loading is done faster.

Portable bag pilers and conveyors are used for storing and for handling the filled bags to cars.

Power shovels are employed for unloading the raw materials from the box cars in which they are received.

Sand and Gravel

The handling of sand and gravel from pit to cars usually follows a certain general flow sheet, and consists of an excavator and some means of getting the material to a washing and sizing plant. The material can then be loaded direct into cars or trucks from the bin or delivered to storage for future loading. The type of excavator and conveying equipment is usually determined by the nature of the deposit or by the shape or contour of the property.

For a small plant, handling up to 50 or 60 tons per hour, with a fairly deep deposit either above or below water, or both, a slack-line cableway excavator is sometimes used. This machine consists of a track cable which can be raised, lowered, and shifted. One end of the track cable is anchored at ground level, and the other is attached to a high mast. The mast is usually high enough to permit the bucket to dump into a hopper over the washing screens. The track cable is slacked, and the bucket, carried on a trolley, runs down by gravity to the digging point. The drag line is then pulled forward, loading the bucket. When the bucket is filled, the track line is raised, and the bucket is pulled up along the track cable to the dumping point. This machine serves as both excavator and conveyor.

Where deposits are below water, or both above and below water, and capacities are over 100 tons per hour, a pump may be used to dig the material. If the distance to the plant and the lift to the top of the plant are not too great, the pump may be used to force the material through a pipe line to the top of the washing plant. It then acts as both excavator and conveyor. Where the lift is too great for economical pumping (perhaps over 30 ft.), the pipe line may be carried on a horizontal line at ground level to a sump at the foot of the plant.

A special digging elevator may then be used to dig the material out of the sump and elevate it to the washing screens.

When the deposit is located quite some distance from the washing plant, the sump may be located at a convenient point and a comparatively short digging elevator used, provided with perforated buckets. This special bucket elevator would dig the material from the sump and drain out the water through the perforations in the buckets. This elevator would then discharge on a belt conveyor leading to the top of the washing plant.

In cases where material is all above water, a gasoline or steam shovel is used to dig the material and place it in dump cars. A locomotive then hauls the cars to a track hopper near the plant. From the hopper the material is discharged by a feeder to a belt conveyor or bucket elevator. Usually the first conveyor discharges the material to a scalping screen which removes the oversize stone.

This large stone is then crushed to proper size and mixed with the other material, and is carried by belt conveyor or bucket elevator to the top of the washing plant.

Whether a belt conveyor or bucket elevator should be employed depends on the lay of the ground and the space available. In most cases where sufficient space is available, the belt conveyor is preferable for handling sand and gravel. It is simple and economical and requires a minimum of attention.

In some deposits with material both above and below water, a boom drag line is used to load the material into cars for transfer to the washing and sizing plant.

In some cases where space is limited, a skip hoist has been installed to take material from a track hopper and elevate it to the screens. The use of the skip hoist is also desirable when gravel contains many boulders above 12 or 14 in. in diameter.

In handling screened and washed material from plant to storage, several systems are used, depending on the amount of storage required and on the shape and location of the space available.

For a moderate amount of storage, the product may be drawn from the bins into railroad cars and handled along the track to a convenient point. Here a crawler or locomotive crane will unload the car into stock piles along the track, as well as reclaim the material from piles to cars or trucks.

Where large storage is needed, a belt conveyor system may be used. A conveyor from the bins is carried on a trestle 50 or 60 ft. high and arranged with a tripper or shuttle conveyor to discharge at any desired point along the trestle.

This stock pile may then be recovered and loaded with a crane. In other cases a tunnel is located under the conveyor trestle, the material being drawn through gates to a belt conveyor in the tunnel. This belt conveys and elevates the material to a loading point along the railroad track or into truck loading bins.

There is no single type of handling equipment that is best or suitable for every condition. Each installation should be studied as an individual problem and a type of conveyor chosen to meet the requirements at that particular location.

Foundry Sand Handling and Preparation

It has been computed that foundries have to handle as much as 150 to 200 tons of material per ton of castings produced. This includes pig iron, coke, limestone, scrap, molding sand, core sand, flasks, patterns, cores, bottom boards, gaggers, the hot metal, sprues, castings to the tumblers, sand to be shaken out and reconditioned, and the returning of the flasks and patterns to follow the same cycle over again if it be a foundry producing a repetitive product, such as automobile parts, sanitary ware, etc.

This industry consequently has great need for mechanical-handling equipment, and there is no denying that within the past 10 years it has made big strides in the economical, production-expediting application of such devices, especially for the handling, preparation, and reconditioning of the sand, and for the handling of molds on slow-moving horizontal tray conveyors to permit continuous molding and pouring.

Instantaneous delivery of prepared sand from overhead hoppers located along the molding zone of the mold conveyor, and pouring "on the move" along the metal-pouring zone, are doing much to speed production methodically and efficiently and to keep the handling costs down to a minimum. Give a molder his conditioned sand ready to drop into a flask, a molding machine, a

mechanical metal carrier, and a shake-out grate, and he can produce for himself in wages and for his employer in profits much more than any molder can by the old shovel and muscle method, provided the right type of machinery has been installed and in the right way for the purpose.

It is true that the foundry sand-handling machinery originally did not accomplish what was expected of it, but time and added experience bring many changes and today the foundryman can get a system for handling his sand just as satisfactorily as for the handling of other materials.

Proper sand conditioning is essential to economical foundry operation. Today mechanical sand conditioners deliver a uniformity and excellence of sand unheard of in the day of the sprinkling can and shovel.

Bucket elevators, belt conveyors, flight conveyors, apron conveyors, rotary screens, paddle mixers, sand-revivifying machines, vibrating shake-out screens—all have found their place in the foundry in the process of handling and preparing molding sand and in conditioning it for re-use.

It should be borne in mind, however, that not all foundries lend themselves to the application of conveying equipment or to the introduction of such complete handling systems. For example, a "jobbing foundry" may profitably employ a simple sand-handling unit, but have no use for mold conveyors. The matter of the producing capacity required or desired, nature of the work, whether many or only a few molds are to be made from the same pattern at a time, and other conditions all enter into determining whether a foundry should install mechanical-handling equipment and also how much or how little.

Crushed Stone, Lime, Cement, Gypsum

An enormous tonnage of crushed stone is quarried annually for the manufacture of chemicals, cement, lime, gypsum, steel, etc., and the building of highways and concrete structures. One plant alone, located in Michigan, has a producing capacity of 10,000,000 tons of crushed limestone per season of seven to eight months.

Normally, about a third of this plant's output is flux stone. Gigantic dipper shovels load the rock into 30- and 40-yd. side-dump air-operated steel cars, which a steam locomotive hauls to a 60-in. gyratory primary crusher in 15-car trains.

The crushed rock is conveyed to the top of the screen house on a 54-in. wide belt conveyor, 700 ft. between centers, which probably employs the heaviest and thickest belt ever installed anywhere. The belt weighs approximately 30 tons, and is about $1^1/_2$ in. thick, being of 14 by 16 step-ply construction, using 42-oz. duck, with $^{13}/_{32}$ in. rubber cover. Anti-friction tapered-roller-bearing heavy-duty belt idlers are used. The drive is of the dual-motor type using 250-hp. motors. This conveyor and the primary crusher are installed in duplicate.

The belt conveyor discharges to a large roll grizzly at the top of screen house, the oversize going to roll crushers and the "throughs" to roll grizzlies below, followed by another set of two roll grizzlies below that, spray-jet washing having started on this third set of grizzlies. At this point the flow of stone is divided to 12 vibrating screens, and continues on down over three more sets of 12 vibrating screens, making 48 of these screens on the four levels.

Open-hearth, flux, and commercial sizes of stone are carried off on belt conveyors at the several floors, and are conveyed and then discharged over belt trippers to their respective storage piles. Belt conveyors operating in tunnels underneath these storage piles serve to reclaim any size of stone that may be wanted. These conveyors deliver to other belt conveyors, and through this medium it is possible to load a 10,000-ton capacity vessel in about five hours or less.

This Michigan plant employs about 21 miles of belt conveyors of widths ranging from 30 to 54 in.

A 50-ton-per-hour stone crushing plant for making, say, four sizes of stone besides dust, may require nothing more than the crusher, a cylindrical rotary screen with dust jacket, and a continuous bucket elevator, with its buckets carried either on chain or a flat belt, for feeding the crushed stone to the screen.

On a 100-ton-capacity plant several crushers may be required, with an apron feeder delivering the quarry-run stone uniformly from a dump-car receiving hopper to the primary crusher, a rotary scalping screen between primary and secondary crushers, a continuous bucket elevator or two, a rotary finishing screen with dust jacket, and a single deck vibrating screen to further "size" the material passing through the meshes of the dust jacket.

The multiple revolving roll grizzly is being widely used now instead of the cylinder type of revolving screen. When the material to be elevated is of large size, such as the product of a primary jaw crusher, an all-steel pan or apron conveyor may be preferable to the use of a bucket elevator.

Each plant is an individual engineering study in the matter of determining the proper type and size of crushing, handling, and screening equipment for accomplishing the desired results—depending to a great extent upon whether quarry cars are loaded by hand or by dipper shovel and, if the latter, on the size of stone the dipper will conveniently handle.

At a modern chemical lime-producing plant we find a dipper shovel reclaiming the raw limestone from storage pile to industrial cars hauled by gasoline engine to the feed hopper of a fully automatic skip hoist which employs a 60 cu. ft. skip bucket, elevates the $2^1/_2$ tons of stone 130 ft. to two 400-ton storage tanks at a speed of 180 ft. per min., making a complete trip every two minutes and delivering 75 to 80 tons an hour.

From these tanks the stone is fed by belt conveyor over a weighing device to a rotary kiln delivering to a cooling kiln, from which belt conveyors receive the lime and feed it to a continuous-bucket elevator discharging to a vibrating screen which makes three separations. The tailings, or pebble lime, and the screenings are fed to their respective bins through screw conveyors. Steel-encased centrifugal-discharge chain and bucket elevators and more screw conveyors form important links in the mechanical handling system from here on through the hydrate building.

A portable-belt-type box-car loader loads pebble lime into box cars which are "spotted" on the plant siding by an electric capstan-type car spotter.

At many of the Eastern lime plants we find the hot lime being drawn from a series of vertical kilns on to steel pan conveyors. The lime may be delivered to a cooling bin by a car system and reclaimed from underneath by a steel-pan conveyor delivering over a shaking screen, the "fines" being taken by a belt conveyor to the hydrating plant and the lumps delivered over a second pan conveyor to a pivotally mounted adjustable-boom conveyor-type box-car loader which, with the aid of an attendant, serves to fill both ends of car to the desired height.

The core may be picked from the second pan conveyor by hand.

The arrangement and type of handling equipment employed necessarily differ according to the local plant conditions and the nature of the lime.

At the cement mill the mechanical handling processes will vary somewhat according to whether cement is being produced by the dry or the wet process and whether limestone or marl is being used as a raw material.

An apron conveyor usually feeds the rock from dump hopper to crusher. From the crusher the material is elevated to the mill

storage bins or the marl to a washing plant by an inclined belt conveyor. Continuous steel bucket elevators, when employed, must be of rugged design and construction to withstand this service. The belt conveyor should be used wherever space permits.

The product of the ball mills, Hercules mills, and other granulators can be conveyed very readily by means of screw conveyors or belt conveyors.

From the cooler the cement clinker may be handled by a pivoted bucket carrier, a belt conveyor, or a cast-steel drag-chain conveyor, or perhaps the hot clinker falls into a pit from which an overhead grab bucket crane takes it to outside storage, returning with a bucketful of seasoned clinker for grinding.

The finished cement may be delivered to silo storage bins by screw conveyors, by a belt conveyor, or by bucket elevator. Screw conveyors are ideally suited for conveying finished cement, and usually handle it from underneath storage bins to packing house. Finished-cement conveyors and elevators should be fully housed and made dust-tight.

In the wet-process plant the limestone and other raw materials are drawn from storage bins and fed to the grinders of the various types, with a sufficient quantity of water to make a thin slurry when the mixture is ground.

Before cement clinker is ground, the proper amount of gypsum is added to regulate the setting time of the finished cement. Gypsum lends itself to mechanical conveying in about the same way as the limestone or marl and the shale or clay.

The kilns may be fired by oil, gas, or coal, and when the fuel is coal, this coal must be crushed, dried, and pulverized. The finely pulverized coal is delivered to storage bins by bucket elevator, and later fed into kilns with the proper amount of air pressure.

Coal Mining and Preparation

Where a seam of coal is close to the surface, the overburden is removed by dipper shovels and the coal is mined from an open pit. When the seam is a considerable distance below ground level, the coal may be hoisted to the surface from a shaft. For example, a certain Illinois bituminous mine employs a balanced-bucket skip hoist, using buckets of 9½ tons' capacity each and automatic loaders to fill them. The coal is hoisted a distance of 525 ft. at the rate of 1400 tons an hour, or 11,200 tons per day of 8 hours, the buckets, in portions of their travel, attaining a speed of 3000 ft. per min.

In other mines the coal is brought to the surface on conveyors, either apron, flight, or belt conveyors being used. This is an economical system in cases where the depths below surface are not very great.

The Colonial Mine of the H. C. Frick Coke Company employs 20 belt conveyors placed end to end, running 500 ft. per min. and carrying 10,000 tons of coal per 8 hours a distance of 4½ miles to the surface.

Their Palmer Dock Mine employs 11 belt conveyors placed end to end, also operating at 500 ft. per min. and handling 10,000 tons of coal or more per 8 hours a distance of 3½ miles from mine-car dump in the mine up to the tipple bin for loading into barges. Particularly impressive is the noiseless conveying of such a large tonnage, with so little degradation and manual attendance. Special attention was of course given to the design of the chutes for transferring the coal from one belt conveyor to the next.

Where mine openings are on mountainsides, the mine cars are brought to the surface, hauled to a power-driven dumper, and dumped into a hopper from which the run-of-mine coal is fed to a retarding conveyor for lowering it to the screening and preparation plant. Apron, belt, and flight types of retarding conveyors are in use, including the rope-and-button style of flight conveyor.

Down in the mine one will see conveyors used as loaders for carrying coal from the faces to pit cars.

Certain types of caterpillar or track-mounted loading machines are provided with gathering mechanisms which pull coal, shot down at the faces, on to inclined conveyors lifting it into cars for transportation to the tipple. A loader of this type, with a crew of three men, will handle 200 tons per 8 hours, whereas the best miners will average no more than 10 tons per man. And while the miner, besides loading 10 tons, also does other work, such as drilling coal, laying track, and timbering, the loader-equipped mine shows a big saving, even though a separate crew is hired to perform the supplementary work just mentioned.

In other mines, inexpensive conveyors, mounted on light flanged wheels and without coal-gathering mechanism, are used. Their handling capacity is limited, since the miners must shovel the coal on the lower ends of these units. These pit-car loaders do have the advantage that rock and bone can be removed by hand before loading and can be left in the mine, whereas this is difficult of accomplishment with loaders of the digging or self-gathering type. These smaller machines can be used where roof conditions are bad and the timbers close together. Like the self-gathering loader, they are of the flight-conveyor type, being made up with two parallel strands of roller chain connected by bars at intervals, to serve as the conveying medium for dragging the coal along the conveyor trough.

Where coal is mined from long faces, one also will find "face conveyors" in use for carrying the coal shoveled into them to pit cars at loading points which may be several hundred feet away. Thus the conveyor is made to serve as an efficient mine-transportation unit. Apron conveyors, mat conveyors, and belt-and flight-type conveyors have all been used, but the drag-chain conveyor has been found to be the simplest and most satisfactory. This consists simply of a rugged chain, with wings at intervals, sliding in a trough.

Shaking or jigging conveyors also are used in the mine. These consist of steel troughs, roller-supported, to which a quick return stroke is imparted by the drive units.

The idea was imported from Europe, but American inventive genius improved it by developing a shovel end, which is pushed into the "shot" coal and serves to load the shaking conveyor. The shovel has a telescoping end to permit its being advanced as the coal is loaded. These conveyors are made in 10-ft. to 13-ft. sections, the shovel end having an extension equal to the length of one section of conveyor. When the conveyor has been extended its full length, an additional section of trough is inserted, and the advance is again begun.

Let us now pass to the preparation plant, tipple, or washery, the function of which is to clean and properly size the coal for the market. Mechanical loading in the mine, while a step in the right direction, has on the other hand served to increase the amount of impurities to be taken out of the coal. This means that in some plants additional equipment is needed for adequate preparation.

The sizing is done over screens, usually of the shaking and vibrating types. The cleaning is accomplished by either of three methods, the method that should be selected depending upon the nature of the coal and its impurities. The three methods are:

(1) Manual inspection and picking on picking-table conveyors.

(2) Separation of free impurities by dry-cleaning mechanical methods.

(3) Wet concentration, or "the washery process."

The washing process is the most effective method of separating the impurities or refuse from the coal. The difference in specific gravity of coal and refuse is utilized to effect a separation in the

water, by means of pulsations which lift the coal into the upper current, while the refuse drops on the sieves and passes out through refuse gates.

The cleaned and sized coal may be loaded into open-top railroad cars by means of a pan-type loading-boom conveyor, the discharge end of which is adjustable by an overhead hoist arrangement, to permit placing the coal in the car gently, with practically no drop to cause degradation. When coal is loaded into box cars, the pivoted-boom conveyor-type box-car loader is used, the discharge height of this loader also being adjustable to minimize breakage of coal in the loading process.

Coal and Ashes Handling at the Power Plant

Coal is delivered to its destination by water or by rail.

Ordinarily, water-borne coal, arriving in vessels, is removed by clamshell buckets which are hoisted and dumped direct into storage or into a hopper, whence it is conveyed to a bunker located in front of and above the boilers. The buckets are operated from hoisting towers, or bridges, of either stationary or traveling type.

At power houses which receive their coal in railroad cars, the coal may be shoveled by hand over the side of the car, but it seldom is, as this method is too slow and expensive, demurrage is high, and it is wasteful to use man power which can be employed to better advantage for other purposes. This method of unloading cars is in the same category as the hand firing of boilers.

Car Unloading. It is customary to dump part or all of a car's contents into a hopper, located under the coal supply track—a satisfactory method when bottom dump cars are available, when adequate trackage and switching facilities are provided, when the coal is not partially frozen in the cars, and when not more than 300 to 400 tons of coal are to be unloaded during a period of 8 hours. The dump car and track hopper are by no means an altogether satisfactory combination, hence are generally used only by plants which consume too little fuel to warrant the larger investment required to install better methods.

The large or super-power house must unload its coal quickly and cheaply. There dare be no shortage of fuel, even though shipments be delayed, coal be partially frozen in the cars, or bottom dump cars unobtainable. To meet all these troublesome conditions the rotary railroad-car dumper was developed, which overturns and discharges the contents of the smallest or largest open-top car of any type within two minutes when desired.

Coal Elevating and Conveying. Since coal-crushing and pulverizing equipment is heavy and requires substantial foundations, it is, in practically every case, located at or near the ground level. Therefore the crushed product must be elevated and conveyed to the overhead storage bunker to be drawn off as required by the stokers, by the direct-fired pulverized-firing system or by the central pulverizing system when pulverized coal is stored.

For the quite small plant, which can support only a small investment in equipment, the centrifugal discharge bucket elevator fulfils well the purpose of elevating and discharging the coal into the storage bin.

When the coal-handling requirements are fairly moderate, the continuous bucket elevator is preferable to the centrifugal type, and if the overhead storage bin extends in front of several batteries of boilers, the elevator is supplemented by a scraper flight conveyor overhead, which distributes the coal along the length of the bin.

Medium-sized boiler houses may be provided with several types of handling equipment, the selection of which depends largely upon the surrounding conditions, such as location of track upon which coal is received, location available or desirable for storage bunker, and whether the coal and ashes may be advantageously conveyed and dumped into their respective storage bins by the same system. The leading types are: pivoted bucket carrier, gravity-discharge elevator-conveyor, belt conveyor, and skip hoist.

Pivoted-Bucket Carrier. Probably the overlapping pivoted-bucket carrier is the most generally used and applicable type of conveying equipment for serving the medium-sized plant, as it performs the several functions of receiving both the coal and ashes, separately of course, in its lower run, conveying the material to one end of the boiler house, elevating it, and then conveying it to the overhead coal or ash bunker, where the contents of the buckets are dumped and distributed. After 30 years of use, it is still a leading type for handling coal and ashes within its range of capacity and height. Its maintenance cost is low and its dependability high.

Gravity-Discharge Elevator Conveyor. The development of this combined elevator and conveyor preceded the pivoted bucket carrier, but it still is used widely for handling just the coal, in power plants of moderate capacity. In fact, it is this type of conveyor that is in use in the boiler house of the world's largest building, the Merchandise Mart in Chicago. Their conveyor follows a rectangular path of approximately 203-ft. horizontal centers by 30-ft. vertical centers, travels at a bucket speed of 50 ft. per min., and handles 30 tons of coal per hour. The ashes are discharged direct into tunnel cars and hauled away by an electric engine.

Belt Conveyors. Large and super-power houses require large capacity and high lifts. The belt conveyor and skip hoist types of coal-handling equipment are generally used to meet these exacting requirements.

When the coal-supply track is located so as to permit a belt conveyor to attain sufficient height before reaching the boiler house to enable it to pass over the storage bunker and distribute its load to that bunker by means of a tripper, the belt conveyor makes an excellent selection. For distributing the coal to the large overhead bunker, which should contain a supply for at least a 24-hr. maximum boiler demand, the belt conveyor is provided with a traveling tripper of the automatic or semi-automatic type. Preferably, the tripper should have anti-friction bearings, large shafts, and pulleys, and its traveling mechanism should be driven by an independent motor direct-connected to an efficient speed reducer mounted on the tripper frame. This type of drive is superior to the type driven from the conveyor belt, which requires troublesome gearing, clutches, and reversing mechanism.

Should the supply track be located alongside or at one end of the boiler house, the skip hoist is well adapted to elevate coal from the crusher and dump it into the overhead bin direct or into a hopper feeding a belt or flight conveyor for distribution to a long overhead bunker. As the skip is hoisted vertically or at a slight inclination, it requires but a minimum of space. It has the additional advantages of being adapted to high lifts and to large or small tonnage capacities per hour.

Overhead and Reserve Storage. Few designers have made the mistake of providing too much overhead storage for coal ready to be delivered to the stokers, but many have erred by having insufficient storage capacity to allow for delays in receipt of shipments, thus necessitating urgent messages by the operating department.

For the quite small plant, requiring a few tons a day, accommodations should be for at least two days' consumption, plus a full carload. The reason is obvious.

The medium-sized plant should have two days' maximum supply overhead storage and an auxiliary ground storage supply for one or two weeks' requirements adjacent to the boiler house, which can be drawn upon immediately in case of delayed shipments or bad weather. If it is impossible to provide an auxiliary storage, the overhead bunker capacity should hold at least one full week's supply.

The large or super-boiler plant burns such enormous amounts of coal that it seldom is practicable to provide more than 24-hr. overhead storage. These plants, however, always provide a secondary storage adjacent to the boiler house, from which they can quickly and economically draw enough coal for their daily requirements to insure uninterrupted operation. This ground storage, sufficient to take care of operation for one or more months, is usually equipped with cranes, bridge tramways, or drag scrapers, for economically storing and reclaiming large tonnages per hour. A large auxiliary storage permits purchasing coal at favorable prices, in addition to insuring a desirable quality and quantity of coal at all times.

Ordinarily, the overhead storage is located in front of the boilers, but there are conditions where it is desirable to place it centrally with or at one end of the boiler house and to distribute the coal to the stokers as needed by means of a traveling weigh larry. This latter arrangement is necessary where stokers, overhead storage, and coal-handling equipment are installed in an existing boiler plant, to avoid remodeling the boiler-house building and roof.

Coal-Weighing Tools. Track scales serve to weigh and check carload shipments as they are received for unloading to either overhead or ground storage.

Weightometers are used to weigh and record automatically the weight of coal passing over the conveyors on which they are installed, before the coal is discharged to the bunker. The weightometer provides an accurate record of all coal delivered to the bunkers, practically eliminating the human element.

Weigh larries are located under the overhead storage bunker to permit drawing the coal from storage into the larry hopper, for weighing and transferring to the stokers as needed, the larry traveling on overhead tracks in front of the boilers and under the bunker. The weigh larry accomplishes the following desirable results: It transfers the coal from overhead storage to stokers; weighs and records the total amount of coal drawn from the overhead bunker; weighs and records the total amount of coal delivered to each stoker; and transfers coal from any part of overhead storage to any stoker, thus avoiding dead or unusable overhead storage over boilers which are not in operation.

Weigh larries vary in capacity from 1000 to 20,000 lb. or more, depending upon the size of plant, the length of travel, and the number of boilers served.

When the boilers are fired with pulverized fuel, automatic coal scales may be introduced between the coal bunker and each boiler, thus obtaining an accurate record of the amount of coal burned by each boiler.

Foreign-Substance-Removal Equipment. Coal, as received, will contain blocks of wood, timbers, coupling knuckles, links and pins, mining-machine cutter bits, and other refuse. Unless these are removed, they are likely to damage subsequent handling equipment seriously; therefore means are provided to get them out. Frequently an attendant is supposed to watch for injurious material as the coal flows from the car, and remove it. Unfortunately, man is not a dependable tool for many purposes, and is therefore unsatisfactory.

Hammer and ring mills have effective arrangements for catching and removing tramp iron or other materials that will not pass through their cages. The Bradford breaker prevents the larger pieces from passing through its perforations and reaching the bunkers. Apron feeder conveyors may be provided with magnets at their discharge end to remove tramp iron of all kinds and prevent its reaching and injuring the crusher.

When coal is to be pulverized, it is especially important to remove all iron before it can reach the fine pulverizers. This is done by installing a magnetic pulley on a belt conveyor over which the coal passes, which will effectively remove the iron.

Ashes Handling. Where a small amount of ash is to be handled, the wheelbarrow and shovel still hold first place. When the ashes must be elevated and discharged at street level into wagons or into a small overhead bin, the ash lift and centrifugal discharge bucket elevator serve the purpose and require minimum investment. Frequently portable belt conveyors or elevators are advantageously used to load piles of ashes into wagons or trucks.

For conveying and elevating the ash and discharging it into an overhead bin, the steam jet is well adapted where the ash must travel a tortuous path and where there is little space for other types of equipment. This system requires considerable steam if the quantity of ashes hoisted is large; also the abrasive quality of ashes tends to wear out the turns and pipes, unless they are specially reinforced.

If the ashes can be handled in the same equipment that is used for handling the coal, the overlapping pivoted bucket carrier makes an efficient and satisfactory installation.

The skip hoist is the best ash-handling tool, when the quantity is large enough to warrant the necessary investment, and when separate ash-handling equipment is required or advisable. Frequently the ashes are drawn from pits or hoppers into industrial cars which are pushed along a track and dumped into the skip bucket for hoisting and dumping. As the skip bucket is suited for handling large clinkers, small ash, or soot, it is commonly used for them as well as for the coal. The equipment is the same as for coal except that the automatic loader is seldom used.

Large boilers produce large quantities of ash, hence are often provided with big cast-iron ash hoppers and gates lined with tile; also with clinker chambers, clinker grinders, sluiceways, nozzles, and pumps; or they may have railroad tracks located under the ash hoppers, so that the ashes may be emptied directly into railroad cars. Storage bins of concrete, of cast-iron sections, or of steel lined with tile, brick, or concrete, are employed for storage of ashes until a sufficient quantity has accumulated to be drawn into the railroad cars or trucks.

Locomotive Coaling Stations

Elevating and conveying machinery is extensively used in the locomotive coaling stations of the country.

At a large, all-steel, six-track coaling station on the New York Central, for example, one will see coal being delivered to the overhead storage pocket by means of a balanced-bucket skip hoist.

As in power plants, the pivoted bucket carrier also is a popularly used type on locomotive coaling station work, being employed for handling both the coal and the ashes.

At another station, with two unloading tracks and four coaling tracks under two 500-ton concrete pockets of circular shape, there are two track dump hoppers with apron conveyors feeding the coal to a gravity-discharge elevator-conveyor, which delivers it to either of the two storage pockets.

The selection of the type of conveyor or elevator to be used depends upon the capacity desired, local conditions, and individual preferences.

Retail Coal Yards

The retail coal yard may employ overhead pockets and deliver the coal to these pockets either by means of a gravity-discharge bucket elevator, a pivoted bucket carrier, or a trolley-bucket conveyor system.

The gravity-discharge V-bucket elevator has been used a great deal, and discharges the coal to the various bins direct when it follows a rectangular path with the upper run operating the length of the building. The discharge is through gates in the bottom of the trough along which the buckets push the coal.

The discharge from a scraper flight distributing conveyor is similar, when a flight conveyor is used in combination with a

gravity-discharge bucket elevator delivering to it at the one end of the pocket lantern.

To avoid degradation, some form of lowering chute is used to "ease" anthracite coal to the bottom of pocket or top of pile.

There is a growing tendency to build silo-type concrete pockets and to install the trolley-bucket system of handling the coal.

The latest form of trolley-bucket system is fully automatic, serving to unload coal (or coke) from hopper-bottom railroad cars direct to any point along the top of storage piles of any height, at maximum speed, without transfer, and with minimum degradation, since the trolley bucket automatically lowers itself to the pile—this contact of bucket with pile effecting the discharge of the bucket's contents. There is but a short drop from the track dump hopper loader into the trolley bucket. The discharge to the pile (usually inside of a pocket) is a gentle spreading or flowing of the material over the pile. The material handled is at rest with respect to the bucket while the latter is being trolleyed up and on to its predetermined discharge point.

The system permits the use of high bins or silos, thus saving ground space and eliminating mechanical handling for loading trucks. The delivery chutes are so located in the bins that coal can be drawn off direct to trucks or to a bagging platform which is level with the truck body. The attendant can devote most of his time to seeing that coal keeps flowing from the railroad car into the track hopper.

Many coal yards, especially the smaller ones, unload the coal on a trestle siding through hopper bottoms of cars to ground bins, and do the reloading to delivery trucks by means of portable loaders, which are either of the belt conveyor, flight conveyor, or bucket loader type, depending upon the kind and size of coal handled and other conditions.

For cleaning anthracite coal (or coke) before delivering to truck, the bucket loader is often equipped with a screen of either the shaking or vibrating type.

The flight conveyor loader is widely used for reloading bituminous coal, especially coal with large lumps, as the flights keep the lumps from rolling back.

Making and Handling Coke

At the steel mill, coal must be converted into coke before it can be used to provide the heat for melting the mixture of iron ore and limestone.

The coke must be tough, range in size between 2-in. and 3-in. cubes, and be so constituted chemically as not to add to the undesirable impurities in the iron. To produce such coke, the coking coal should be carefully selected and prepared. Generally, three or four kinds of coal are mixed to provide the proper chemical ingredients.

In getting this coal from the track dump hopper to the crusher and to the mixing bin, it is handled by the usual forms of conveyors, such as belt conveyors, bucket elevators, pivoted bucket carriers, or skip hoists, depending upon the plant layout.

A motor-propelled larry car handles the prepared coal from a bin to the coke ovens, charging them through openings in their tops.

The coal requires up to 24 hours for coking, depending on the quality of coke required and the nature of the by-products to be derived. A coke-pusher car pushes the coke from the desired oven into a quench car running on tracks on the other side of the row of ovens. This car takes the coke under a quench tower where the coke is water-drenched, and then discharges it to the coke wharf for cooling.

The wharf is a long, flat surface sloping downward about 25 deg. from the horizontal, and the coke slides to the lower ledge, from which it is discharged through manually operated finger gates on to a belt conveyor, which carries the coke to the crushing and screening plant. Here the coke is prepared for blast furnace or foundry use in the quantities needed, and the remainder is sent to outside storage or to the domestic-coke screening plant. Coke that has been in storage is unsuited for blast-furnace use.

In the preparation plant one may see the liberal use of belt conveyors for handling the coke from one operation to another, coke crushers and screens of the shaker, multiple-disk, revolving grizzly, and vibrating types—the latter particularly being used in sizing and cleaning the coke for domestic use, just before delivery to trucks.

At the coal-gas plant the coke is a by-product. At one plant in mind, for instance, the accumulation of coke amounts to about 2000 tons daily. To dispose of such large quantities profitably requires the use of efficient mechanical-handling equipment and the preparation of the coke in such a way that it will meet the competition of anthracite coal for domestic use, as well as the growing demand on the part of householders for a cleaner coke. The oversize coke is sold as foundry coke.

Coke breaks up very easily, and for this reason should be handled as gently as possible so as to avoid an accumulation of "fines" or breeze. The belt conveyor is an ideal coke-conveying unit. The construction of transfer points should be watched so that chutes are made amply wide for the coke to flow over them in a thin stream, and no steeper than necessary. Pocket chutes which fill with coke have proved very satisfactory by eliminating chute wear and causing the coke to roll rather than to slide.

At the coal-gas plant one is likely to see the hot coke conveyed from under the vertical retort-type ovens by a de Brouwer drag-chain conveyor, consisting of two strands of short-pitch special sliding chain with flat crossbars at intervals of about 2 ft., and operating in a structural-steel trough which is lined with hard white iron, in view of the abrasive nature of coke. This conveyor is light in weight and operates slowly, and the joint wear is comparatively small due to the fact that there practically is no joint movement under stress except when passing around the head sprocket wheel.

This conveyor, which also acts as a quenching or cooling conveyor, takes the coke to storage or to the screening plant. Pivoted bucket carriers also have been employed for handling the hot coke, or a quenching car may dump the coke into a pit from which a monorail grab-bucket machine will reclaim and handle the coke to the storage pile or to a receiving hopper of the crushing and screening plant.

At the water-gas plant one is likely to see an automatic skip hoist alongside of the railroad track, serving to receive coke from the track dump hopper through an automatic loader and to deliver it into an overhead storage bunker, from the bottom of which the coke is delivered to the retort-charging buggies over a vibrating screen. This screen removes the breeze, the cleaner coke resulting in greater efficiency of the retort fire.

Many of the gas plants are now using a crawler-type bucket loader equipped with vibrating screen for reclaiming coke from pile direct to delivery truck. A self-feeder serves to feed the coke to within reach of the buckets. The vibrating screen cleans the coke thoroughly. There is also a smaller capacity bucket loader, hand-moved over ground, which cleans the coke very well over a shaker screen.

In the oil fields one may find petroleum coke pulled out of chambers on to a portable apron conveyor discharging to a portable belt conveyor for delivery to side-dump cars running to the screening plant.

Ore-Handling Equipment

Iron Ore. The steel companies are ever searching for ways and means to lower ore-mining costs, improve the quality of the

ore for blast-furnace use, and reduce the cost of handling the ore from the mine to the furnace.

Iron ore is mined underground and hoisted to the surface in skips or from an open pit by means of large-capacity stripping shovels delivering into cars. Restricted area and a deep pit may necessitate "milling" the ore to haulage level and raising it through a shaft which is sunk adjacent to the pit when the pit-car track grades have reached the limits of economical haulage with standard locomotives.

The cars used in underground mines are of 3 to 5 tons' capacity. They are dumped directly into skips or else discharged to underground loading pockets with measuring pockets between them and the skips. The skips range from a holding capacity of $2^1/_2$ tons to 8 tons, skips with a capacity of 4 and 5 tons being mostly used. In the open-pit mining the haulage cars may be the air-dump type, with a capacity of 20, 24, and 30 cu. yd.

When lean ore is mined it must be "benefited," thereby making the ore more suitable for use, besides effecting a saving in freight charges through not having to pay freight on the weight of the impurities and moisture which are removed by the treating processes used. There are five processes commonly employed, these being roasting, sintering, washing, washing and jigging, and magnetic separation. The nature of the ore and of its base determines the process that should be used, and all five methods involve mechanical handling of the ore.

At the primary screening and crushing plant where trainloads of run-of-pit ore are dumped, metallic apron feeders are employed to feed the oversize ore uniformly from the bar grizzly to the primary crusher. The crushed ore is then delivered by rubber belt conveyors to secondary crushers or is delivered to railroad cars for shipment to ship-loading docks.

The apron feeders must necessarily be of very rugged design and construction to withstand the punishment to which normal service subjects them; in fact, it was ore-feeding service that brought forth, 8 or 9 years ago, the distinctive type of heavy-duty ore feeder now used.

The sprockets, chains, rollers, and overlapping pans are usually made of cast manganese steel. The chains have no small rollers (therefore there are none to be lubricated), and are placed underneath the pans, out of the dirt, at locations which break up the span of the pans. The broad wearing surfaces of the chain links form substantial tracks for the large-diameter manganese steel rollers supporting the material-carrying run. These rollers are closely spaced and are mounted on heavy through-shafts turning in rugged, accessibly located bearings.

On short feeders the lower or empty return run pans and chains are unsupported between the head and foot shafts. Medium or long center feeders are provided with manganese steel rollers, suitably spaced, to carry the return run.

Troughing-type three-pulley anti-friction belt conveyors are used, practically without exception, for handling crushed iron ore in today's modern crushing, washing, or screening plant.

Copper Ore. Copper ore is mined underground or by open-pit method, and the system of getting the ore to the surface is similar to that at iron-ore mines.

The heavy-duty manganese steel apron feeder, already described under "Iron Ore," is employed for feeding the run-of-pit ore from dump hopper to crusher direct or to a conveyor.

At one property in mind, the feeder is 8 ft. wide and employs 18-in. pitch conveyor chains. This feeder has been operating over three years at a handling capacity of approximately 8000 tons per day. It requires no attention and receives none, except greasing once a day and cleaning the floor underneath of a small quantity of "fines" which stick to the pans due to rather heavy sprinkling of the ore before it is dumped.

Trains from the pit dump six 40-ton cars into a bin, from which the feeder, which is 72 ft. long, conveys the ore to a 56-in. gyratory crusher. Besides the "fines," the feed contains a large percentage of coarse material, with pieces 5 or 6 ft. long and 3 ft. by 3 ft. or more in cross-section.

The feeder speed is $9^1/_2$ ft. per min., and the handling capacity is 900 tons per hour. There is no leakage through the top run nor spill between the skirt boards and the conveyor's side flanges. The control is by push-button, the operator (located at the crusher) shutting down the feeder just before the bin is emptied. There is no gate at the bin throat, and there is never any jamming.

Belt conveyors form the ideal and popular conveying medium for handling the crushed ore.

Nickel Ore. This ore is mined very extensively in the Province of Ontario, by the underground method—the Frood mine, for example, being designed to hoist the nickel-copper ore in two 10-ton balanced skips at the rate of 6000 tons a day.

This mine has levels of 1200, 1600, 2000, 2200, 2400, 2600, and 2800 ft. Primary crushing is done below the 2800-ft. level, the jaw crusher located at this point being fed from an ore pass from this level and the levels above. The crushed ore is discharged into an ore pocket, and then into measuring hoppers just before delivery to skips.

Treatment of ore in the Frood rock house involves crushing, coarse screening by trommel screens, fine screening on vibrating screens, hand picking of waste on belt conveyors, and magnetic separation, entailing the use of a large number of anti-friction belt conveyors.

At the Levack mine, designed to hoist 3000 tons a day, there are one heavy-duty-type manganese-steel apron feeder and one rolled-steel pan conveyor in use in the rock house and 18 belt conveyors. The apron feeders also serve as picking conveyors. A feature worth mentioning is the extensive use of magnetic head pulleys on the belt conveyors for separating the ore from waste rock, the product of this mine containing a larger percentage of nickel than copper.

The crushing mill, concentrator, and smelter at Copper Cliff, Ontario, is designed to crush, concentrate, and smelt approximately 8000 tons of nickel ore per day. Inasmuch as the ore is handled entirely by conveyors from the time it leaves the receiving bins until it reaches the roasting furnaces in the smelter, a great many of them are necessary. Consequently they form a very important link in the process. In the smelter conveying system are 47 belt conveyors, using a total of 24,000 ft. of rubber-covered conveyor belt. All cold material is handled on belt conveyors, ranging in width from 24 in. to a maximum of 48 in.

In addition there are several scraper flight conveyors, chain drag conveyors for handling hot calcine, and a number of encased 14-in. spiral ribbon conveyors, as well as drag conveyors, for handling all dust.

Gold. The handling of materials at underground gold mines, of which there are a number in operation in the "north country" of Canada, is very similar to the procedure for extracting nickel-copper ore, but not quite so extensive.

The ore is received from the mine by skips and discharged into ore bins in the head frame. From these bins the ore may be fed by pan conveyor, chain conveyor, rotary grizzly, etc., either direct to a jaw crusher or to a belt conveyor and then to the crusher. The crushed material is then usually passed over a vibrating screen and the oversize is fed to a cone crusher for further reduction. The screens and cone crusher are usually in a closed circuit, and the ore is handled almost entirely on rubber-covered belt conveyors. Inclined belts are used in the mines of Ontario almost to the exclusion of bucket elevators.

The ore, after being crushed to about $^3/_8$ in., is conveyed by belt conveyor to the fine-ore bins in the mill. From these bins

the ore may be fed by roll feeders, pan conveyors, or short-center belt conveyors to a belt conveyor for distribution to the fine-grinding department, which may comprise ball, tube, or rod mills. At this point the ore is made into a free-flowing solution, and circulated through classifiers, thickeners, agitators, precipitating equipment, filters, etc. The "tailings" from the filters are usually discharged to a flat belt conveyor.

As different ores require different methods of handling, there also will be deviations from the general flow sheet just outlined.

Earth Excavation

The removal of Denny Hill, 110 ft. high, containing 4,500,000 cu. yd. of dirt, was a big undertaking, and it is significant that this little mountain, which tended to retard the commercial expansion of the City of Seattle, was conveyed a distance of about 3000 ft. to the sea entirely on belt conveyors within 24 months after the contract was let, the first 6 months of this period having been consumed entirely in the assembling of the equipment and placing it in operation.

Belt conveyors were decided upon as the most logical, since such equipment was light in weight and could be readily supported on a wooden structure above the street-car trolley lines. The belt conveyor also fitted admirably into the scheme, as it operates so noiselessly, and by its use it was possible to keep the streets clean and to permit traffic to proceed normally and without interference. All belt idlers were of the anti-friction type, thus keeping power costs and attention to lubrication down to the absolute minimum.

The dirt was dug by six 2-yd. electric shovels and delivered to portable field belt conveyors through steel feeding hoppers fitted with apron conveyor feeders. Whenever required, the material was crushed or mashed by the shovel dipper through grizzly bars set 10 in. apart over the hopper. There were 12 field conveyors 250 ft. between centers, delivering to a central receiving hopper serving the belt conveyor, 960 ft. between centers, which carried the dirt up over the street. This conveyor delivered to a conveyor, 1560 ft. between centers, and it, in turn, to a belt conveyor, 440 ft. between centers, running along the face of one of the docks at the waterfront, where the dirt was delivered by chute to self-dumping scows.

The scows were constructed with the top and bottom alike, and with internal water chambers divided into two parts. After the scow was loaded, it was towed out into the bay and water was let into the one side of the empty chamber in the scow hull, and as the scow listed, due to the water entering this chamber, the dirt began to roll, and this shifting of the load resulted in the scow turning completely over during the operation of dumping. Then what was the bottom of the scow became the top, ready to receive another load of dirt and to be dumped in the same manner, each time turning over in the dumping operation.

All belts used were 36 in. wide. Those over the street operated at a speed of 600 ft. per min. The portable conveyors ran at a belt speed of 400 ft. per min. The average handling capacity of the main belt line was 600 cu. yd. per hour, bank measurement, and the equipment worked three shifts per day, $7^3/_4$ hours per shift. Many of the conveyor belts lasted through the entire job, and the maintenance was not great, taking into consideration the conditions under which the conveyors had to operate and the service rendered.

As the dirt built up rapidly on the steel chute bottoms, the transfers between conveyors over the street were made by having the conveyors discharge direct to the belt of the conveyor ahead, with nothing more than chute sides at these points.

The contract price for removing Denny Hill was 25 cents per cubic yard. The digging was sublet at 9 cents a yard delivered to field belt conveyors. this figure being included in the total price of 25 cents per yard.

Portable belt conveyors also have been employed in the process of removing dirt from excavations for large buildings.

Conveying of Other Materials

Only a few of the many other materials handled in bulk will be covered under this section, and this will be done briefly.

Salt. This is handled by bucket elevators, skip hoists, belt conveyors, flight conveyors, and screw conveyors. The belt conveyor is widely used, and to avoid incrustation and corrosion of the conveyor idlers, these idlers have been granitized, with good results. Screw conveyors handling wet salt to driers have been made of Monel metal to prevent discoloration and contamination of the salt.

Sawdust. This is usually conveyed by wide-link chains which push the material along the trough in which the conveyor operates.

Hogged Fuel. The waste wood of the saw mills and paper mills is conveyed by wide-link chain conveyors as used for sawdust, by double-strand flight conveyors using wooden flights intermittently mounted on chains, and on belt conveyors.

Pulpwood Chips. Pulpwood chips for making paper have been handled by flight conveyors, screw conveyors, bucket elevators, and belt conveyors. The belt conveyor is the ideal type, as it carries the chips, thus preventing their breakage. Small, dirty chips produce a short-fiber paper pulp, which is no longer desirable in the manufacture of high-grade finished paper.

Fullers' Earth. This material, used in large quantities in filtering oil at refineries, is handled by bucket elevators, pivoted bucket carriers, suspended roller flight conveyors, and belt conveyors. At one of the most modern of these refineries the earth is delivered to filters and to storage bins by means of anti-friction belt conveyors equipped with traveling trippers.

Mixed Concrete. This is now often handled over a system of portable belt conveyors in the construction of large concrete buildings and in the pouring of foundation piers. For the construction of Diablo Dam, recently completed, a unique arrangement of swinging belt conveyors, suspended from 400-ft.-high steel concreting towers, was worked out to reach all parts of the job.

Crushed Ice. Ice for refrigeration lends itself to handling in screw conveyors, and this, for example, is how it is handled from the crusher to the packers in the extensive lettuce-packing industry of California.

Sludge, Sand, and Grit. These are removed from sewage settling tanks by a special form of flight conveyor.

Coffee. Coffee berries are handled by bucket elevators and belt conveyors in the modern coffee-roasting plant.

Corn. In the corn-canning plant conveyors handle green unhusked corn, clean corn on the cob, corn husks, and cobs in bulk. The flight conveyor usually is employed for handling the green corn, husks, and cobs, and the belt and apron types for the clean corn on the cob.

Rotary Mud. This is now salvaged over vibrating screens. Thus the oil-drilling industry is able to remove the cuttings very effectively and place the mud so necessary in drilling, in good reusable shape, free from foreign materials. The screen also performs the important function of degassing the mud.

Conclusion

Thus, from these concrete examples, there may be learned the amazing extent to which mechanical conveying equipment enters into the expediting of deliveries, the enlarging of plant handling capacities, and the reduction of industry's production costs.

MH-53-9

The Industrial-Type Internal-Combustion-Engine Locomotive

By S. B. SCHENCK,[1] EAST PITTSBURGH, PA.

The paper seeks to show that the internal-combustion type of locomotive can be used to advantage and at a much lower operating cost than the steam locomotive for industrial-plant use. The electric drive, in combination with gasoline or Diesel engines, places the internal-combustion type on a par with the steam locomotive so far as mobility is concerned, and ahead of the latter type with respect to availability for service, economy of operation, safety, visibility, convenience, and general performance. The electric drive is declared to be better than the conventional clutch and gear transmission between engine and axle as being better to operate, having more tractive power on grades, and in being much easier on the engine. There is available a wide range of internal-combustion type of locomotives using electric drives.



IN GENERAL, industrial rail transportation presents a marked exception to the prevailing progressive changes in different elements, methods, and machinery of industry. Many industrial plants that install up-to-date machinery or that use new methods of production still continue to use the steam locomotive for their rail transportation.

Probably this situation exists because of the fact that the steam locomotive has served industry long and faithfully as a motive-power unit, and it is still accepted as entirely suitable by many operators. On the other hand, when the almost exclusive use of the internal-combustion engine is considered for such essential industrial elements as construction machinery, airways, marine shipping, or highway transportation, the question arises whether the internal-combustion engine could be adapted satisfactorily to rail transportation. This question is emphasized when the cost of operating steam locomotives is reviewed critically—and incidentally such cost analyses often disclose real surprises to progressive plant managements, particularly when the steam locomotives are of ancient vintage.

The internal-combustion engine can be utilized on locomotives to good advantage from economy and operating standpoints, although much depends on the method employed to transmit the power of the engine to the rail, or in other words, on the type of locomotive. Recent developments in locomotive power transmissions have produced the electric drive, which, in combination with gasoline or Diesel engines, places the internal-combustion-engine type of locomotive on a par with the steam locomotive with respect to mobility and ahead of the steam locomotive with respect to such operating factors as availability for service, economy of operation, safety, visibility, convenience, and general performance. The electric drive has many advantages over the conventional clutch and gear transmission between engine and locomotive axle, in that it is easier to operate, gives better pulling power and superior overall performance on grades, is much easier on the engine, and produces a locomotive of greater general reliability and availability. It is rapidly supplanting the mechanical drive on locomotives weighing between 12 and 25 tons, and is still more widely used on locomotives weighing 30 tons and upward.

Gasoline-electric locomotives for industrial service are now available over a weight range from 12 to 90 tons, utilizing engine powers from 120 to 600 hp. Diesel-electrics range between 30 and 110 tons at present, although this weight range will doubtless be widened in the near future. Varied designs of locomotives have been produced, and for most industrial work the following locomotive characteristics are well established:

1 Reliable power plants
2 Pulpit, or center-type, cab with double control stations
3 Maximum visibility
4 One-man operation
5 Adequate locomotive auxiliaries
6 Swivel-type trucks
7 Engine power ordinarily about 6 to 8 hp. per ton of locomotive weight.

These features are embodied in the locomotives illustrated, except for the type of truck. All of the features are found on locomotives using dual power plants, center-type cab, and swivel trucks, as shown by Figs. 1, 2, 3, 7, and 9. Fig. 8 illustrates a single power-plant locomotive with swivel trucks and a pulpit type of cab. Other combinations of these features are shown by Fig. 6, where the dual power plant is adapted to the rigid wheelbase, two-axle type, or by Figs. 4 and 5, that typify construction ordinarily used on the smaller locomotives, weighing about 25 tons or under, where a single engine, four wheels, and end type of cab give very satisfactory results.

Equal power on locomotives up to about 500 to 600 hp. can be obtained by the use of either single or dual power plants. The single power-plant type, illustrated by Fig. 8, is preferred by many operators on account of there being only one engine and generator to control, inspect, and maintain. Obviously, a single large engine has fewer wearing parts than two smaller engines. The large engine usually operates at slower speed than an engine of half size, and it therefore can embody greater general ruggedness, stamina, and design factors which tend toward longer life, great reliability, and satisfactory maintenance.

On the other hand, the use of dual power plant provides a flexibility in locomotive operation and safeguarding of service unobtainable by a single power plant. Because locomotives weighing in a range from 30 to 110 tons have been equipped with dual power plants, a brief review of the operating features of this type should be interesting.

From the standpoint of first cost, at present dual power plants totaling about 500 hp. or lower usually are less expensive than single power plants of equal capacity, on account of the smaller

---

[1] General Engineering Department, Westinghouse Elec. & Mfg. Co. Mr. Schenck has had four years' experience in this department in the application and equipment of self-propelled cars and locomotives.

Presented at the "Three-M" (Management, Materials, Maintenance) Congress held in Cleveland, Ohio, April 13 to 17, 1931, under the auspices of the A.S.M.E. and the S.I.E.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

engines being in standard production by their manufacturers. This is particularly noticeable where gasoline engines are concerned. There are fewer available sizes of locomotive-type oil engines, so that this situation does not prevail to the same extent as with gasoline engines, and in some cases the single power plant may have lower first cost. Two engines also afford greater safeguard to the service, since it is unlikely that both engines would

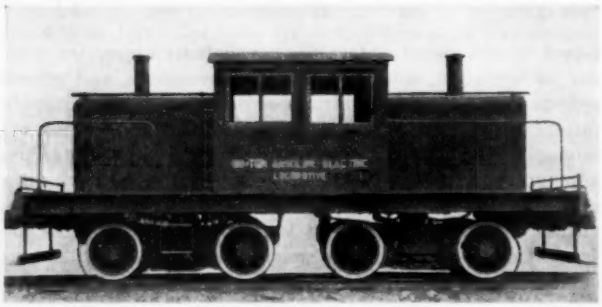


FIG. 1 30-TON, 230-HP. GASOLINE-ELECTRIC LOCOMOTIVE, USED IN INDUSTRIAL-PLANT SWITCHING
(Built by H. K. Porter Co., Westinghouse equipped.)

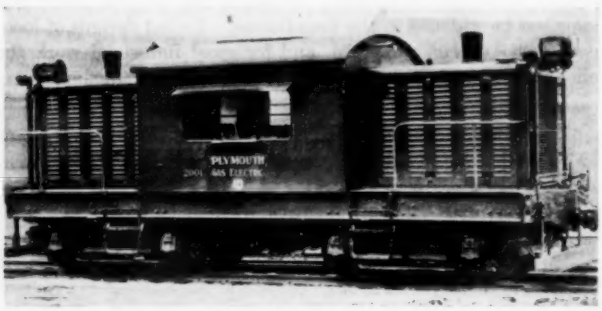


FIG. 2 50-TON, 350-HP. GASOLINE-ELECTRIC LOCOMOTIVE, FOR GENERAL SWITCHING
(Built by Plymouth Locomotive Works, Westinghouse equipped.)



FIG. 3 60-TON, 360-HP. DIESEL-ELECTRIC LOCOMOTIVE, FOR GENERAL SWITCHING
(Built by H. K. Porter Co., Westinghouse equipped.)

be out of service at the same time, and experience has shown that minor repairs can be made to one engine while the locomotive continues in service, working from the other engine. Under such circumstances service interruptions are minimized, for with one engine working, the locomotive can operate at half load and full speed or at full load and half speed.

The dual power plants may also assist in securing minimum fuel consumption, for in light service, such as occurs in much of the usual switching duty, one engine can be shut down at the will of the operator and the service easily be covered by the remaining power plant. With gasoline engines this can show a saving in fuel of 10 per cent or more as compared to working two engines in the same service, and it can show a greater saving if one large engine of equal power were used. The saving is considerably smaller for Diesel engines, due to inherent differences between the light-load fuel-consumption rates of the gasoline and Diesel engines. For either type in suitable service, the operation of a single engine of the dual power plant secures more nearly full load for a larger proportion of the time, hence at better fuel-consumption rate that results in reduced fuel consumption for the locomotive.

It is interesting to note that in much switching service large

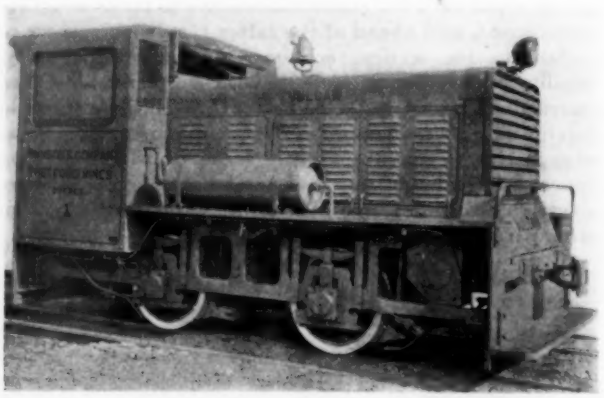


FIG. 4 12-TON, 115-HP. GASOLINE-ELECTRIC LOCOMOTIVE, USED IN MINE HAULAGE
(Built by Vulcan Iron Works, Westinghouse equipped.)

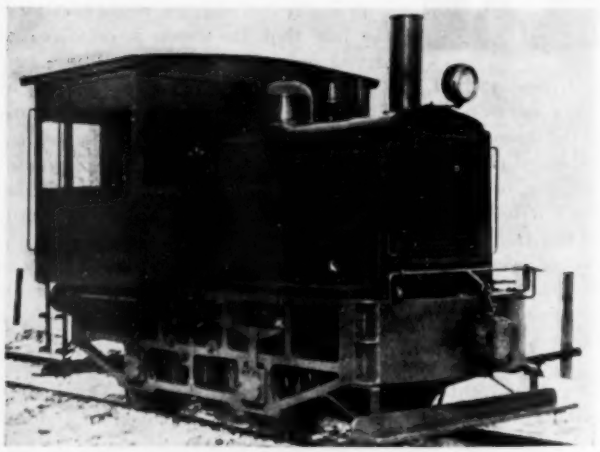


FIG. 5 17½-TON, 165-HP. GASOLINE-ELECTRIC LOCOMOTIVE, USED FOR LIGHT INDUSTRIAL-PLANT SWITCHING
(Built by G. D. Whitcomb Co., Westinghouse equipped.)

power in the engine is not required continuously, nor can it always be used advantageously. As an example, switching the cars from floats and classifying the cars in a yard showed the following daily average division of engine operating time and cars handled by a dual power plant, 330-hp., 50-ton gasoline-electric locomotive:

Time, locomotive in service, min	429
Time locomotive standing, total min	275
Time locomotive moving, total min	154
Distribution of locomotive moving time:	
Two engines and four motors, 30 min., cars moved	252

One engine and four motors, 96 min., cars moved	575
One engine and two motors, 28 min., cars moved	82
Cars moved, total	909
Total number of movements	283
Average length of run with trains, ft	278
Estimated ton miles, including light locomotive	2907
Estimated locomotive miles, total	15.2
Gasoline consumption, gal	60

In similar service, but with two engines working continuously, the fuel consumption averaged 72 gal. per 8-hour day. In the service referred to, the single 175-hp. engine and four-motor combination had no difficulty in maintaining the schedule of the yard operation and could kick cars as fast as the ground crew could handle them conveniently. The yard is comparatively level, and the trains were relatively short, being limited to the load on the float or in most cases to the maximum number of cars the locomotive could handle from the float at one trip up the 3

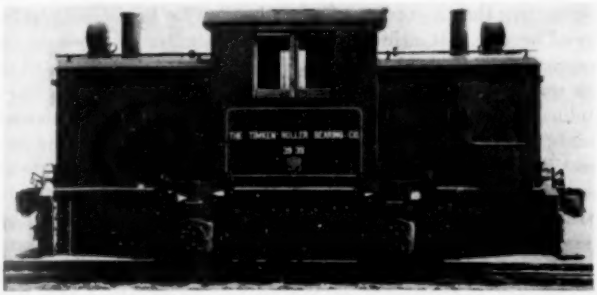


FIG. 6 35-TON, 214-HP. GASOLINE-ELECTRIC LOCOMOTIVE, USED FOR BLAST-FURNACE HAULAGE
(Built by H. K. Porter Co., Westinghouse equipped.)



FIG. 7 60-TON, 360-HP. DIESEL-ELECTRIC LOCOMOTIVE, FOR GENERAL SWITCHING
(Built by Heisler Locomotive Works, Westinghouse equipped.)

per cent grade and 18-deg. curve. The 50-ton gasoline-electric locomotive was handling larger trains from the float than the 65-ton steam locomotives were reported to handle.

A gasoline- or Diesel-electric locomotive with the pulpit, or the center type of cab, and with double control stations, allows operation with one man, thus reducing the usual switching-crew expense between 20 and 25 per cent per locomotive hour. Double control stations permit the operator to step from side to side of the cab and operate the locomotive from either side, enabling him always to operate from the side that has maximum visibility along the train or from where the ground-crew signals are most easily seen. When box cars are handled in large numbers, between buildings, an elevated operator's station in the center cab gives exceptional vision toward the crew on the tops of the cars. For example, the brakeman can be seen easily on the top of the fourth or fifth box car from the operator's station of a locomotive recently placed in yard switching. These features all aid materially in speeding up the switching service, thus indirectly reducing the expense in addition to saving the wages of the second man in the cab. In comparison with a steam locomotive, the visibility in the gasoline- or Diesel-electric is generally much better, and the fireman is never needed, although sometimes local operating rules require the second man in the locomotive cab.

For locomotives built as shown by Fig. 3, with 165-hp. to 225-hp. engines, visibility is much increased, not only by the raised platform in the center cab, but by the very low height of radiators and engine hoods. With this locomotive a man on the ground standing half a car length ahead of the locomotive can be seen by the seated operator, looking diagonally across the hood and radiator. A steam locomotive could hardly be built to give such maximum visibility.

When engine power and locomotive size require large engines,

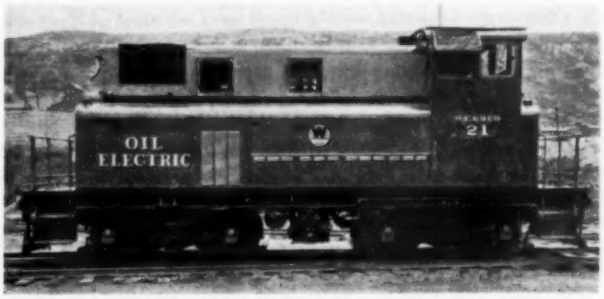


FIG. 8 70-TON, BALDWIN-WESTINGHOUSE 400-HP. DIESEL-ELECTRIC LOCOMOTIVE, FOR GENERAL SWITCHING AND HAULAGE

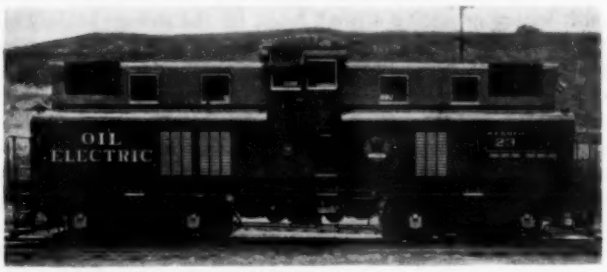


FIG. 9 110-TON, BALDWIN-WESTINGHOUSE 800-HP. DIESEL-ELECTRIC LOCOMOTIVE, FOR HEAVY-DUTY SWITCHING AND HAULAGE



FIG. 10 39-TON, 310-HP. LOCOMOTIVE-TYPE LARRY CAR
(Built by Differential Steel Car Co., Westinghouse equipped.)

it is evident that the same type of construction cannot be used as with smaller engines. The 800-hp. Diesel-electric locomotive shown by Fig. 9 has center cab with elevated operator's stations which give excellent visibility for this size of locomotive. Obviously the enclosure will be much larger than for the 400-hp. engines and different from that for the 180-hp. engines for the locomotive shown by Fig. 3, so that the operator does not have the same near vision across the hood and radiator as on the smaller locomotive, but he still gets better visibility than on a corresponding size of steam locomotive.

Where a large single engine is used, visibility in one direction is difficult to obtain on account of the bulk of the engine being in front of the cab. The locomotive shown by Fig. 8 is a good example of maximum visibility, convenience in operation, and compact assembly where the pulpit-type end-mounted cab is

used. Fig. 5 shows similar general features on the small single power-plant locomotive, where the size of engine permits good visibility to be obtained with conventional cab arrangement.

Often an industrial plant has use for both large and small locomotives. Where a certain size of power plant can be used on the small locomotives, and two of them used on larger locomotives, there is a gain in the standardization from the standpoint of the smaller variety, fewer number, and less expensive repair parts to stock, less total locomotive time lost waiting for special non-stock repair parts, and easier maintenance, as there is greater interchangeability of assembled parts. The size of the engine will not change the importance of these factors.

Where the industrial-plant service requires a large locomotive for a few operations and several small ones for most of the work, the small locomotives can be arranged for operating two in multiple, and thus give the same performance as a single locomotive of double the power. This is the equivalent of double-heading steam locomotives, except that only one man is required for the multiple-unit locomotive and the locomotives will operate in synchronism. Or again, multiple-unit operation may be desirable to distribute the required locomotive weight over numerous axles and at the same time obtain very short wheelbases for minimizing track upkeep or for operating large power on light-weight rails and around very sharp curves.

One-man operation is a feature of all the gasoline- or Diesel-electric locomotives. It takes no more effort for the operator to run a 110-ton locomotive than it does to run one weighing one-tenth as much. Starting the engine is a matter of operating a push button or setting a small lever, for the storage battery on the locomotive cranks the engine until it starts firing. The absence of smoke and steam enables the operator to observe crew and roadside signals with ease. He is comfortably housed and seated, and there is no boiler coal to handle, no steam pressure to keep up, and no ashes to dump, so that he can give full attention to his job of keeping the locomotive moving. There are no gears and clutches to handle, as on a mechanical-drive type of internal-combustion locomotive; hence the operator's duty is much lighter on the gas- or Diesel-electric, and he does not become so tired as the day's work progresses. This assists in improving the quantity and quality of work done by these types of locomotives. Experience has shown a reluctance on the part of the usual steam engineer to begin running a gas- or Diesel-electric, but after a few days' operation on either of these types, most engineers do not want to go back to operating a steam- or a mechanical-drive locomotive.

The average industrial railroad or yard involves sharper curves than on a steam railroad, and very often the condition of the track is only fair. Hence, the industrial type of locomotive should have a short and flexible wheelbase to give good tracking characteristics and to reduce rail wear on curves. Fig. 9 shows the usual swivel-truck arrangement that has proved exceptionally well suited for the average industrial track. Trucks of similar general design on lighter weight locomotives have given very satisfactory service on temporary tracks such as are laid in open-pit mines. In one mine the steam locomotives were derailed almost daily, but there have been very few derailments for the two years of operation of gas-electric locomotives. This mine also found the gas-electric could usually rerail itself, something that could seldom be done with the steam locomotive. In another large industrial plant the use of oil-electrics with swivel trucks, as typified by Fig. 9, is reported to have reduced track maintenance by about 20 per cent. These factors indicate some of the indirect savings made by the internal-combustion-engine type of locomotive.

Two general methods are used to operate locomotive auxiliaries, such as radiator cooling fans, air compressor, and main-motor blowers. The electric drive produces the best general result, although the mechanical drive has given satisfactory performance in switching services. The electric drive is necessary when radiator or blower fans are to be operated intermittently or when the fans cannot be driven direct from the engine or when the air compressor is to be operated at full speed with the engine either idling or running at operating speed. These conditions occur more often on the larger locomotives, and the locomotives shown by Figs. 8 and 9 are so equipped.

Electric auxiliary drive involves separate motors and their control, so that the comparative simplicity of the mechanical drive makes a strong appeal to many operators, although it has certain inherent operating and installation handicaps. It has given satisfactory service on industrial and switching locomotives of the type shown by Figs. 3 or 5, where the radiator fan is belt driven and the air compressor is belt or chain driven.

Fig. 7 shows a distinctive type of internal-combustion-engine locomotive that is exceptionally well suited for heavy-duty, slow-speed service. It utilizes power plants similar to those on the locomotive illustrated by Fig. 3, but the motors are mounted on the under side of the locomotive frame and are geared to a longitudinal drive shaft that transmits the power through universal joints and bevel gears to one axle on each truck. Siderods connect this axle to the other on the same truck. The axles are all connected mechanically through the siderods and drive shaft, with the result that the locomotive develops exceptionally high tractive effort before the wheels slip. On recent test, with sand on fair rail, the drawbar pulled developed repeatedly was over 40 per cent of the weight on the locomotive drivers. This type of locomotive provides a means for securing large power on narrow gages, as a change in track gage usually affects only the trucks and locomotive body width, so that within limits the same power plant and motors can be used for different gages.

An entirely different gas-electric locomotive is the burden-bearing type, illustrated in general by Fig. 10. This is in reality a dump car equipped with two 155-hp. gasoline engines and electric drives. It is designed to be more of a haulage unit than a general switching locomotive, but when loaded it is well adapted to handle trailers, and when empty to do the small amount of switching required around quarries, storage yards, etc. Doing double duty in this way, investment in equipment is reduced and the range of locomotive operation is widened.

Another type of industrial locomotive uses internal-combustion engines, either gasoline or Diesel, in combination with a storage battery. The battery acts as a reservoir capable of supplying power much in excess of that from the engine alone, thus materially improving the locomotive performance for short-time peak tractive-effort demands. Since the battery is charged from the engine-generator set mainly when the locomotive is not working or when the engine is idling, it is evident that this type of locomotive is best suited for services where there is much engine-idling time or where the locomotive is not kept in continuous duty. It is advantageous when the service requires locomotive operations inside of buildings where engine fumes are undesirable. Careful analysis of the service is always necessary to insure the proper size of battery and satisfactory balance between battery and engine power. If the battery is too small or if worked too hard, comparatively short battery life and consequent abnormal battery replacement expense may be expected. The analysis may also disclose service which is "lop-sided," so to speak, where the battery would be quickly discharged and there would remain insufficient time or available engine power to recharge it, so that the locomotive would have to be taken out of service for battery attention, thus reducing locomotive availability. The combination locomotive is heavier for equal continuous power than a straight gasoline or oil-electric, because

the battery is heavier than an engine of equal continuous output. Extra weight is very useful sometimes, but cannot always be considered advantageous. These factors indicate that the combination type of locomotive is adapted better for comparatively light intermittent duty or for special applications, rather than for general, all-round switching and transfer work.

The engine power which has been applied on industrial gasoline- or Diesel-electric locomotives varies much, depending on service, investment, available engine sizes, kind of fuel, and other factors. At present, satisfactory performance is obtained with gasoline-electric locomotives having an engine power range from 5 to 10 hp. per ton of locomotive weight, with an average of between 7 and 8 hp. per ton. Diesel-electrics range from 5 to about 7 hp. per ton, with an average of about 6 hp. per ton. There are many indications that the horsepower per ton will increase as suitable sizes of engines become available at satisfactory prices. A much wider range of suitable gasoline engines is available as compared to Diesel engines, and most engine builders are in standard production for gasoline engines of around 250 hp. or under, while comparatively few engine builders have Diesels in large production. The result is approximate price variations as follows:

	Dollars
Under 150 hp. gasoline, per horsepower	8 to 10
165 to 225 hp. gasoline, per horsepower	10 to 12
250 hp. gasoline, per horsepower	15 to 20
300 hp. gasoline, per horsepower	22 to 25
180 to 240 hp. Diesel, per horsepower	25 to 35
300 to 400 hp. Diesel, per horsepower	40 to 50

With both gasoline and Diesel engines available in comparable sizes, but at different prices, it naturally follows that the Diesel electric locomotive will cost more than the corresponding size of gasoline-electric. Either type has higher first cost than a corresponding steam locomotive. However, the better pulling power, ability to do more work, and smoother accelerating characteristics of the gasoline- or oil-electric locomotive enable it to start a heavier load than a steam locomotive of the same weight so that either type can be 25 to 30 per cent lighter in weight than a steam locomotive of corresponding tractive effort. If the first cost of an up-to-date steam locomotive is taken at 100 per cent, the approximate first-cost ratios of the other types are as follows:

65-ton steam	100
50-ton gasoline-electric	145 to 160
50-ton Diesel-electric, two engines	195 to 210
50-ton Diesel-electric, single engine	200 to 215

It appears that the internal-combustion-engine type of locomotive must show great savings over a steam locomotive to overcome the handicap in first cost. Both gasoline- and Diesel-electrics do this without question. One company operating a 35-ton gasoline-electric in 24-hour-a-day service in place of a 50-ton steam locomotive reports $8.60 per day saved, including fixed charges.

Another company reports that their 45-ton gasoline-electrics show, among other savings, the elimination of about $9 a day as the cost for keeping up steam during off-turns for the steam locomotives.

A 70-ton Diesel-electric in railroad switching service has given a saving of at least $40 per 24-hour day over steam switching. In heavy-duty mill switching, certain 60-ton 300-hp. Diesel-electrics operate at a cost of $2.93 per locomotive hour, including fixed charges, as compared to $5.10 per hour for steam locomotives. In the smaller sizes a 12-ton gasoline-electric has been operating at between one-half and two-thirds of the cost of the replaced steam locomotive. At another plant one 15-ton gasoline-electric has replaced two steam locomotives. These examples are typical, although costs and savings will vary over wide ranges, depending on the service. Accounting methods also play an important part in fixing the costs and savings.

In the final analysis, the economies of the respective types of gasoline-electric, Diesel-electric, mechanical-drive, or combination battery-Diesel will usually determine the type best suited for a given application. For example, a gasoline-electric in the same service as a corresponding Diesel-electric will use roughly twice as many gallons of fuel, and the gasoline may cost three times as much per gallon as the fuel oil. The gasoline-electric locomotive will therefore have higher fuel expense than the Diesel-electric, but against this must be set the lower fixed charge of the gas-electric and its other operating expenses which tend to be lower, such as maintenance. Since the service fixes the fuel consumption, and therefore sets the fuel expense, the total operating cost of the gas-electric will exceed the Diesel-electric when the service requires gasoline at the rate of 6 or 7 gal., or more, per locomotive hour.

It may be seen therefore that there is available a wide range of internal-combustion-engine type of locomotives using electric drives. These locomotives have been proved in industrial service, with satisfactory results. The use of this up-to-date motive power has so many advantages over the steam engine and shows such attractive savings that it well merits the attention of every user of industrial locomotives.

MH-53-10

Types of Materials Used in Materials-Handling Equipment

By HERVEY J. SKINNER,[1] BOSTON, MASS.

Materials-handling equipment means to most people a conveyor for carrying solids, but the author defines it also as a pump for handling liquids or a steel cylinder for transporting gases. The material used in the construction of such equipment must meet a wide variety of conditions. The author discusses wood, concrete, glass, rubber, copper, lead, nickel, aluminum, and iron and steel. The fundamental research initiated within the last ten years by the large steel companies furnishes a foundation which few other industries possess for the production of many new alloys.



MATERIALS-handling equipment in its broadest sense includes any apparatus in which material, whether solid, liquid, or gaseous, is contained or by which it is moved from one location to another. In the minds of many people, materials-handling equipment means some sort of a container or conveyor for solids, although a pump for handling liquids or a steel cylinder for transporting highly compressed gases is just as truly a piece of materials-handling equipment.

The material used in the construction of such equipment must meet a wide variety of conditions. In practically all cases tensile strength and resistance to abrasion or wear are essential. In addition there are many instances where ability to resist heat and corrosion and other special characteristics are necessary.

In the construction of materials-handling equipment the basic materials available are comparatively few, although there are many ramifications of these materials. The basic construction materials which may be mentioned are wood, concrete, metals, glass, rubber, and ceramic materials. During the past 10 or 15 years there has been a remarkable increase in the kinds of material from which equipment is constructed. Many of these may be termed freak constructions and have proved impractical, but others have won a well-deserved place in equipment manufacture. Rubber-lined tanks and glass piping are examples of equipment that only a few years ago would have been regarded as visionary.

Wood

Wood finds many uses in any plant. Its use for tanks is well established and familiar to plant engineers. Wooden piping, however, is not so well known. Wood has certain advantages over iron and steel. Its first cost is low and its maintenance almost nothing. It is a non-conductor of heat, and is not subject to corrosion. Steel tanks require paint both inside and out at more or less frequent intervals. While wooden tanks are often painted on the outside for sake of appearance, it is generally considered better not to paint them, since it does not help to preserve them. Wood absorbs a certain amount of water, and with the outside of the tank sealed with a coating of paint, the evaporation of this absorbed water, which may cause deterioration, is retarded since the wood cannot breathe. Being a non-conductor, the contents of a wooden tank will remain cool for a longer period in summer, and in winter are better protected from the frost. Although wood tanks in exposed locations may require a heating system, the heating unit need be only about one-third the size required for a steel tank of the same size.

Wood pipe is made in two types, one consisting of a barrel of California redwood staves bound together by individual steel bands or spirally wound wire, and the other a bored pipe made by boring through the center of a piece of straight-grained clear redwood. The former type is built in sizes ranging from 2 in. to 15 ft. in diameter and will withstand static-pressure heads up to 350 ft. Wood pipe has good flow qualities, since it has no lap joints or connection irregularities. It has proved its value in many cases where conditions require pipe which will resist acid and other corrosive influences, and in the disposal of certain types of gases which are destructive to ordinary flue materials.

Concrete

The use of concrete should be mentioned, although its limitations are well known. It has advantages for certain types of apparatus where strength is desired, but it is bulky as compared with steel. It is affected by acids and is not wholly resistant to alkalis or even water. Alkalis have a slow deteriorating action, the concrete losing its strength and gradually disintegrating. Water exerts a slight solvent action on the lime of the cement, precipitating iron from the water which is deposited on the surface of the concrete. It is believed that this coating of iron helps preserve the concrete from further deterioration.

When it is advisable to use concrete for acid or alkaline conditions, the best treatment is to apply a protective coating of tar or asphalt, but these are not always permanent and usually cannot be reapplied. Various materials have been tried in the concrete mix for preventing corrosion, but their use has not been successful, and while sometimes a harder surface more resistant to abrasion is obtained, they do not increase appreciably the resistance of the concrete to corrosive agencies.

Lean concrete deteriorates more rapidly than concrete rich in cement on account of its greater porosity and permeability. In the construction of cement-lined pipe a very rich mortar is used which is practically watertight. For this reason solvent action is reduced to a minimum, and cement-lined pipe has been in use for many years with marked success. The amount of lime dissolved by the water is slight and will cause no appreciable trouble where there is a reasonable flow. Instances are on record,

---

[1] President, Skinner & Sherman, Inc. Mr. Skinner's firm of consulting chemical engineers is engaged in research and development problems for various industries. He is technical adviser to the Associated Industries of Massachusetts, a member of the American Institute of Chemical Engineers, the American Water Works Association, and other organizations.

Presented at the "Three-M" (Management, Materials, Maintenance) Congress held in Cleveland, Ohio, April 13 to 17, 1931, under the auspices of the A.S.M.E. and the S.I.E.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

however, where water has stood in cement-lined pipes and has become sufficiently hard to be objectionable for many purposes.

Another form in which concrete pipe is becoming industrially important in the handling of liquids, particularly water and sewage, is centrifugally cast reinforced-concrete pipe. This method of production gives the concrete a smooth, even appearance which reduces friction to a minimum.

Glass

Glass is a most striking example of the development of a familiar material whose application has been extended to new industrial uses. The manufacture of mechanically strong glass, chiefly of the borosilicate type, is responsible for this extension in the use of glass. "Pyrex" glass made by the Corning Glass Works is a glass of this type. It has a very low expansion coefficient, which renders it resistant to sudden changes of temperature. It also has high mechanical strength so that it is capable of withstanding sudden shocks due to rough handling. Its high surface hardness, which is only three points below that of the diamond, and its exceptionally smooth surface cause it to resist certain kinds of abrasion better than steel.

"Pyrex" glass tubing and piping are used extensively in many industries and are available in both flanged and bell-and-spigot joints. Flanged pipe can be obtained up to 3 in. in diameter and socket pipe up to 12 in. in diameter. The smooth surface minimizes friction loss, and the resistance of the glass to chemical attack prevents contamination of product. The transparency of glass is a valuable feature as it allows full visual evidence and control of flow or movement inside the piping, and also enables one to observe the surface cleanliness.

Considerable research has been carried on by the Corning Glass Works with a view to improving the earlier methods of making joints with glass piping. The first installations were made with bell-and-spigot joints, using cements, or with straight pipe runs, using rubber-hose connections. The limitations of these types of joint led to the flanged type with a thin gasket and split metal flanges for clamping together the glass flanges.

This method has proved very satisfactory where a pipe line is installed permanently and where it is important that the liquid handled has a minimum contact with anything other than glass. The food industries have welcomed this addition to the types of piping available for their purposes, and one of the foremost beverage plants in the country has replaced all of its pipe lines with the "Pyrex" glass-flange type.

More recently a new type of joint has been developed by mechanically shaping the ends of plain tubing so that the bore remains constant but the external diameters increase. In other words, the ends of the pipes are cones, with the bore of the tube extending through the cone and the base of the cone forming the end face of the tubes. These are clamped together with a specially designed metal clamp which is easy to assemble.

Glass tubing finds wide application in heat-exchanging operations, particularly where corrosive substances have to be handled, one factory alone using 18 miles of "Pyrex" piping. Although the actual heat transmission is greater through metals than through glass, it has been found that the permanently smooth surface of glass counterbalances the greater heat transfer of metal, and that after a few months of use the heat-transfer values of glass and metal not only approach each other, but that there is an advantage in favor of glass. It has been demonstrated that metal tubes lose some of the initial efficiency because of internal roughening, due to corrosion, and also because the external surface in contact with the cooling liquid becomes roughened and coated.

As a result of these investigations, glass-tube heat exchangers are becoming quite a well-known piece of equipment, not only in the chemical industries, but in the public-utility field, as an equipment well suited to the handling of ammonia liquors resulting from the operation of gas plants.

Another interesting application of glass is as a lining for tanks and similar equipment. "Glascote," made by the Glascote Company, has positive adherence to steel and is consequently durable and resistant to physical shocks. Its coefficient of expansion is the same as that of steel, so that it stands up under quick changes of temperature, and its composition makes it resistant to chemical action. Completely glass-lined mixing tanks, reaction vessels, and other equipment, with glass-coated agitators and coils, are available and used in many industries where cleanliness and the prevention of corrosion are essential considerations.

The Pfaudler Company also builds acid-resisting glass-lined steel equipment for various purposes, including glass-lined pipe and fittings. Freight cars equipped with glass-lined tanks are commonly used for the transportation of milk and other products.

Rubber

Rubber is another material whose field of usefulness has been greatly extended by comparatively recent inventions. Rubber is chemically inert and is resistant to the action of most corrosive solutions. Hard-rubber piping and fittings have been in use for a long time. More recently its use has been extended to larger pieces of apparatus and to the lining of metal equipment. The "Ace" process of the American Hard Rubber Company interposes a relatively thin layer of soft rubber between the metal surface and the hard-rubber lining, and bonds the latter securely to the metal.

The "Vulcalock" process of the B. F. Goodrich Rubber Company is another advance in the use of rubber in materials-handling equipment. By this process it is possible to unite rubber to metal, wood, concrete, and other materials. The bond is so strong that the rubber will often tear before it can be separated from the metal surface to which it is applied. The Vulcalock process can be used with the most elastic grades of soft rubber as well as with flexible hard rubber. More than one hundred steel tank cars lined with rubber by this process are in use for the transportation of acids and other corrosive liquids.

Rubber-lined tanks, piping, valves, and other fittings are in general use. Rubber-covered fans in which the entire wheel shaft and interior of the housing are covered with rubber find special application in ventilating systems where corrosive gases and fumes have to be handled.

"Armorite," another Goodrich product, is a rubber compound developed for high resistance to abrasion. In certain types of abrasive service it has been found to outwear ten times its thickness of hard steel. It is particularly valuable in lining chutes, hoppers, and spouts that are subject to abrasive wear. Rubber-coated screen wire for the screening of sand and gravel and other abrasive material finds useful application. These rubber linings can be applied to concrete tanks, aluminum, stainless steel, and certain compositions of brass. Its life is almost indefinite so long as the tank is intact. It can be used at temperatures up to 150 deg. fahr., but will deteriorate when exposed to oils, particularly mineral oils.

Metals

The metals most commonly used in materials-handling equipment are iron, copper, nickel, aluminum, lead, tin, and zinc and their various alloys. With the advance in our knowledge of the science of applied metallurgy, many other elements have been added to these basic metals to obtain desirable properties. As such additive metals may be mentioned chromium, nickel,

manganese, tungsten, vanadium, molybdenum, bismuth, and cadmium.

Copper

Copper is one of the oldest metals and its uses are well known. Its advantages are due to a combination of characteristics. The metal is easy to fabricate and can be made in almost any form required. It has a considerable degree of resistance to a limited variety of corrosive reactions. As the metal itself, it is used mainly in sheets for lining or covering equipment made of wood or other materials, and in the form of special apparatus.

The alloys of copper with other metals are numerous and find many important applications. The various brasses and bronzes, which are alloys of copper, zinc, lead, and tin are used extensively for piping, valves, and other fittings under conditions which are too corrosive for iron or steel. "Admiralty Metal," an alloy of copper, zinc, and tin, was developed especially for resistance to sea-water corrosion and is used generally for condenser tubes in which the cooling medium is salt water.

Among other bronzes developed especially to withstand corrosion may be mentioned "Aterite," a nickel-brass, "Meco Metal," a copper-nickel-zinc alloy, and "Ampco," an aluminum-bronze.

Lead

Lead in the form of pure metal finds many uses, although they are for the most part confined to the lining of tanks, pumps, piping, etc. It is used extensively where sulphuric acid enters into any chemical process. Its purity is of great importance, and unless it is highly pure its resistivity is of a low order. It lacks the characteristic of physical strength, which is sometimes offset by the addition of other alloying metals like antimony, with a corresponding reduction in its resistive properties.

Lead-lined iron pipe has found many uses for conveying liquids which are corrosive to iron. "Crawlproof sheet lead," made by the National Lead Company, is one of the newest developments in sheet lead, and is a chemical lead sheet reinforced inside with antimonal-lead supporting bars to prevent buckling and wrinkling.

Nickel

In certain specialized equipment, nickel has been found advantageous. It is both rustproof and highly resistant to corrosion, and its hard, non-porous surface is easy to clean and to keep clean. It finds considerable application in the food industry and in the manufacture of pharmaceuticals. Nickel tubing is used extensively in these industries as well as others for heating and cooling purposes.

Of the alloys of nickel, Monel metal is perhaps the most important. This is an alloy of nickel and copper with high nickel content. It has unusual strength and toughness which, together with its non-corrosiveness, makes it an ideal metal of construction in the food and dairy industries. Laundries have found it of value because of its rustproof qualities, and dyehouses are using Monel-metal equipment on account of its being unaffected by the dye liquors. It is used in the manufacture of machine parts such as gears and pinions, and because of its elasticity and toughness it is well adapted to the weaving process of making wire cloth.

Nickel is used extensively for plating purposes. It is easily deposited on iron, takes a light polish, and keeps it well, although it requires repolishing from time to time. A thin skin of chromium deposited on top of the nickel gives a light nickel-colored surface of extreme hardness, so hard in fact that it cannot be polished. Chromium is not so extremely hard in itself, but the combination which it forms with hydrogen during the plating process gives the hardest metal coating of which we know. The chromium layer need be only 0.000017 in. in thickness to give a coating that resists both rust and corrosion. The chromium-plating industry has grown enormously, and chromium-plating plants have been established in every important center of the country.

Aluminum

Aluminum is in a class by itself on account of its lightness. It is very sensitive to the action of alkalis, but is resistant to the action of many acids. Its resistance to corrosion is dependent on its purity, very slight amounts of certain impurities having a marked influence on its properties. It is available as sheet metal, castings, forgings, structural shapes, bars, rods, and tubing.

The heat treatment of aluminum alloys has multiplied its use and has given it the ductility and strength of mild steel.

The use of aluminum alloys has already opened up a wide field. "Duralumin," an alloy with copper, manganese, and magnesium, has found many valuable applications where lightness and strength are desired. It gave the Germans the material for the construction of airplanes during the latter part of the war, and it is now used for the framework of dirigible airships.

This alloy suffers from intercrystalline corrosion, and the attempts to overcome this defect met defeat until a coating of pure aluminum was sprayed on the surface by the Schoop process described below. Accelerated tests over a period of a few years indicate the efficiency of this coating to protect against intercrystalline corrosion. As a result of these experiments with coating, it is possible to cast a duplex ingot containing "Duralumin" in the interior and pure aluminum on the outside. This duplex ingot, known as "Alclad," can be rolled and worked into desired shapes.

The addition of a small proportion of magnesium to aluminum gives an alloy, "Magnalium," that is almost as light as aluminum and almost as strong as steel. An alloy of aluminum and calcium is lighter and harder than aluminum and more resistant to corrosion.

In connection with aluminum, mention should be made of the "calorizing" process patented by the General Electric Company whereby aluminum is driven into the surface of another metal, usually mild steel, to form an aluminum alloy. This alloy is immune to oxidation up to a temperature of 1650 deg. fahr., and completely resists corrosive reactions. Calorized steel pots provide durable containers for salt and lead baths, and the alloy has been very effective in the petroleum industry in protecting tubes in oil-cracking stills.

Metal Coating

Non-ferrous metals for coating other metals are made use of extensively in equipment for the handling of material. These may be applied by immersion in molten metal, known as hot dipping, by electroplating or by metal spraying. Tin is usually applied by the hot-dipping process. A sheet of iron dipped in melted tin comes out coated with a thin layer of the latter metal and furnishes the material so universally used as tin cans. If the tin is scratched, the iron beneath rusts more rapidly than if the tin were not there, on account of the electrolytic action which is set up, and iron being the negative element of the couple suffers at the expense of the tin.

With zinc, the opposite is the case, since it is negative toward iron, and when the two are in contact and exposed to corrosion, the zinc oxides first and the iron remains intact until the zinc has entirely dissolved. Ordinary galvanized iron is usually made by hot dipping in zinc, although the latter may also be applied electrolytically. "Sherardizing" consists in tumbling the articles to be coated in a drum with zinc dust which

is heated to 800 deg. fahr. The zinc at this temperature combines with the iron and forms a series of alloys, ranging from pure zinc on the top to pure iron at the bottom of the coating. This gives an efficient coating, since even if cracks appear, the iron is more or less protected so long as zinc remains. Electro deposits of zinc are apt to be more porous.

A more recent development of coating metals is the "Schoop" process, by which a wire of zinc or other metal is fed into an oxy-hydrogen air blast of such heat and power that the molten metal is projected with the speed of bullets, and any object which is subjected to this metallic spray may be coated with a deposit as thick as desired. A distinct advantage of this metal-spraying process is that it can be applied to the finished article even after the latter is in place, since the spraying apparatus is easily portable. A long list of metals can be successfully sprayed.

Iron and Steel

Iron and steel, of course, find the greatest use in the construction of equipment for the handling of material. Pig iron, the base of all iron and steel products, is the product obtained in the blast furnace from the crude iron ores. The essential properties of iron and steel depend upon the amount of certain other elements present, such as carbon, phosphorus, silicon, manganese, and sulphur. Cast iron contains roughly 93 per cent iron, and the other elements mentioned make up the remainder. It is essentially an alloy of iron, carbon, and silicon. It has relatively little strength, and cannot be used where strength or ductility is an essential property.

The treatment of cast iron to produce iron and steel products in useful form consists essentially in varying the properties of the elements other than iron and also in obtaining these elements in proper combinations. Malleable iron, for example, is remelted cast iron subjected to a heat treatment so that the carbon is uniformly distributed throughout the metal in the form of rosettes. This gives to the iron good machinability, fair strength, and fair ductility. Wrought iron is the name given to commercial iron made by refining the pig iron at a temperature so low that it is obtained in a pasty condition which permits the slag formed during the operation to be uniformly distributed throughout the mass. The process of making steel consists of several operations carried out in such a manner that the product is free from slag and contains the necessary modifying elements in proportions to produce the characteristics desired.

In recent years many new steel products have been developed which will withstand heat and corrosion to a marked degree. Corrosion is, as is well known, a serious factor, and an immense amount of research work has been carried on looking to the development of products which will resist it. Anything like an accurate estimate of the loss resulting from the corrosion of metals in common use is obviously out of the question. It has been estimated that the annual replacement of iron and steel products alone due to corrosion may reach as high as 2 per cent of the total tonnage in use. Assuming that there are 500,000,000 tons of iron and steel products in the United States, the loss would amount to 10,000,000 tons, or something over 30 per cent of our annual production of iron and steel products.

Chemically pure iron is not a commercial product. In fact, comparatively few people have ever seen the pure metal, which is soft, ductile, and white like silver. Commercially pure iron, however, containing less than approximately 0.16 per cent of impurities, is available. It is useful for structural work and other purposes where atmospheric conditions are severe. "Armco Iron," made by the American Rolling Mill Company, is an example of commercially pure iron.

High-silicon cast iron is resistant to the corrosive action of most acids and is extensively used for large pieces of apparatus. "Duriron," made by the Duriron Company, and "Tantiron," made by the Bethlehem Foundry and Machine Company, are examples of these ferro-silicon alloys. They contain 14 to 15 per cent of silicon, and the balance is iron with small amounts of carbon, manganese, sulphur, and phosphorus. In order to obtain non-corrosive properties it has been necessary to sacrifice toughness, but the composition is balanced so as to produce a metal with the best acid-resisting qualities, combined with a moderate amount of transverse strength and resistance to shock. These alloys are silvery white, close-grained iron, very hard and rather brittle. They are available only in castings, being too hard to machine with ordinary cutting tools, but may be finished by grinding into shape with some of the newer abrasives. They can be remelted by the oxyacetylene flame, although careful preheating and after-cooling of the welded parts are essential. They are rustproof and will withstand the action of most of the common acids except muriatic. Standard equipment in the shape of pipes and fittings, pans, chutes, kettles, pumps, valves, and cocks are available, and it is also possible to produce large castings for large-scale production.

Stainless steel and rustless iron are meeting a variety of uses. The element which is responsible for the stainless properties is chromium, when present in percentages between 12 and 15 per cent. The stainless steels usually contain 0.3 to 0.4 per cent of carbon and, in order to obtain the property of stainleasness, special heat treatment is required. Heating to a bright red heat, followed by rapid quenching, produces an alloy of iron and chromium which retains in solution the carbides of these two metals, thereby preventing an otherwise corrosive action. When special resistance to deterioration is required, the content of chromium may be increased to 20 or even 30 per cent.

Stainless steel found its first wide application in cutlery, where the advantages of cleanliness immediately established the steel in that field. More recently it has come into use where other corrosive conditions must be met, such as for exhaust valves in gas engines, piston rods for steam engines, and pumps for acid mine waters. Steels with the lower carbon contents have the best stainless properties, although they are not so hard as those with a high carbon content. These steels when suitably chosen for the particular purpose in view are practically immune to rusting and are very resistant to mild-acid corrosion. Stainless steel is available in bar stock in a wide variety of sizes and sections. Forged stainless pipes may also be secured in sizes beyond the limits of the ordinary piercing mills.

Stainless alloys low in carbon, which are not hardened, are often termed "rustless iron." They find wide application in sheets and in drawn and stamped articles. The sheets are annealed to be as soft as possible, as high ductility is more important than mechanical strength. Tank cars made of rustless iron are being used in the shipment of nitric acid.

Closely allied to the chromium steels but quite distinct are the chrome-nickel steels and alloys. The earliest use of chrome-nickel steels was for armor plate, and for many years these were the only alloy steels which were widely used. Later they came into general use for automotive steels, and are now used for all kinds of construction. The typical chrome-nickel steel will run from 0.60 to 1.50 per cent of chromium and 1.25 to 3.50 per cent of nickel, with 0.3 to 0.5 per cent of carbon.

When chromium and nickel are used in larger percentages, very tough stainless steels are obtained which possess high resistance against corrosion and remarkable ductility. Typical of these alloys are those containing 18 per cent of chromium and 8 per cent of nickel, made under the Krupp Nirosta patents.

Several manufacturers are licensed to make these Nirosta KA alloys. "Rezistal," made by the Crucible Steel Company, "Standardalloy," made by the Standard Alloy Company, Inc.,

and "Enduro," made by the Republic Steel Company, are typical examples. "Allegheny Metal," made by the Allegheny Steel Company, is another alloy of substantially the same composition These alloys are ductile and malleable, and are twice as strong as mild steel. They may be drawn, stamped, machined, cast, or welded. The corrosion-resisting properties are superior to any other ferrous alloy available in commercial quantities. Another unique feature is their ability to take a polish to a mirror-like surface which because of its rustproof properties is permanent, as distinguished from a plated surface which will wear or peel off. Although these alloys are expensive, the first cost sinks into insignificance when compared to the expense of replacement made necessary by rust and corrosion.

The application of these alloys has been most diversified, and in some cases the question has been raised as to whether their utilization by industry has not followed too closely on the heels of the metallurgist without sufficient tests as to durability over a period of time. This is particularly true where parts made from these alloys are used under high-pressure conditions.

The chemical industry has been quick to appreciate the advantages of these alloys, and they are found in nitric acid plants, in ammonia plants, and in many other cases where they are used for piping, condensers, heat exchangers, and other purposes. In the petroleum-oil industry they are used extensively in cracking stills and other parts subject to corrosion at elevated temperatures. The non-tarnishing properties make them ideal for the food and dairy industries and for kitchen equipment in hotels, restaurants, and hospitals. Laundries find them particularly valuable as they are not affected by bleach or other chemicals which are sometimes used.

More recently the possibility of these alloys for architectural and ornamental purposes has been recognized. In this connection it is of interest to note that the top of the Chrysler Building in New York is fabricated from these chrome-nickel alloys. In the new Empire State Building, "Allegheny Metal" is used, not only for ornamental purposes and for the dirigible mooring mast, but for mullions and window trim.

This brief survey of some of the many types of materials available for materials-handling equipment and the advances which have been made in comparatively recent years may indicate the immense range of possibilities which lie ahead.

It is only within the past decade or so that the large steel companies have undertaken fundamental research, or research looking toward the development of new products on a large scale. Heretofore the research of the steel companies has been confined more or less to improving methods of analysis, and the very accurate and rapid methods which have been developed whereby it is possible to determine carbon, manganese, silicon, phosphorus, sulphur, and other elements to a hundredth or a thousandth of a per cent are a tribute to the steel industry. They give the steel companies a foundation for further research which probably no other industry possesses, since the effect of even traces of these other elements is most marked, and unless methods are available for their determination, many false steps are easily possible in any research program. Such a foundation will surely be productive of many new steel alloys.

Great developments may be expected from the science of metallurgy. Of the more than fifty metals known, not half of them are in common use. The number of possible combinations of metals in alloys is infinite, and it is impossible to predict what properties may be obtained with new combinations and with some of the newer and rarer elements. Tantalum, for example, the metal used in incandescent lamps before the tungsten filament was developed, is now being produced by the Fansteel Products Company and is just coming into use for the lining of equipment. It is as hard as cold-rolled steel, yet ductile and workable, and inert to practically all corrosion.

Columbium, the sister-metal to tantalum, is awaiting development. It is a beautiful white metal with properties similar to tantalum, except that its density is only one-half as great. Beryllium is another metal with interesting possibilities. It is lighter, much harder, and four times as elastic as aluminum.

There can be little doubt that the future holds in store much greater advances than have been witnessed in the past few years, and it behooves every plant engineer, as well as others in charge of materials-handling equipment, to follow the progress of developments so that he can apply them as soon as they have been demonstrated to be practical.

Suspended Cables With Large Sags

By ROBERT C. STRACHAN,[1] NEW YORK, N. Y.

The curve assumed by a cable having its own weight and the applied load uniformly distributed over its length differs but little from a parabola if the deflection or sag is small; but as the actual curve is a catenary, modified more or less by the elastic deformation of the cable, the difference between the two curves becomes too great to be disregarded when, in the case of suspended structures other than suspension bridges, the sag exceeds a certain limit. The catenary being an exponential curve, its solution presents unavoidable difficulties. As a means of overcoming these in great measure, the author has compiled a table of hyperbolic radians tabulated as angular functions, whose use he shows by means of several worked-out examples.



IN THE design of cables for suspended structures other than bridges, the conditions encountered make it sometimes necessary to allow the cable to deflect or sag a greater distance below the supports than that which is suitable in suspension bridges.

The curve assumed by a cable having its own weight and the applied load uniformly distributed over its length differs but little from a parabola if the deflection is small; but as the actual curve is a catenary, modified more or less by the elastic deformation of the cable, the difference between the parabola and the actual curve becomes too great to be disregarded when the sag exceeds a certain limit.

The catenary is an exponential curve, and its solution therefore presents unavoidable difficulties. As a means of overcoming these difficulties in great measure, Table 1 (see Addenda) has been compiled. By its use, as there explained and illustrated, the angle A, in degrees, corresponding to a hyperbolic ratio u may be found without difficulty, and hyperbolic functions of u are trigonometric functions of A.

The equation of the catenary referred to an origin situated outside the curve on the axis of symmetry, at a distance a from the vertex, is

$$y = a \cosh\left(\frac{x}{a}\right) \quad \ldots\ldots [1]$$

The distance a is the parameter. The ratio x/a is a measure denoting hyperbolic radians or "hyps," designated by the symbol u, and the angle of slope of the curve at any point is the Gudermannian angle corresponding to u for that point. If the supports are on a level and L is the span, x becomes $\frac{1}{2} L$ at either support, and therefore for those points

$$u = \frac{L}{2a} \quad \ldots\ldots [2][2]$$

and A, the angle made by the terminal tangent with the x-axis, is

$$A = \text{gd } u \quad \ldots\ldots [3]$$

With u known, A may be taken from the table, or vice versa.

The horizontal component H of the tension at any point of the curve is equal to wa, where w is the unit weight. Hence, denoting the direct tension at the support by T,

$$H = \frac{Lw}{2u} \quad \ldots\ldots [4]$$

$$T = H \sec A \quad \ldots\ldots [5]$$

If a limiting value of H is fixed, u for any span may be calculated by Equation [4], thus determining all the remaining elements of the curve. Otherwise, u and A must usually be obtained by successive approximation and interpolation. An illustration of this process follows.

Let the sag v be 0.2 L; then

$$\frac{\text{exsec } A}{2u} = \frac{v}{L} \quad \ldots\ldots [6]$$

Trial angles are assumed and the exsecant and u taken from tables. The guide for finding a trial angle is that tan A is a little more than $4v/L$. When two angles sufficiently close together have been found to bracket v/L, the desired functions may be obtained by interpolating between them, with results as below:

	exsec	u	exsec/2u	
39°–58′	0.30477	0.76215	0.19994	
			0.20000	= v/L
39°–59′	0.30509	0.76253	0.20005	

From which:

$$A = 39°\text{–}58.55'$$
$$\sec A = 1.30494$$
$$u = 0.76236 \text{ hyps}$$

For the assumed case, therefore, L and w being known, Equations [4] and [5] give

$$H = \frac{Lw}{2 \times 0.76236} \qquad T = 1.30494H$$

and for C, the length of the curve,

$$C = L \frac{\tan A}{u} \quad \ldots\ldots [7]$$

or

$$C = \frac{L \times 0.83838}{0.76236}$$

If C and L are known, a close approximation is obtained by the following formula in which q represents C/L:

$$u = \sqrt{(120q - 20)^{1/2} - 10} \quad \ldots\ldots [8]$$

After finding A corresponding to u, the remaining elements may be calculated by Equations [4], [5], and [6].

[1] Consulting Engineer. Mem. Am.Soc.C.E. From 1900 to 1927 Mr. Strachan was identified with the design and construction of New York City bridges, buildings, and tunnel and subway structures, first as engineer of design in the various departments having control of bridges, tunnels, and transportation and later as assistant chief engineer of the Department of Public Markets. His work comprised both original constructions and alterations on steel and reinforced concrete buildings as well as cantilever, movable and suspension bridges, and tunnel design. In 1927–1928 he was engineer of terminals for the Port of New York Authority.

[2] See Addenda for this and other formulas cited.

From Equation [7], $q = \tan A/u$, and the above approximate u may be corrected by bracketing as illustrated, using tan A instead of exsec A. The results found with Equation [8], however, are sufficiently close for many purposes, with sags as large as $0.25L$.

If the curve C and sag v are known, approximations are unnecessary, as the parameter may be found directly by the formula

$$a = \frac{\frac{1}{4}C^2 - v^2}{2v} \quad \ldots\ldots\ldots\ldots\ldots\ldots [9]$$

and A is found by

$$\tan A = \frac{\frac{1}{2}C}{a} \text{ or exsec } A = \frac{v}{a}$$

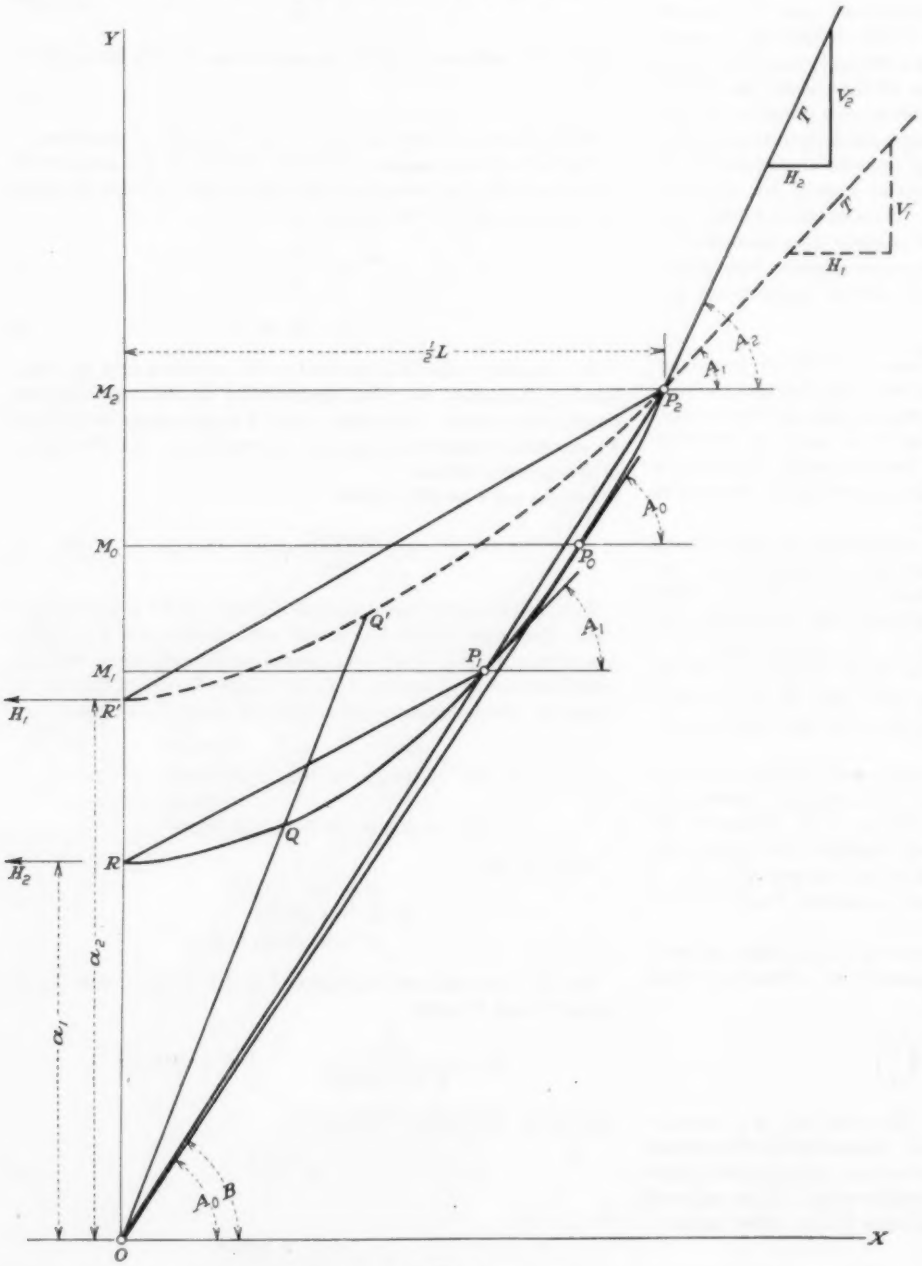


FIG. 1

OP_0 is tangent to catenary; OP_2 is secant.
$u_1 = M_1P_1/a_1 = L/2a_1$; $u_2 = L/2a_1$
$u_0 = M_0P_0/a_1 = 1.19968$ hyps
$A_1 = \text{gd } u_1$; $A_2 = \text{gd } u_2$; $A_0 = \text{gd } u_0 = 56° 27' 57''$.
cosec $A_0 = 1.19968$
$\cosh u_1/u_1 = \cosh u_2/u_2 = \tan B$
w = unit weight of cable and load
$H_1 = wa_2 = Lw/2u_1$; $H_2 = wa_1 = Lw/2u_2$
$V_1 = w \times \overline{P_2Q'R'} = Lw \tan A_1/2u_1$
$V_2 = w \times \overline{P_2P_1QR} = Lw \tan A_2/2u_2$
$T = w \times \overline{OM_2} = \frac{1}{2}Lw \tan B = \sqrt{H_1^2 + V_1^2} = \sqrt{H_2^2 + V_2^2}$
T is minimum when $B = A_0$; $T_{min} = 0.75444\,Lw$

After finding u corresponding to A, all other elements are then given by Equations [2], [4], and [5].

As T is necessarily greater than H in every case, it is often preferable to fix a limit for T rather than for H. Referring to [4] and [5], sec A = cosh u, therefore

$$T = \tfrac{1}{2}Lw\frac{\cosh u}{u} \quad \ldots [10]$$

and for a given span and tension the ratio cosh u/u is determined.

In Fig. 1 it is seen that a line drawn from the origin O at an angle A_0 with the x-axis is tangent to the curve at P_0, and any other line from O at an angle B greater than A_1 and less than 90 deg. cuts the curve at two points, P_1 and P_2. From the similar triangles it is evident that the condition

$$\frac{\cosh u}{u} = \tan B$$

is fulfilled at both P_1 and P_2, and

$$\frac{\cosh u_1}{u_1} = \frac{\cosh u_2}{u_2} = \tan B. \quad [11]$$

Drawing P_2R' parallel to P_1R and constructing the curve $P_2Q'R'$ so that for any point thereof

$$\frac{OQ}{OQ'} = \frac{OP_1}{OP_2}$$

we have two arcs with respective sags M_2R' and M_2R, the same span L, and the same tension T; for these two curves,

$$H_1 = \frac{Lw}{2u_1}; \quad H_2 = \frac{Lw}{2u_2};$$

$$T = \tfrac{1}{2}Lw \tan B \ldots\ldots [12]$$

The lengths of the curves are

$$C_1 = \frac{L \tan A_1}{u_1}; \quad C_2 = \frac{L \tan A_2}{u_2}$$

$$\ldots\ldots [13]$$

Both u_1 and u_2 may be found when tan B is known. As noted on Fig. 1, for the point of tangency,

$$u_0 = 1.19968 \text{ hyps}$$
$$u_1 < u_0 \text{ and } u_2 > u_0$$

Trial angles may be used to bracket the value of tan B, which is equal to both sec A_1/u_1 and sec A_2/u_2.

Another way is to take a limited number of terms of the series resulting from the expansion of Equation [11], replacing u_1 and u_2 by x. If powers higher than the sixth are neglected we shall have the following, after arranging the terms in the form of the general equation:

$$x^6 + 30x^4 + 360x^2 - 720x \tan B + 720 = 0 \ldots . \quad [14]$$

This equation has two positive roots, one of which is u_1 and the other u_2, and by well-known methods both may be evaluated to any desired number of decimal places.

Horner's method is especially adaptable. The requisite number of terms in [14] will depend upon the desired degree of precision for the final results. If the roots are carried to one or two places only, these approximations may be used as guides in bracketing.

With suitable modifications all the methods outlined will apply to spans in which the supports are at different elevations.

The Addenda following comprise statements of the relations of elements and the derivation of formulas.

Addenda

THE fundamental relations between the ratio u expressed in hyberbolic radians (hyps) and its corresponding angle A in degrees, are, as shown with Table 1,

$$\sin A = \tanh u \qquad \cos A = \operatorname{sech} u$$
$$\tan A = \sinh u \qquad \cot A = \operatorname{cosech} u$$
$$\sec A = \cosh u \qquad \operatorname{cosec} A = \coth u$$

The angle A is known as the Gudermannian angle in recognition of the mathematician Gudermann who first demonstrated its properties; and the symbol "gd," analogous to "sin," "cos," "sinh," "cosh," etc., is used to express this fact; thus:

$$A = \text{gd } u$$

and following the analogy of inverse functions,

$$u = \text{gd}^{-1} A$$

In a catenary with supports on a level, let

L = span
C = length of curve
v = deflection or sag
A = angle made by terminal tangent with horizontal
a = parameter of the curve.

TABLE 1 HYPERBOLIC RADIANS TABULATED AS ANGULAR FUNCTIONS[1]

Explanation of Table. If A, an angle in degrees, is the Gudermannian angle corresponding to a ratio u, expressed in hyperbolic radians (hyps), their relation, indicated thus,

$$A = \text{gd } u, \text{ or } u = \text{gd}^{-1} A$$

is such that

$$\sin A = \tanh u \qquad \cos A = \operatorname{sech} u$$
$$\tan A = \sinh u \qquad \cot A = \operatorname{cosech} u$$
$$\sec A = \cosh u \qquad \operatorname{cosec} A = \coth u$$

The table gives u in hyps for angles at 10-min. intervals, and intermediate values are to be derived by interpolation in the usual way. The angle A for any value of u having been found, all the hyperbolic functions of u may be taken out of trigonometric tables.

EXAMPLE 1. Given, $u = 0.84532$ hyps. Required, tanh u.
From table the next lower $u = 0.84482 = \text{gd}^{-1}\, 43° 30'$ (Diff. = 40.2 per min.)

$50 \div 40.2 = 1\frac{1}{4}'$

$\text{gd } u = 43° 31\frac{1}{4}' = A$

$\tanh u = \sin 43° 31\frac{1}{4}' = 0.68862$ Q. E. F.

EXAMPLE 2. Given, coth $u = 1.07405$. Required, u.
$\operatorname{cosec} A = 1.07405 \quad A = 68° 36'$
From table, $\text{gd}^{-1}\, 68° 30' = 1.66149$ (Diff. = 79.6 per min.)
79.6×6 min. $= 478$

$\text{gd}^{-1}\, 68° 36' = 1.66627 = u$ Q. E. F.

EXAMPLE 3. Given, chord of parabola $L = 200$ ft. Sag $v = 60$ ft. Required, exact length of arc $= C$.
Inclination of terminal tangent $= A$.

$\therefore \tan A = 4v/L = 1.2. \quad \therefore A = 50° 11.7'$
From table, $\text{gd}^{-1}\, 50° 10' = 1.01521$ (Diff. = 45.5 per min.)
45.5×1.7 min. $= 77$

$1.01598 = \text{gd}^{-1} A$

$0.84665 = \text{gd}^{-1} A/1.2$
$1.56205 = \sec A$

$2.40870 =$ coefficient of $L/2$.

$$C = \frac{L}{2}\left(\frac{\text{gd}^{-1} A}{\tan A} + \sec A\right)$$
$$= \frac{200}{2} \times 2.40870 = 240.87 \text{ ft.} \quad \text{Q. E. F.}$$

[1] Copyright, 1930, by R. C. Strachan.

Deg.	0′	10′	20′	30′	40′	50′	Diff. per min.
0	0	0.00291	0.00582	0.00873	0.01163	0.01454	29.1
1	0.01745	0.02036	0.02327	0.02618	0.02909	0.03200	....
2	0.03491	0.03782	0.04074	0.04365	0.04656	0.04947	....
3	0.05238	0.05529	0.05820	0.06112	0.06403	0.06695	....
4	0.06987	0.07278	0.07570	0.07862	0.08154	0.08446	29.2
5	0.08738	0.09030	0.09322	0.09614	0.09906	0.10198	....
6	0.10491	0.10783	0.11076	0.11369	0.11662	0.11955	29.3
7	0.12248	0.12541	0.12834	0.13128	0.13421	0.13715	....
8	0.14008	0.14302	0.14596	0.14890	0.15184	0.15478	29.4
9	0.15773	0.16067	0.16362	0.16657	0.16952	0.17247	29.5
10	0.17543	0.17838	0.18133	0.18429	0.18725	0.19021	29.6
11	0.19318	0.19614	0.19910	0.20207	0.20504	0.20801	29.7
12	0.21099	0.21396	0.21694	0.21992	0.22290	0.22588	29.8
13	0.22886	0.23185	0.23484	0.23783	0.24082	0.24381	29.9
14	0.24681	0.24981	0.25281	0.25582	0.25882	0.26183	30.0
15	0.26484	0.26785	0.27087	0.27389	0.27691	0.27993	30.2
16	0.28295	0.28597	0.28900	0.29204	0.29508	0.29812	30.4
17	0.30116	0.30420	0.30724	0.31029	0.31334	0.31640	30.5
18	0.31946	0.32252	0.32558	0.32865	0.33171	0.33478	30.6
19	0.33786	0.34094	0.34402	0.34711	0.35020	0.35329	30.8
20	0.35638	0.35948	0.36258	0.36568	0.36878	0.37189	31.0
21	0.37501	0.37813	0.38125	0.38438	0.38751	0.39064	31.3
22	0.39377	0.39690	0.40005	0.40320	0.40635	0.40950	31.5
23	0.41266	0.41582	0.41899	0.42216	0.42533	0.42851	31.7
24	0.43169	0.43487	0.43807	0.44127	0.44447	0.44767	32.0
25	0.45088	0.45409	0.45730	0.46052	0.46374	0.46697	32.3
26	0.47021	0.47345	0.47669	0.47994	0.48319	0.48645	32.5
27	0.48972	0.49299	0.49626	0.49953	0.50281	0.50610	32.7
28	0.50939	0.51269	0.51599	0.51930	0.52261	0.52593	33.1
29	0.52925	0.53258	0.53591	0.53925	0.54260	0.54595	33.4
30	0.54931	0.55267	0.55604	0.55941	0.56279	0.56617	33.7
31	0.56956	0.57296	0.57636	0.57977	0.58318	0.58660	34.1
32	0.59003	0.59346	0.59690	0.60035	0.60380	0.60726	34.5
33	0.61073	0.61420	0.61768	0.62116	0.62465	0.62815	34.9
34	0.63166	0.63517	0.63869	0.64222	0.64575	0.64929	35.3
35	0.65284	0.65639	0.65995	0.66352	0.66710	0.67069	35.8
36	0.67428	0.67788	0.68149	0.68510	0.68872	0.69235	36.2
37	0.69599	0.69964	0.70329	0.70695	0.71062	0.71430	36.7
38	0.71799	0.72169	0.72539	0.72910	0.73282	0.73655	37.2
39	0.74029	0.74404	0.74780	0.75156	0.75533	0.75911	37.7
40	0.76291	0.76671	0.77052	0.77434	0.77817	0.78201	38.3
41	0.78586	0.78972	0.79359	0.79747	0.80136	0.80526	38.9
42	0.80917	0.81309	0.81702	0.82096	0.82491	0.82887	39.5
43	0.83284	0.83682	0.84081	0.84482	0.84884	0.85287	40.2
44	0.85690	0.86095	0.86502	0.86909	0.87317	0.87727	40.8
45	0.88138	0.88550	0.88963	0.89377	0.89792	0.90209	41.5
46	0.90628	0.91048	0.91469	0.91890	0.92313	0.92737	42.3
47	0.93163	0.93590	0.94019	0.94449	0.94880	0.95313	43.0
48	0.95747	0.96182	0.96619	0.97057	0.97497	0.97938	44.0
49	0.98381	0.98825	0.99271	0.99718	1.00166	1.00616	45.0
50	1.01068	1.01521	1.01976	1.02433	1.02891	1.03351	46.0
51	1.03812	1.04275	1.04740	1.05207	1.05675	1.06145	47.0
52	1.06616	1.07089	1.07564	1.08042	1.08521	1.09001	48.0
53	1.09483	1.09967	1.10453	1.10942	1.11432	1.11924	49.0
54	1.12418	1.12914	1.13411	1.13911	1.14413	1.14917	50.0
55	1.15423	1.15931	1.16441	1.16954	1.17469	1.17986	51.5
56	1.18505	1.19027	1.19551	1.20076	1.20604	1.21135	53.0
57	1.21668	1.22203	1.22741	1.23281	1.23823	1.24368	54.0
58	1.24916	1.25466	1.26019	1.26574	1.27132	1.27693	56.0
59	1.28256	1.28822	1.29391	1.29963	1.30538	1.31116	57.5
60	1.31696	1.32279	1.32865	1.33454	1.34046	1.34641	59.5
61	1.35240	1.35842	1.36447	1.37055	1.37666	1.38281	61.5
62	1.38899	1.39520	1.40145	1.40773	1.41405	1.42040	63.5
63	1.42679	1.43322	1.43968	1.44618	1.45272	1.45930	65.8
64	1.46591	1.47256	1.47926	1.48600	1.49278	1.49960	68.2
65	1.50646	1.51336	1.52031	1.52730	1.53434	1.54142	70.8
66	1.54855	1.55573	1.56295	1.57022	1.57753	1.58490	73.7
67	1.59232	1.59979	1.60731	1.61489	1.62252	1.63020	76.8
68	1.63794	1.64573	1.65358	1.66149	1.66945	1.67748	80.3
69	1.68557	1.69372	1.70193	1.71020	1.71853	1.72693	84.0
70	1.73541	1.74395	1.75256	1.76124	1.76999	1.77881	88.2
71	1.78771	1.79668	1.80573	1.81486	1.82407	1.83336	92.9
72	1.84273	1.85219	1.86173	1.87136	1.88107	1.89088	98.1
73	1.90079	1.91079	1.92088	1.93107	1.94136	1.95176	104.0
74	1.96226	1.97286	1.98358	1.99441	2.00535	2.01641	110.6
75	2.02759	2.03889	2.05032	2.06187	2.07355	2.08537	118.2
76	2.09732	2.10942	2.12166	2.13404	2.14658	2.15927	126.9
77	2.17212	2.18514	2.19832	2.21167	2.22520	2.23891	137.1
78	2.25280	2.26689	2.28118	2.29566	2.31036	2.32527	149.1
79	2.34040	2.35576	2.37136	2.38720	2.40328	2.41963	163.5
80	2.43625	2.45314	2.47032	2.48779	2.50557	2.52366	180.9
81	2.54209	2.56086	2.57998	2.59947	2.61935	2.63962	202.7
82	2.66031	2.68143	2.70300	2.72504	2.74758	2.77063	.....
83	2.79422	2.81838	2.84313	2.86850	2.89453	2.92125	.....
84	2.94870	2.97693	3.00596	3.03586	3.06668	3.09847	.....
85	3.13130	3.16524	3.20038	3.23679	3.27456	3.31382	.....
86	3.35467	3.39727	3.44175	3.48830	3.53712	3.58844	.....
87	3.64253	3.69972	3.76037	3.82493	3.89394	3.96807	.....
88	4.04813	4.13515	4.23048	4.33585	4.45365	4.58719	.....
89	4.74135	4.92368	5.14683	5.43451	5.83998	6.53313	.....
90	Inf.	.....	.....	.....	.....	.....	.....

(1) Referring to Fig. 2,

If $x = \frac{1}{2} L$, $\frac{x}{a} = \frac{L}{2a}$

$$\frac{L}{2a} = u = \text{gd}^{-1} A$$

$$\frac{C}{2a} = \sinh u = \tan A$$

$$\therefore \frac{C}{L} = \frac{\sinh u}{u} = \frac{\tan A}{u}$$

$$\frac{v}{a} = \cosh u - 1 = \text{exsec } A$$

$$\therefore \frac{v}{L} = \frac{v}{2au} = \frac{\text{exsec } A}{2u}$$

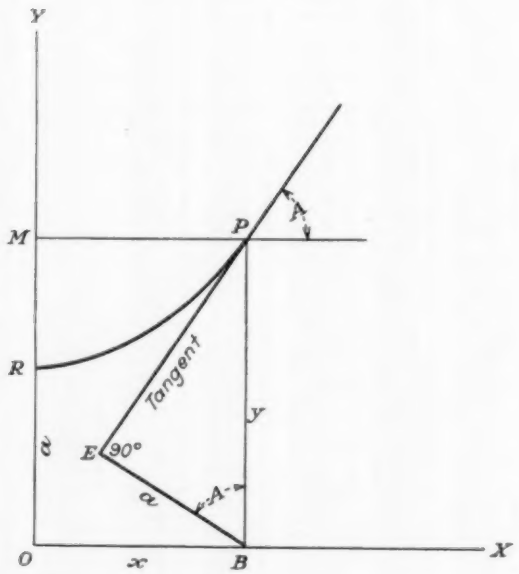


FIG. 2

Arc $\overline{RP}$ is a catenary with origin O.
$x/a = u$; $A = \text{gd } u$; $u = \log_e \tan (\frac{1}{2}A + 45°)$
$y/a = \cosh u$; $\overline{RP}/a = \sinh u$; $\overline{RM}/a = \cosh u - 1$
$EP = \text{arc } \overline{RP}$; $EB = a$
$a^2 = y^2 - \overline{EP}^2 = (a + \overline{RM})^2 - \overline{RP}^2$
$\therefore a = (\overline{RP}^2 - \overline{RM}^2)/2\overline{RM}$

(2) Equation [8] is derived as follows:

Let $q = \frac{C}{L} = \frac{\sinh u}{u}$

$$qu = \sinh u = u + \frac{u^3}{6} + \frac{u^5}{120} + \text{etc.}$$

$$q = 1 + \frac{u^2}{6} + \frac{u^4}{120} + \text{etc.}$$

Neglecting powers above the fourth,

$$u^4 + 20u^2 + 120 = 120q$$

hence

$$u = \sqrt{(120q - 20)^{1/2} - 10}$$

(3) Equation [14] is derived as follows:

$$\tan B = \frac{\cosh u}{u}$$

$$\therefore u \tan B = \cosh u = 1 + \frac{u^2}{2} + \frac{u^4}{24} + \frac{u^6}{720} + \text{etc.}$$

Neglecting powers above the sixth and reducing,

$$u^6 + 30u^4 + 360u^2 - 720u \tan B + 720 = 0$$

which becomes Equation [14] by writing x instead of u.

(4) *Point of Tangency in Fig. 1.* If from any point P on the curve a tangent and an ordinate be drawn to intersect the

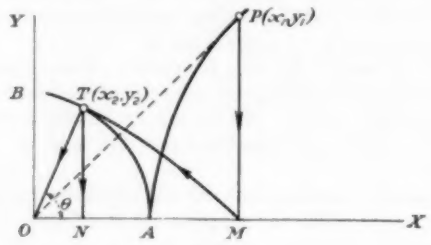


FIG. 3

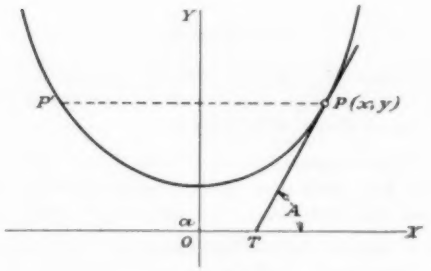


FIG. 4

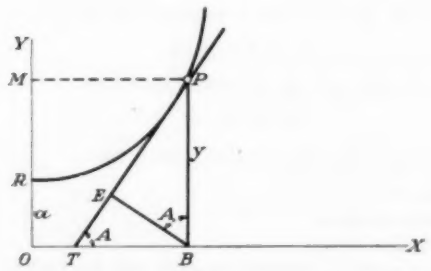


FIG. 5

x-axis, the distance between the intersections is coth u when the parameter is taken as the unit. For the point P_0 the tangent passes through the origin, hence in this case

$$u = \coth u = \text{cosec } A$$

By successive approximation and bracketing, this condition is found to be fulfilled when

$$u = 1.19968 \text{ hyps}$$

$$A = 56° 27' 57''$$

(5) *Length of Parabolic Arc as Stated in Explanation of Table 1.* A well-known formula is

$$\frac{C}{L} = \frac{\sqrt{1 + 16k^2}}{2} + \frac{\log_e \left(4k + \sqrt{1 + 16k^2}\right)}{8k}$$

Where $k = \frac{v}{L}$ $\therefore 4k = \tan A$

If $u = \text{gd}^{-1}A$, $\tan A = \sinh u$, and $\sqrt{1 + 16k^2} = \sec A = \cosh u$

$$\therefore \log_e (4k + \sqrt{1 + 16k^2}) = \log_e (\sinh u + \cosh u) = u$$

Hence $$C = \frac{L}{2}\left(\frac{\text{gd}^{-1}A}{\tan A} + \sec A\right)$$

Discussion

R. B. DALE.[3] The author presents an interesting and valuable mathematical analysis of the catenary curve. It is doubtful whether the true catenary may be found in practical engineering structures; in any event its occurrence is very infrequent. The author recognizes this fact in his statement, somewhat casually made, that ". . . the actual curve is a catenary, modified more or less by the elastic deformation of the cable. . . ." This fact deserves to be emphasized more fully.

The mathematical analysis of the catenary curve given in this paper properly belongs to the mechanics of rigid bodies. For application to practical structures, the effect of the elasticity of the material of the cable should not be neglected. The actual cable will stretch or deflect a very considerable amount, particularly with large sags, if the structure is loaded to the economic value of the material. The theoretical catenary curve for which this analysis is made then no longer exists in the actual structure. Two methods are ordinarily used to determine the elastic effect. The first and more approximate method is called the "elastic theory," and the second and more exact, the "deflection theory." These methods have been worked out quite fully in the case of suspension bridges. Discrepancies in the application of these two theories to practical structures are of a nature to indicate that deflections cannot safely be ignored.

Dr. Arvid H. Baker in his thesis entitled "Suspension-Bridge Analysis by the Exact Method Simplified by Knowledge of Its Relations to the Approximate Method,"[4] appropriately states as follows: "In work of this nature involving several days of labor to properly establish even a minor relationship, one is prone to draw conclusions from meager data and fail to state the restrictions to which the conclusions are subject, thus very often leading astray those who might accept the conclusions as the truth." Dr. Baker further points out that the discrepancies between the "elastic theory" and the "deflection theory" as applied to suspension bridges are very considerable, and he offers correction factors by which the labor of determining stresses is greatly reduced and simplified.

It would seem desirable for the author to continue his praiseworthy efforts by adding to his contribution an analysis of the catenary as modified by the more exact "deflection theory" as applied to a specific structure.

In this discussion the writer is joined by his associate, Mr. J. E. Thompson, who offers several interesting notes with reference to the mathematical theory of the paper.

J. E. THOMPSON.[5] The Gudermannian angle, function, and notation occur in connection with the hyperbola instead of the catenary, but as the hyperbolic cosine (cosh) is defined in connection with the hyperbola and the graph of the cosh is a catenary, there is a relation between the Gudermannian angle and the catenary. This relation is briefly outlined here, and a few notes are added on the equations of the paper.

Referring to Fig. 3, the arc ATB is a quadrant of the circle $x^2 + y^2 = a^2$, of the radius $OA = a$. The curve AP is part of the rectangular hyperbola $x^2 - y^2 = a^2$, of hyperbolic "radius" $OA = a$.

Through any point $P(x_1, y_1)$ on the hyperbola erect the ordinate $PM = y$, from M draw the tangent MT to the circle T, draw the radius $OT = a$, making the angle $MOT = \theta$, and erect $TN = y_2$.

If h is the area of the hyperbolic sector $OAPO$, then the ratio $2h/a^2$ is called the "hyperbolic angle" U. That is,

$$\frac{2h}{a^2} = U \qquad [15]$$

The circular angle θ is said to "correspond" to the hyperbolic angle U and is defined as the "Gudermannian angle" of U (from the mathematician Gudermann, who first made extensive use of its properties). This relation is written

$$\theta = \text{gd}\, U \qquad [16]$$

From the figure,

$$\overline{MT}^2 = \overline{MN}^2 + \overline{NT}^2 = (x_1 - x_2)^2 + y_2^2$$

and

$$\overline{OM}^2 = \overline{OT}^2 + \overline{MT}^2, \text{ or } x_1^2 = a^2 + \overline{MT}^2$$
$$\therefore x_1^2 = a^2 + (x_1 - x_2)^2 + y^2$$
$$= a^2 + x_1^2 - 2x_1x_2 + x_2^2 + y_2^2$$

But from the circle,

$$x_2^2 + y_2^2 = a^2$$

$$\therefore x_1^2 = a^2 + x_1^2 - 2x_1x_2 + a^2$$

whence

$$x_1x_2 = a^2$$

and

$$\frac{x_1}{a} = \frac{a}{x_2} \qquad [17]$$

This relation is also obtained directly from the right triangle OTM and its altitude TN.

In the right triangle TON, $\frac{TO}{ON} = \frac{a}{x_2} = \sec\theta$. Hence by [17],

$$\frac{x_1}{a} = \sec\theta \qquad [18]$$

which is also obtained directly from the triangle OTM.

Now in the circle ATB, by definition

$$\frac{y_2}{a} = \sin\theta, \qquad \frac{x_2}{a} = \cos\theta$$

are the circular sine and cosine of the circular angle θ.

Analogously the definitions of the hyperbolic sine and cosine of the "hyperbolic angle" U of Equations [15] are, in terms of the coordinates and hyperbolic "radius" a,

$$\left.\begin{aligned}\frac{y_1}{a} &= \sinh U\\ \frac{x_1}{a} &= \cosh U\end{aligned}\right\} \qquad [19]$$

Comparing [18] and [19], it is seen that

$$\sec\theta = \cosh U \qquad [20]$$

when the circular angle θ is the Gudermannian of the hyperbolic

[3] Head, Department of Mechanical Drawing and Machine Design, Pratt Institute, Brooklyn, N. Y. Mem. A.S.M.E.

[4] Engineering and Science Series, No. 24, Rensselaer Polytechnic Institute.

[5] Instructor, Department of Mathematics, Pratt Institute, Brooklyn, N. Y.

angle U, the corresponding circle and hyperbola being related as shown in the figure.

The expression [20] is the third of the first set of six relations given in the Addenda to the paper, A and u being there used instead of θ and U.

From the ordinary trigonometric relations among the circular functions and the analogous relations among the hyperbolic functions, using Equation [20] as the fundamental relation, the other relations of the set are readily obtained.

Note particularly the first of these six relations, namely,

$$\tan A = \sinh u \quad [21]$$

Use will be made of this relation later.

Consider now the graph of the equation

$$y = a \cosh \left(\frac{x}{a}\right) \quad [22]$$

(Note that x, y, and a here are not the x, y, and a of the hyperbola discussed above.)

This curve is shown in Fig. 4 and is called the catenary. It is the shape taken by a chain or a perfectly flexible but inelastic heavy cord hung between two supports whose distance apart is less than the length of the cord, which is thus allowed to sag but does not stretch.

The variables x, y of Equation [22] are the coordinates of any point P on the curve and a is called the parameter of the curve. Geometrically, it is the y-intercept of the graph.

At the point P, draw the tangent PT to the catenary, making the angle $XOP = A$. The slope of the curve at P is then

$$m = \tan A \quad [23]$$

In order to find the slope of the curve analytically, differentiate its Equation [22], a being constant. This gives

$$m = \frac{dy}{dx} = \sinh \left(\frac{x}{a}\right) \quad [24]$$

If now we call x/a the "hyperbolic angle" u, so as to be able to use tables of hyperbolic functions in plotting, then Equations [22] and [24] can be written

$$y = a \cosh u \quad [25]$$
$$m = \sinh u \quad [26]$$

Comparing [26] and [23], $\tan A = \sinh u$, which is the same as [21].

This result brings out the relation of the Gudermannian to the catenary, and by virtue of this result and the definition relations [16] and [17], we can write

$$A = \text{gd } u \quad [27]$$

where

$$u = \left(\frac{x}{a}\right) \quad [28]$$

Equation [27] is Equation [3] of the paper.

If $P(x, y)$ and P' in the figure are the points of support of the catenary, then $P'P$ is the span L of the catenary suspension. But $P'P = 2x$. Therefore $2x = L$, or $x = L/2$ and

$$\frac{x}{a} = \frac{L}{2a}$$

or, by [28],

$$u = \frac{L}{2a} \quad [29]$$

This is Equation [2] of the paper.

If in Fig. 4 of the catenary, the ordinate $PB = y$ be erected and BE be drawn perpendicular to the tangent PT, the result is as shown in Fig. 5.

Here the angle BPT is the complement of the angle PTB, and also of the angle PBE. Hence the angle $PBE = A$, as shown in Fig. 5.

In the right triangle PBE, $y/EB = \sec A$. Hence, by Equation [20] (or the third of the set of six relations in the paper, $\sec A = \cosh u$),

$$\frac{y}{EB} = \cosh u$$

Also, by Equation [25], $y/a = \cosh u$

$$\therefore \overline{EB} = a$$

as shown in Fig. 5. This figure corresponds to Fig. 2 of the paper.

By the usual formula for the length of a curve, it is found from Equation [25] that the catenary arc

$$\overline{RP} = a \sinh u \quad [30]$$

Also $\overline{RM} = \overline{OM} - \overline{OR} = y - a = a \cosh u - a$.

$$\therefore \overline{RM} = a(\cosh u - 1) \quad [31]$$

In the right triangle PBE, $\overline{EP}/a = \tan A$. Hence by [27],

$$\frac{\overline{EP}}{a} = \sinh u$$

whence

$$\overline{EP} = a \sinh u$$

Comparing this with [30]

$$\overline{EP} = \text{arc } RP \quad [32]$$

Relations [30], [31], and [32] are those used by the author in the analysis of his Fig. 2. His analysis of Fig. 1 in his paper is self-explanatory.

The remaining formulas and equations and the table of the paper are analytical results derived from the results given in these notes, by the ordinary theory of hyperbolic functions.

Author's Closure

In the discussion by Professors Dale and Thompson the suggestion is made that the effects of elastic deformation should be emphasized more fully. The author recognizes the importance of these effects, and the necessity of making the most careful calculation of their amounts in the case of actual structures exceeding moderate dimensions. However, the article was prepared more especially to show that it is possible to simplify the laborious procedure almost always entailed when we discard the assumptions commonly made for the deeply sagging curves often necessary in transmission-line cables and similar constructions.

Any one who has gone through the operations necessary to the solution of a problem involving hyperbolic functions applied to a structure of specific dimensions realizes why approximations so often take the place of theoretically exact methods in such work.

The author hopes that his Table 1, coordinating hyperbolic with trigonometric functions, may be of practical value when applied to either short or long spans in which the sag-span ratio is great. Its field of usefulness is not affected by the factor of elastic deformation, the importance of which depends upon actual dimensions and other construction details.

MSP-53-1

Case-Hardening With Ammonia Gas

By V. O. HOMERBERG[1] and J. P. WALSTED,[2] CAMBRIDGE, MASS.

It has been the dream of metallurgists for many years to produce a steel of good toughness with maximum surface hardness and wear resistance. To produce these properties with a minimum of distortion has been the subject of much research work of a high order. In the past the most successful method of obtaining these properties has been by subjecting low-carbon steel to the action of a carburizing material at high temperatures. Under these conditions the steel takes up carbon in its surface layer, changing that portion of the material into a high-carbon steel capable of hardening, while the core remains low in carbon with the usual toughness and shock-resisting properties. By a suitable double heat-treatment the core can be brought to maximum physical properties while the case can be made hard and wear-resistant. This process of case-hardening by means of carbon is somewhat cumbersome. The necessary heat-treatment after carburizing is a frequent cause of distortion of the hardened parts.

A new process of case-hardening has been developed which has all the advantages of the older method. Besides being free from the troublesome warpage it surpasses the older method in the hardness of the case produced. The classic research of Doctor Fry at the Krupp Works, Essen, Germany, is responsible for the introduction of this valuable metallurgical development. Researches of a very high order in France and in this country have brought the process to its present high state of development. A great deal of credit is due Dr. Norton and Mr. De Fries of the Ludlum Steel Company for the excellent work which they did immediately after the introduction of this process into this country several years ago. This new process, known as "nitriding," now a commercial success, is described by the authors in this paper.

THE PROCESS of case-hardening by means of ammonia, "nitriding," is remarkable for its simplicity. The parts to be hardened are placed in a heated furnace, subjected to the action of ammonia gas for a definite time, and then removed. The articles, as they come from the furnace, are hard and require no further treatment. This is a distinct advantage over the older case-hardening methods.



J. P. Walsted



V. O. Homerberg

The ammonia used in the process is the commercial anhydrous material commonly used in refrigeration. A simple modification of fittings makes it possible to use the commercial product with convenience. The ammonia on passing over the steel breaks down to some extent into nitrogen and hydrogen; the nitrogen in the nascent condition readily entering the steel. The gases passing out of the furnace consist of hydrogen, unabsorbed nitrogen, and undissociated ammonia. The exit ammonia may be conducted to the outdoors or readily dissolved in water and drained into the sewer. As these gases are the only waste products the problem of waste disposal is very simple.

The temperature of nitriding is low compared to other heat-treating processes. This allows the parts to be heat-treated before nitriding without danger of subsequent destruction of desirable physical properties. Since the nitriding leaves a clean surface free from scale a convenient feature of the process is the fact that heat-treating precedes the ammonia hardening. Parts coming from the nitriding furnace are ready for service without further expenditure of work. Moreover, the low temperature removes all danger of warpage during case-hardening, provided that all strains have been previously removed.

The hardness obtained by nitriding is remarkable. The resistance to wear, made possible by this treatment, if intelligently applied will undoubtedly result in great economic saving. The resistance to corrosion shown by nitrided steels has been known for some time. The success of the nitriding, while it does not solve the problem of corrosion, does eliminate this important factor in many instances.

The investigation conducted by Dr. Fry began with a consideration of the combining power of various alloying elements toward nitrogen and the stability of the resulting nitrides. He found that aluminum ranked first, both as to the amount of nitrogen with which it combined as well as the stability of the aluminum nitride formed. This compound showed no loss in nitrogen even when heated at 1832 deg. fahr. Other elements which form nitrides with high nitrogen contents include molybdenum, manganese, chromium, vanadium, titanium, and tungsten.

Plain carbon steels, various alloy steels, including those containing aluminum, and aluminum together with various other elements such as chromium and molybdenum, were subjected to the action of ammonia gas under definite conditions and the resulting surface hardness was determined in each instance. The outcome of this investigation by Dr. Fry was the manufacture of a series of alloy steels containing varying amounts of carbon, together with approximately 1 to 1.25 per cent aluminum, 1.5 per cent chromium, and 0.2 per cent molybdenum. Alloy steels having this particular combination of alloying elements were chosen because they showed the maximum surface hardness when exposed to ammonia gas under definite conditions. Furthermore, this hardness was retained at elevated temperatures and the specimens showed a marked resistance to atmospheric,

---

[1] Associate Professor of Physical Metallurgy at the Massachusetts Institute of Technology and Consulting Metallurgist to the Ludlum Steel Co.

[2] Instructor in Physical Metallurgy at the Massachusetts Institute of Technology.

Presented at the Fourth National Machine Shop Practice Meeting, Chicago, Ill., Sept. 22 to 24, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

water, and salt-water corrosion. These steels are marketed under the name of "Nitralloy," the different ranges of carbon content being designated as grades "G," "H," etc.

Table 1 gives the analyses and Figs. 1 and 2 show the physical properties of two of the Nitralloy steels after proper heat-treatment and before nitriding.

The presence of molybdenum tends to toughen both the core and the case resulting from the ammonia treatment. Particular attention is called to the high impact values which indicate that these steels have high shock-resisting properties.

TABLE 1 ANALYSES OF NITRALLOY "G" AND "H"

Nitralloy	G	H
Carbon	0.36	0.23
Manganese	0.51	0.51
Silicon	0.27	0.20
Aluminum	1.23	1.20
Chromium	1.49	1.58
Molybdenum	0.18	0.20
Sulphur	0.010	0.011
Phosphorus	0.013	0.011

the steel to form nitrides. These nitrides are very likely in a fine state of dispersion in the case, and impart the extreme hardness to the surface of the steel, a hardness which gradually

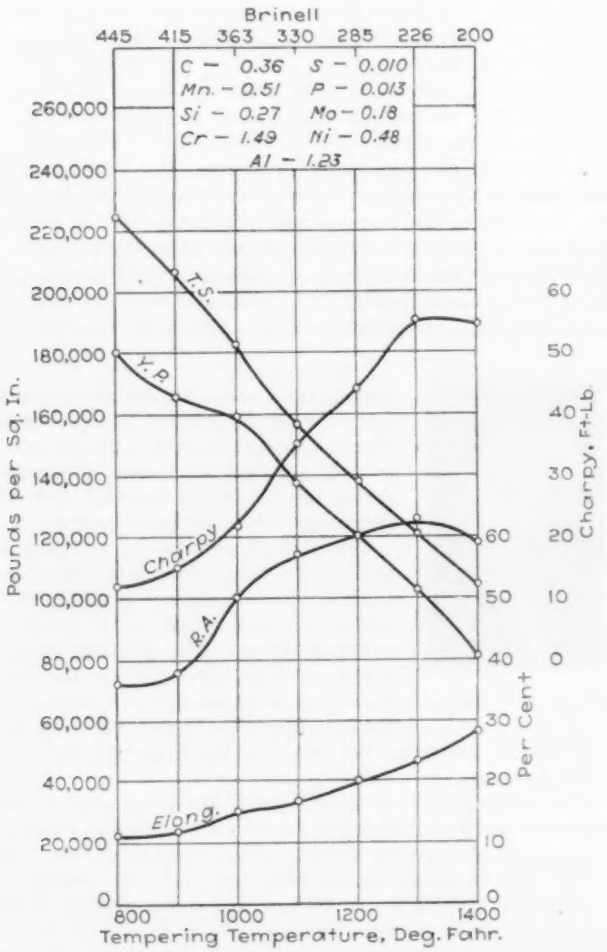


FIG. 1 PHYSICAL PROPERTIES OF NITRALLOY "G" AFTER HEAT-TREATMENT AND BEFORE NITRIDING

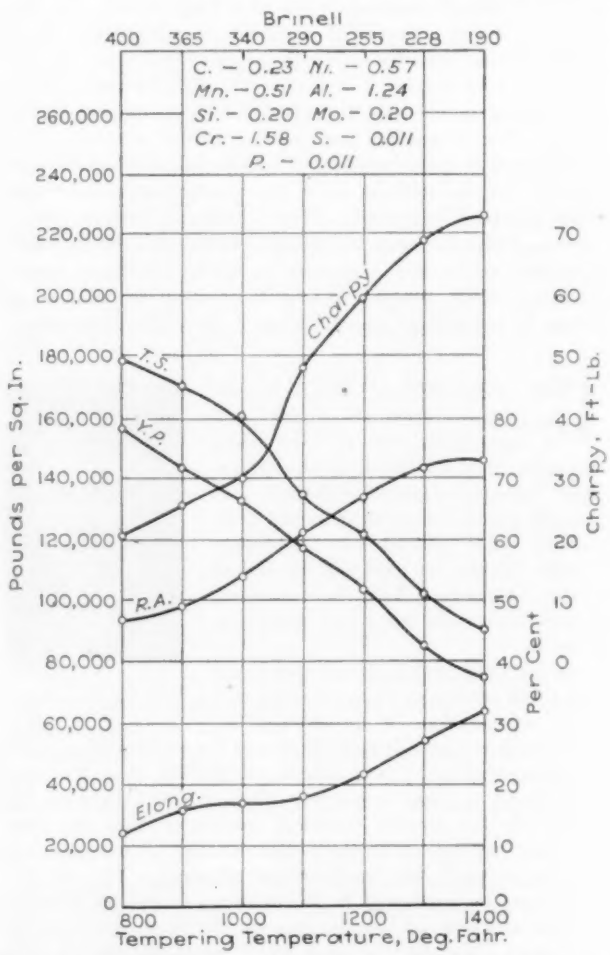


FIG. 2 PHYSICAL PROPERTIES OF NITRALLOY "H" AFTER HEAT-TREATMENT AND BEFORE NITRIDING

The equipment required for the case-hardening of articles with ammonia consists of a suitable furnace provided with automatic temperature control, an air-tight container, which should be made of a material not susceptible to ammonia hardening, for holding the articles to be nitrided, a tank of ammonia provided with a suitable needle valve for controlling the flow of the gas and, finally, a special pipette for determining the extent of the ammonia dissociation.

Special furnaces for nitriding are now being made by a number of furnace manufacturers. Some of these furnaces are provided with a fan in each instance for insuring active gas circulation. The ammonia gas decomposes to a certain extent into nitrogen and hydrogen. The nitrogen, which is very active at the moment of decomposition of the ammonia gas, undoubtedly combines to a certain extent with the iron and with the alloying elements in decreases inwardly until it corresponds to that of the core.

The time of exposure of the articles to the action of the ammonia gas at the nitriding temperature varies from 2 to 90 hr., depending on the service to which they are to be subjected. No heat-treatment is given to the parts after nitriding, the articles being cooled in the box, generally to room temperature. The nitriding temperature generally varies from 900 to 1000 deg. fahr., 975 deg. fahr. being commonly used.

The necessity for the complete removal of strains set up in forging, machining, and in hardening in order to prevent warping or distortion during nitriding has been emphasized. Straightening of parts after nitriding may be required if this precaution is not observed. This is especially likely to occur in thin flat sections or in parts having a large diameter at one end and a long section with a smaller diameter at the other end, such as in cer-

tain types of spindles, fine pump plungers, etc. It has been found quite possible to straighten such parts even when done cold. Since the hardness of the case is not affected at 1000 deg. fahr., it is obvious that the straightening operation can be well applied to the hot material.

Great emphasis should be given to the necessity for the complete removal of the decarburized layer before nitriding. If this precaution is not observed, the nitrided case will lend itself to chipping and flaking. The photomicrograph, Fig. 3, shows the appearance of the case resulting from the nitriding of a decarburized specimen. The black areas represent voids. Sufficient material must be left for final machining or grinding before nitriding to insure the complete removal of all traces of decarburization resulting from the forging and the heat-treating operations.

A typical procedure to follow in the case of an article machined from an annealed bar is to rough-machine, heat-treat, finish-machine, and then nitride. The heat-treatment involves quenching in oil from the proper hardening temperature followed by tempering at a temperature which is at least as high as that used in nitriding in order to obtain proper physical properties for the core and to relieve machining and hardening strains. Sufficient time should be allowed for this operation in order to prevent warping or distortion on nitriding. The final machining or grinding must remove all traces of decarburization. Nitriding for the proper time to meet the conditions completes the operation, except for a possible finishing operation, such as lapping, on the nitrided article. The treatment of forgings should be considered with due precaution to eliminate decarburization before case-hardening with ammonia.

Photomicrographs, Figs. 4 and 5, show the microstructure of Nitralloy "G" nitrided at 975 and 1200 deg. fahr., respectively, for 48 hr. It will be noted that the depth of case as the result of nitriding at 1200 deg. fahr. is much greater than that obtained by nitriding at 975 deg. fahr. However, a marked decrease in the hardness accompanies this increase in case depth. The curve shown in Fig. 6 indicate the results obtained by making the hardness determinations on tapered specimens. In order to take advantage of the greater case depth obtained at the higher nitriding temperature, and of the greater hardness when nitrided at 975 deg. fahr., a double cycle of nitriding is frequently advocated in order to curtail the time of exposure by nitriding first at 1200 deg. fahr. for a suitable length of time, followed by nitriding at 975 deg. fahr. to obtain the desired surface hardness. Nitriding at 975 deg. fahr. followed by a short time exposure at 1200 deg. fahr. has been recommended also in order to produce a tougher case than when 975 deg. fahr. is used alone, although the surface hardness of the article treated in this manner will be equal to or only slightly less than that obtained at 975 deg. fahr.

TABLE 2 GROWTH OF CYLINDERS ON NITRIDING

Nitriding temperature, deg. fahr.	Increase in length, in.	Increase in diameter, in.
900	0.001	0.0003
950	0.002	0.0009
1000	0.003	0.0011

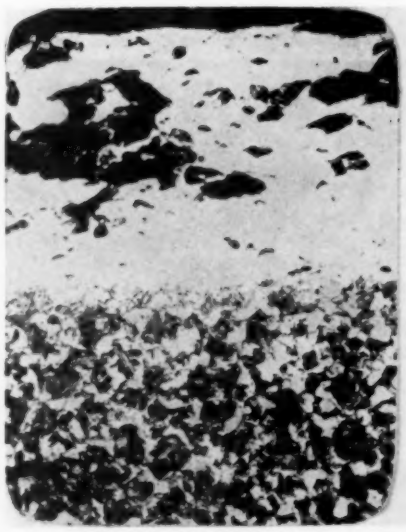


FIG. 3 NITRALLOY "G" DECARBURIZED THEN NITRIDED. NITAL ETCH, MAG. × 100

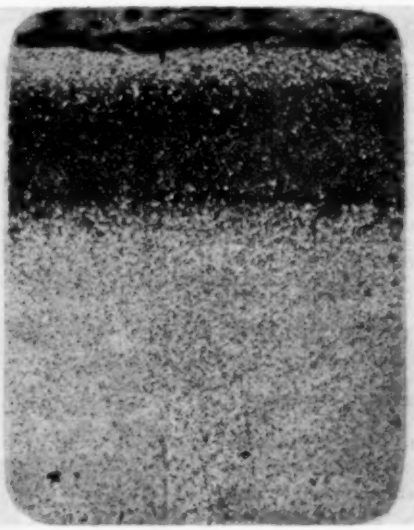


FIG. 4 NITRALLOY "G" NITRIDED FORTY-EIGHT HOURS AT 975 DEG. FAHR. NITAL ETCH, MAG. × 50

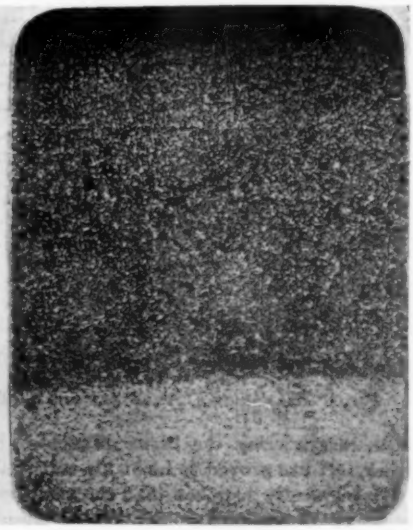


FIG. 5 NITRALLOY "G" NITRIDED FORTY-EIGHT HOURS AT 1200 DEG. FAHR. NITAL ETCH, MAG. × 50

A slight growth takes place during nitriding. Table 2 shows the increase in the length and in the diameter of cylinders of Nitralloy "G" previously oil-quenched from 1650 deg. fahr. and tempered at 1000 deg. fahr. for 4 hr., followed by nitriding at the indicated temperature for 48 hr. Prior to nitriding, these cylinders were ground accurately, after which they measured 5.8160 in. in length and 0.7100 in. in diameter.

This growth can be allowed for in the final machining or grinding before nitriding, or it can be removed afterward by lapping or by some other suitable means. Advantage of this growth can be taken in the case of parts where their usefulness will be destroyed as soon as a slight wear has taken place. The renitriding of such articles, especially at a higher temperature, will produce sufficient growth to make them useful again.

Since a certain amount of growth takes place on nitriding the presence of sharp corners should be avoided. In the case of a cylinder, the growth of the cylindrical surface, together with the growth of the circular section, will combine to push the edge outward. This edge will be very brittle and will chip quite readily. The presence of well-rounded corners and generous fillets should be given careful consideration.

The growth is dependent upon the amount of nitrogen introduced and, therefore, upon the case depth. It is independent of the size of the article. A step-down cylinder, nitrided at

975 deg. fahr. for 48 hr., showed the same growth in the smallest section, which was 1/2 in. in diameter, as in the largest, the diameter of which was 2 in.

The hardness of nitrided articles cannot be determined by the usual Brinell machine, Shore scleroscope, or the present Rockwell tester. Any chipping of the case will give an erroneous reading. The Vickers machine, Herbert Pendulum, Firth Hardometer, and the Shore Monotron are suitable for the purpose. The

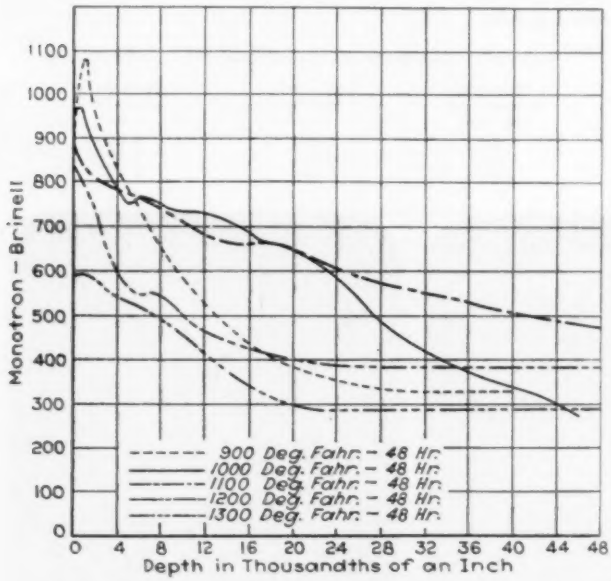


FIG. 6 HARDNESS DETERMINATIONS ON TAPERED SPECIMENS

authors are using the Shore Monotron in their investigative work. It has proved to be highly satisfactory for the purpose.

It frequently happens that threaded portions or other areas must be protected in order not to become surface hardened. Nickel plating gives satisfactory results; tin or solder may be used provided that certain precautions are taken. Only that amount of tin which can be retained by surface tension should be present in order to prevent particles of molten metal from dropping onto articles at a lower level in the nitriding box and thus produce protection at undesirable points. The authors are using a paint of tin oxide mixed with glycerin. The glycerin is an excellent vehicle and decomposes into volatile compounds during the nitriding process so that the tin oxide can be readily reduced to metallic tin by the active hydrogen which is produced as a result of the decomposition of the ammonia.

WEAR RESISTANCE OF NITRIDED SURFACES

The wear resistance of nitrided surfaces should prove to be of great interest to mechanical engineers. The authors are well aware of the fact that wear tests in order to be of maximum value should be conducted in a manner which will similate service conditions.

General conclusions cannot be made after the completion of a series of tests because there is no justification in stating that one metal is more resistant to wear than another except with the qualification that it is true when the tests are conducted under the same conditions. Guillet[3] studied the wearing properties of nitrided Nitralloy, especially with respect to the automotive industry. Some of the data presented by him are included in the following:

[3] Guillet, Lecture to Academy of Sciences in Paris, France.

WEAR OF CYLINDERS

On a car with cast-iron cylinders after 30,000 km. (18,630 miles) the wear was 0.016 in. On a car with nitrided cylinders it was 0.0008 in.

On an aviation motor after 100 hours running the wear of the cylinders of heat-treated steel was 0.0032 to 0.004 in. and the wear of nitrided cylinders was not measurable.

The oil consumption under the same conditions per hp-hr. was:

	New motor	After 100 hours
For a cylinder of heat-treated steel	4–5 grams (60–75 grains)	12–15 grams (180–225 grains)
For a cylinder of nitrided steel	4–5 grams (60–75 grains)	4–5 grams (60–75 grains)

V. T. Malcolm[4] reported on the excellence of nitrided Nitralloy



FIG. 7 APPARATUS USED FOR DETERMINING WEAR RESISTANCE

in its resistance to wear. His paper is of great value because of its practical aspects.

Since there is no universal wear-testing machine nor test methods, the question of wear resistance should be studied under actual service conditions whenever possible.

[4] V. T. Malcolm, Use of Nitrided Steel in High-Temperature High-Pressure Steam Service. Nitriding Symposium, Convention of American Society for Steel Treating, Cleveland, September, 1929.

Lieutenants Kiefer, Crist, and Kindell[1] conducted such tests since they were interested in specific applications of nitrided Nitralloy in which its wear resistance was of paramount importance.

The apparatus used by them, shown in Fig. 7, was made by adapting a drillpress for the purpose. A driver provided with a keyway was substituted for the drill. The upper shaft carries a special fixture for holding the upper specimen. Another special shaft supports the lower specimen and rotates in contact with ball bearings in order that the torque developed by the friction can be determined.

A grooved disk is bolted to the bottom of the shaft for determining the torque. A spring balance is attached to this disk by means of an ordinary door spring which serves to dampen the vibration.

The lower fixture is shown in greater detail in Fig. 8. The base is rigidly fixed to the drillpress table by means of four bolts. The lower shaft (4) is made free to rotate by means of the ball race (5) which is retained in the fixture by means of ring (6). The upper and the lower specimens are shown in contact at (3). The male part (2), which is in the center of the upper driving tool, keeps the upper and the lower shafts in line by rotating in the ball race (1). In this manner (1) and (2) take up all side thrusts and thus prevent wabbling and insure perfect alignment.

A wheel (7) is fitted to the end of the lower shaft and serves to provide the lever arm for measuring the torque. Screw (8) is used to fix a steel wire from the bottom of this shaft to a spring scale for obtaining the other factor (force) which is required for the torque determination.

Hollow cylinders, having an outside diameter of $2^1/_2$ in., an inside diameter of $1^3/_4$ in., and a height of $1^3/_8$ in., were used in the tests.

TABLE 3 NITRIDED NITRALLOY AGAINST NITRIDED NITRALLOY, NITRIDED AT 950 DEG. FAHR. FOR 90 HR.

Run	Specimen	Time hours	Final temp., deg. fahr.	Torque, in-lb.	Friction coef.	Loss in weight, gram	Loss in height, in.	R.p.m.	Total no. of revolutions
1	1	5	401	15	0.237	0.525	...	1150	345,000
	2					0.465			
2	1	5	420	20	0.316	0.606	...	1150	690,000
	2					0.510			
3	1	5	430	22	0.347	0.349	0.001	1150	1,035,000
	2					0.615	0.001		
4	1	5	400	14.4	0.227	0.592	0.001	1150	1,380,000
	2					0.250	0.001		

Upper specimen = No. 1. Lower specimen = No. 2.
Total no. of revolutions = 1,380,000. Pressure = 12 lb. per sq. in.
Total loss in weight of upper specimen = 2.07 g. = 32 grains.
Total loss in weight of lower specimen = 1.84 g. = 28.5 grains.
2 sq. in. of wearing surface on each specimen.
Specimens were finish machined, but not polished, before nitriding.
No scoring occurred.
No lubrication. No loss in hardness took place on surface.

TABLE 4 NITRIDED NITRALLOY AGAINST GRAY CAST IRON

Run	Specimen	Time hours	Final temp., deg. fahr.	Torque, in-lb.	Friction coef.	Loss in weight, gram	Loss in height, in.	R.p.m.	Total no. of revolutions
1	NG	5	420	17	0.268	0.346	0.002	1150	345,000
	CII					0.492	0.002		
2	NG	5	425	20	0.316	0.313	0.000	1150	690,000
	CI					0.818	0.0025		
3	NG	5	400	19	0.300	0.121	0.000	1150	1,035,000
	CI					0.605	0.0025		
4	NG	5	425	18	0.284	0.170	0.000	1150	1,380,000
	CI					0.119	0.004		

Brinell hardness of gray cast iron = 140. NG = Nitralloy G. CI = Gray cast iron.
Total loss in weight of nitrided Nitralloy = 0.95 g. = 14.6 grains.
Total loss in weight of gray cast iron = 2.03 g. = 31.3 grains.
Specimens were finish machined but not polished before nitriding. No scoring occurred.

[1] Lieutenants Kiefer, Crist, Kindell, Thesis, "The Wearing Properties of Nitrided Nitralloy Against Various Alloys," submitted in partial fulfilment of the requirement for the degree of Master of Science from the Massachusetts Institute of Technology, 1929.]

The coefficient of friction was determined by means of the following formula:

$$T = \frac{2}{3} FW \frac{R^3 - r^3}{R^2 - r^2}$$

where

W = weight in lb. acting vertically on the wearing surface
T = torque in inch pounds
F = coefficient of friction
R = outside radius of specimen in inches
r = inner radius of specimen in inches
P = pull in pounds due to torque.

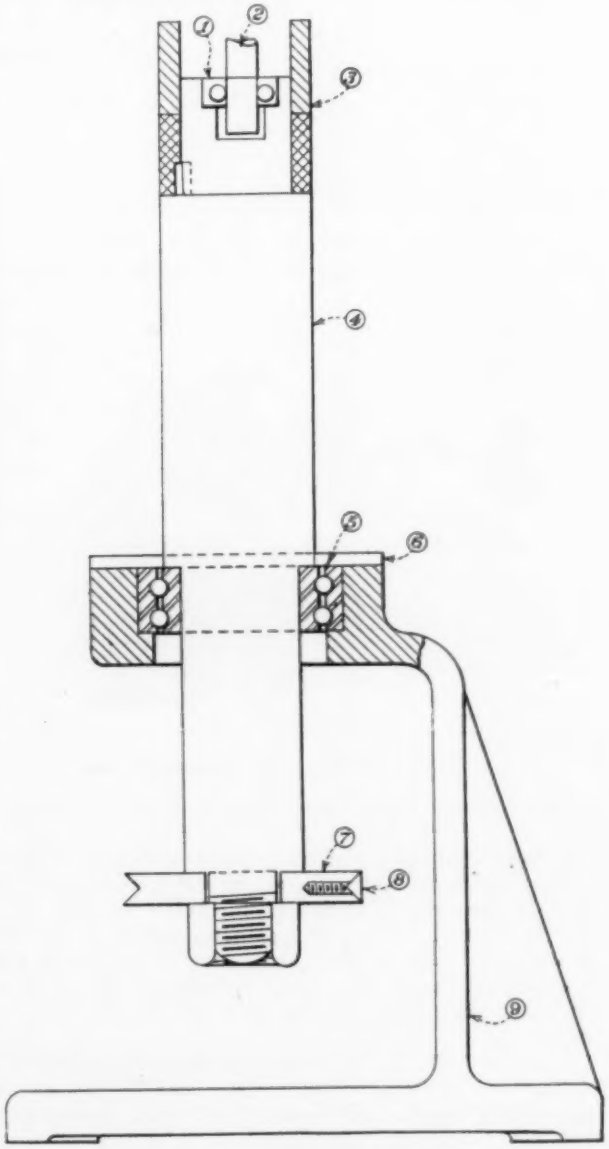


FIG. 8 FRICTION-TESTING FIXTURE

The tests were conducted without lubrication and without the application of external heat. Observations were made at 5-hr. intervals. These observations included the temperature developed by friction, torque, loss in weight and in height, and the condition of the wearing surfaces. The temperature was determined by means of a thermocouple which extended through the lower specimen holder to a point $^1/_{16}$ in. from the rubbing surface. Sections were examined microscopically after the com-

pletion of each test. Differences in the depths of the nitride case at the worn and at the unworn surfaces were noted.

TABLE 5 NITRIDED NITRALLOY AGAINST SPECIAL BRONZE

Run	Specimen	Time, hours	Final temp., deg. fahr.	Torque, in-lb.	Friction coef.	Loss in weight, gram	Loss in height, in.	R.p.m.	Total no. of revolutions
1	NG	5	200	16.6	0.262	0.279	0.0005	300	
	B-1					0.030	0.002		90,000
2	NG	5	205	14.0	0.221	0.306	0.0008	300	
	B-1					0.020	0.000		180,000
3	NG	5	320	12.0	0.190	0.078	0.0005	1150	
	B-1					0.130	0.001		525,000
4	NG	5	275	6.0	0.095	0.000	0.000	1150	
	B-1					0.132	0.001		870,000
5	NG	5	240	6.0	0.095	0.000		1150	
	B-1					0.116			1,215,000
6	NG	5	250	6.0	0.095	0.065		1150	
	B-1					0.110			1,560,000

Brinell hardness of B-1 was 86.
NG = Nitralloy G; B-1 = Special bronze (cast).
No scoring took place with this bronze and there was no pick-up by the Nitralloy. No lubrication was used.
Analysis of special bronze:

Copper	80.2	Tin	11.5
Nickel	5.5	Zinc	2.8
Lead	Trace		

Total loss in weight of nitrided Nitralloy = 0.73 g. = 11.2 grains.
Total loss in weight of special bronze = 0.54 g. = 8.3 grains.

Tables 3, 4, and 5 give the results of tests in which the nitrided Nitralloy was tested against itself, against gray cast iron, and

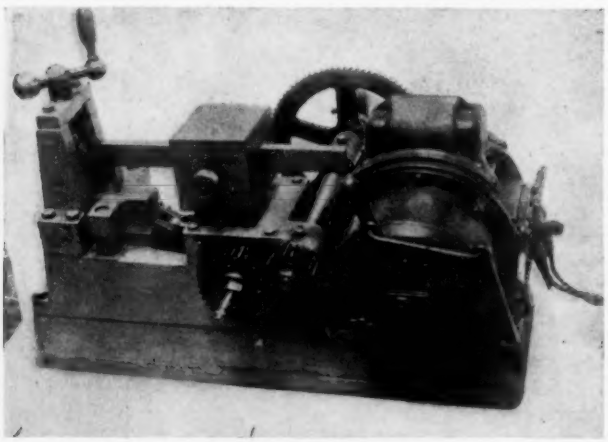


FIG. 9 WEAR-TEST MACHINE

against a special bronze the analysis of which is included in Table 5.

In the test in which nitrided Nitralloy was operated against the special bronze, fine particles from the former became embedded in the surface of the latter. In all cases the surfaces were entirely free from scoring; no appreciable decrease in the depth of the nitride case could be noted when the specimens were examined microscopically. Also, the surface hardness of the nitrided specimens was the same after the test as before it was started.

Lieutenants Blick and Buracker[6] not only confirmed the results obtained by Lieutenants Kiefer, Crist, and Kindell, but also tested the nitrided Nitralloy against itself at a temperature of 1000 deg. fahr. After 13.5 hr. with a total number of revolutions of 858,600, the loss in weight of each specimen was only 0.46 gram (7 grains) with a coefficient of friction of 0.28.

Captain H. V. Hopkins[7] conducted wear tests by means of the apparatus shown in Fig. 9. This apparatus is provided with a variable-speed motor and with a reciprocating mechanism whereby two specimens can be rubbed against each other under a desired load. Specimens 2 in. by 1 in. by $\frac{1}{2}$ in. were used in the tests. The stroke of the machine is 2 in., so that the specimens were rubbed together with a variation of contact area of 1 to 2 sq. in. The pressure, therefore, was twice as great at the end of the stroke as it was at mid-stroke.

The load was applied through a lever mechanism with a sliding weight. A handle operating a screw permitted vertical adjustment of the lever to compensate for the wear and for the varying thicknesses of specimens in order to maintain the lever in a horizontal position and to maintain a constant load. The load was varied from 13 to 150 lb.

All specimens were finished on a fine emery belt to approximately the average commercial surface for bearing purposes All grease and oil or grit were removed from the wearing surfaces before each test. The machine was stopped at frequent intervals to observe the condition of the surfaces. Approximately 60 combinations of materials were tested without lubrication in this machine at varying speeds and loads. Combinations of materials such as were used by Kiefer, Crist, and Kindell, and later by Blick and Buracker were run in this machine with results which checked closely with those obtained by them.

The results obtained in the wear tests in which the wear in one series was a consequence of rotary motion and in the other series of translatory motion indicate that the use of nitrided materials in engine or machine parts which are subject to severe wear conditions and difficult to lubricate offers promising possibilities

CONCLUSIONS

The physical properties of Nitralloy in the un-nitrided condition compare very favorably with those exhibited by the best of the structural alloy steels.

All parts should be heat-treated before nitriding in order to obtain desirable physical properties of the core and to insure proper grain refinement since the original grain size of the material is not reduced during the nitriding operation.

Nitriding at temperatures ranging from 900 to 1300 deg. fahr. inclusive shows that the depth of case increases, but the hardness decreases with the increase in temperature. It is possible to nitride at a relatively high temperature, 1200 deg. fahr. for example, to obtain considerable depth of case and then to nitride at a lower temperature, such as 950 deg. fahr., to obtain the desired surface hardness.

The presence of decarburization before nitriding produces a brittle case which is likely to spall or chip. This decarburized layer must be removed before the subjection of the parts to the action of the ammonia gas. Strains created by machining and by heat-treating of parts must be eliminated if warping or distortion in the finished articles is to be prevented.

Nickel plating and a coating of tin or solder will protect parts against nitriding. A coating of tin oxide in glycerin will also serve as a protective agent. A slight growth takes place during nitriding. Allowance can be made for this growth in the final machining or grinding operation, before nitriding, or by lapping, or by other suitable means after nitriding.

Nitrided articles offer marked resistance to atmospheric, water, and salt-water corrosion. Nitrided Nitralloy exhibits excellent wearing properties even when tested without lubrication against certain other alloys, notably against nitrided Nitralloy itself, and a special bronze containing nickel. In view of the many desirable properties possessed by nitrided articles, this subject should prove of increased interest to mechanical as well as to metallurgical engineers.

[6] Lieutenants Blick and Buracker, Thesis, "The Wearing Properties of Nitrided Materials," submitted in partial fulfilment of the requirement for the degree of Master of Science from the Massachusetts Institute of Technology, 1930.

[7] Captain H. V. Hopkins, Thesis, "The Wear Resistance of Certain Ferrous and Non-Ferrous Materials," submitted in partial fulfilment of the requirement for the degree of Master of Science from the Massachusetts Institute of Technology, 1930.

MSP-53-2

Nitriding Analyses—Their Physical Properties and Adaptability

By R. S. SERGESON[1] and M. M. CLARK,[1] MASSILLON, OHIO

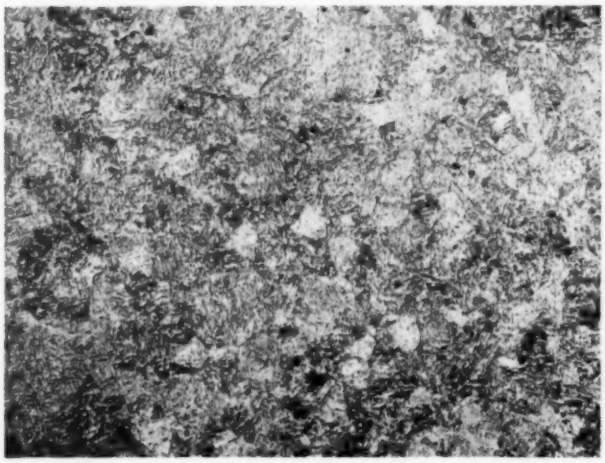


Fig. 1 Nitralloy Type G Water-Quenched From 1700 Degrees Fahrenheit—HNO_3 × 100—Normal Structure and Grain Refinement

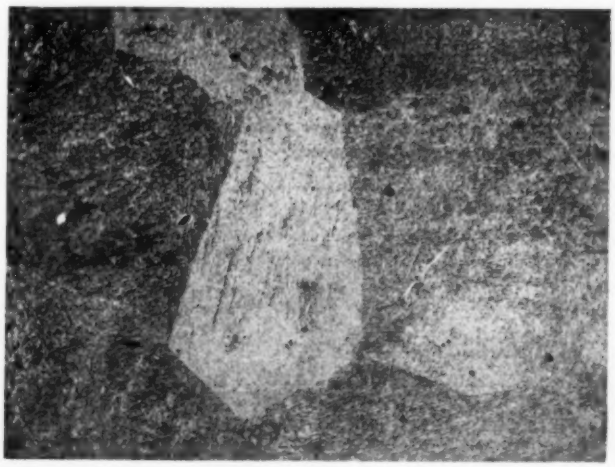


Fig. 4 Nitralloy Type G Water-Quenched From 2200 Degrees Fahrenheit—HNO_3 × 100—Highly Coarse Grained

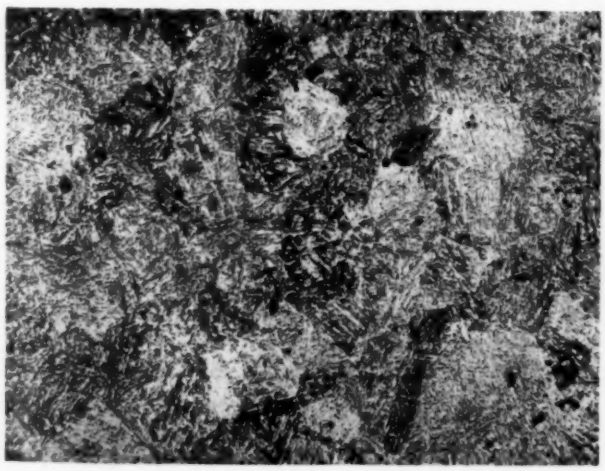


Fig. 2 Nitralloy Type G Water-Quenched From 1800 Degrees Fahrenheit—HNO_3 × 100—Slight Grain Growth



Fig. 3 Nitralloy Type G Water-Quenched From 2000 Degrees Fahrenheit—HNO_3 × 100—Coarse Grained

In this paper the authors give the range of analyses of the present chrome-aluminum-molybdenum nitriding steels. The physical properties of these analyses are given in detail both in regard to tensile and impact values. Some data on fatigue tests have been added. Mention is made as to the proper quenching temperatures and there is a suggestion as to the amount of stock to be removed for elimination of decarburized surface obtained previous to machining. Many applications which are operating highly successfully are given.

A LITTLE over two years ago all nitriding steels were in their early stages of development. They were made in the electric furnace in small lots and were high in price. If one wished to nitride he was forced to build his own equipment, and commercial nitriding was unknown. Today, however, nitriding analyses can be made in large tonnage in the open hearth, thus reducing the cost. Most of the furnace builders are marketing nitriding equipment, and many plants are taking

[1] Metallurgical Department of the Republic Steel Corporation.

Presented at the Fourth National Machine Shop Practice Meeting, Chicago, Ill., Sept. 22 to 24, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

on commercial nitriding. The demand for the nitriding analyses is steadily gaining and it is quite important that a thorough understanding of their properties be obtained.

The following types are now available on the commercial market:

Type	C	Mn	Cr	Mo	Al
I	0.10/0.20	0.40/0.70	0.80/1.30	0.15/0.25	0.60/1.20
H	0.20/0.30	0.40/0.70	0.80/1.30	0.15/0.25	0.60/1.20
G	0.30/0.40	0.40/0.70	0.80/1.30	0.15/0.25	0.60/1.20
High C	0.55/0.65	0.40/0.70	0.80/1.30	0.15/0.25	0.60/1.20

These analyses are being forged with no difficulty at temperatures of 2175–2250 deg. fahr.

Before determining the physical properties of these analyses a preliminary investigation was conducted to determine the proper range of quenching temperature. This was accomplished

TABLE 1 QUENCHING CYCLES

Analyses	C	Mn	S	P	Cr	Mo	Al
Type G	0.341	0.52	0.02	0.017	1.14	0.15	0.95
Type H	0.266	0.50	0.02	0.019	1.04	0.15	0.95

Heating Phase				Cooling Phase			
Temp., deg. fahr.	Medium	Brinell Hardness G	Brinell Hardness H	Temp., deg. fahr.	Medium	Brinell Hardness G	Brinell Hardness H
1280	Water	228	207	1700	Water	555	512
1300	Water	228	202	1600	Water	555	512
1340	Water	228	202	1500	Water	555	430
1380	Water	217	187	1450	Water	444	340
1420	Water	202	196	1400	Water	387	196
1460	Water	212	202	1380	Water	248	196
1500	Water	321	302	1360	Water	183	196
1540	Water	402	387	1340	Water	183	196
1580	Water	444	418	1320	Water	183	183
1620	Water	512	512	1300	Water	174	183
1640	Water	532	512	1280	Water	174	183
1680	Water	555	512	1260	Water	174	183
1700	Water	555	512				
1800	Water	555	512				

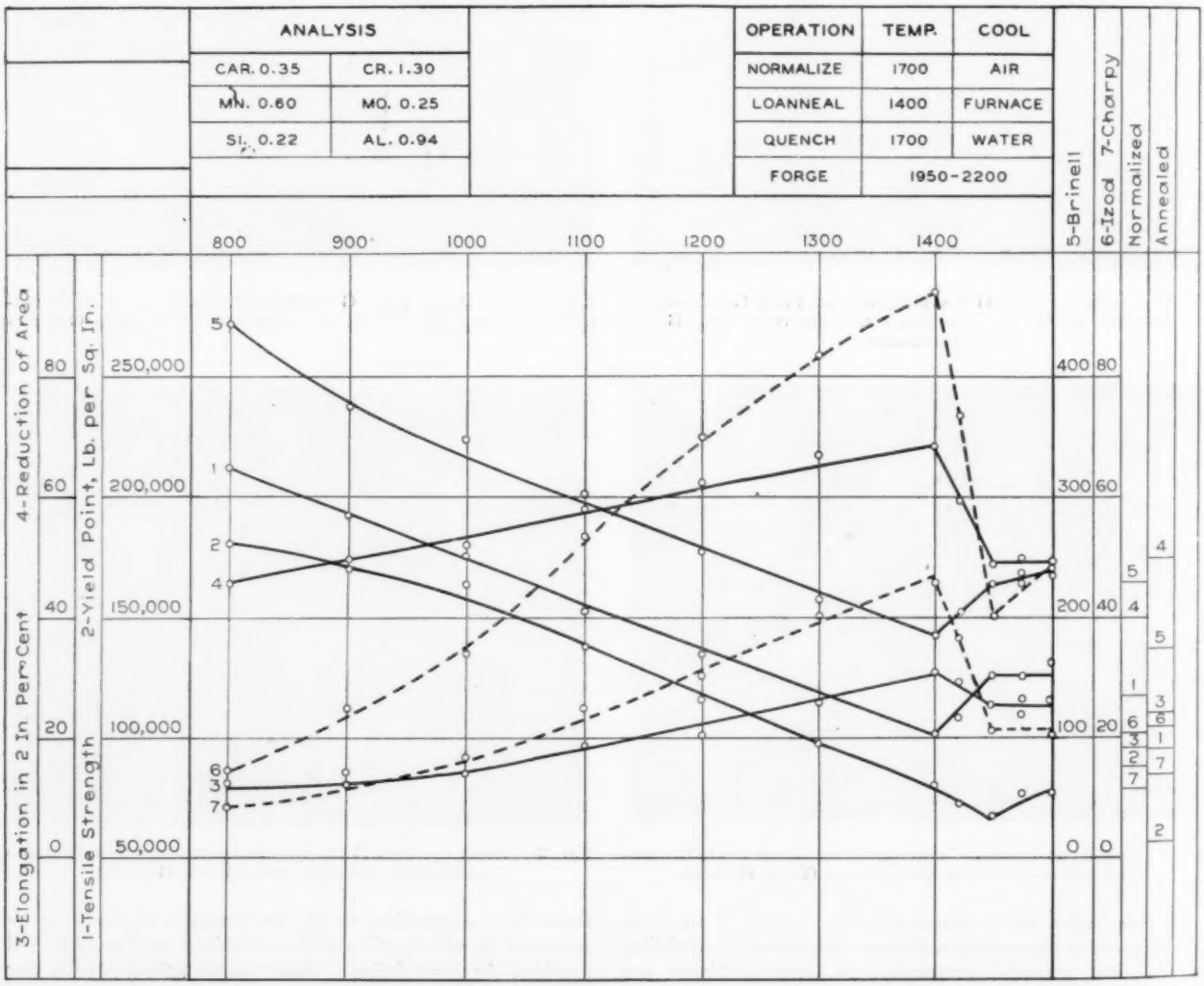


FIG. 5 ELONGATION, REDUCTION AREA, TENSILE STRENGTH, AND YIELD POINT FOR NITRALLOY TYPE G

in each case by quenching a series of specimens from successive increasing and decreasing temperature intervals and studying their hardness and structure. This is termed a quenching cycle. Typical cycles are shown in Table 1 for type G (0.34 carbon) and type H (0.26 carbon). From results shown in this table and a study of the microstructures of quenched specimens, a quenching temperature range of 1700–1750 deg. fahr. has been recommended for obtaining the most uniform structure and hardness. Specimens were also quenched from temperatures of 1700, 1800, 2000, and 2200 deg. fahr. to note the effect of temperature on coarsening of grain. At 1800 deg. fahr. grain growth is evident and increases with temperature. Photomicrographs of this effect of temperature are shown in Figs. 1, 2, 3, and 4.

PHYSICAL DATA—TENSILE AND IMPACT VALUES, TYPES G AND H

With the quenching temperature settled, physical-property

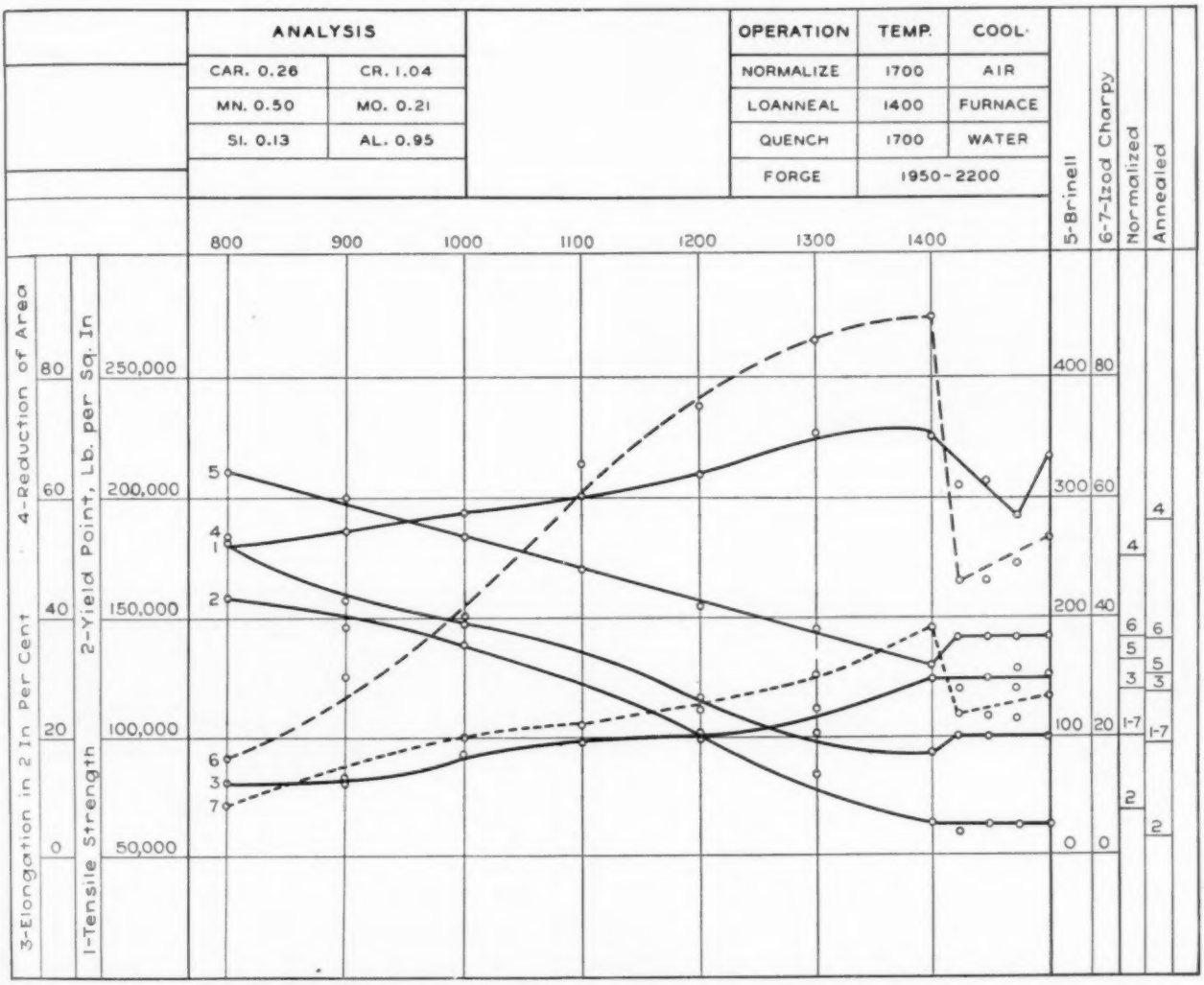


FIG. 6 ELONGATION, REDUCTION AREA, TENSILE STRENGTH, AND YIELD POINT FOR NITRALLOY TYPE H

TABLE 2 PHYSICAL PROPERTIES

Temp., deg. fahr.	Quench	Draw	Time, hr.	Size in. round	Yield	Tensile	Elong.	R.A.	Brinell
				Type G					
1725	Water	1000	4	4	134800	155900	13.0	40.8	340
1725	Water	1100	4	4	113300	132800	16.5	47.8	302
1725	Water	1200	4	4	100500	124900	18.0	51.9	269
1725	Water	1000	3	3	140500	158800	13.0	40.4	340
1725	Water	1100	3	3	120500	140000	15.5	47.8	293
1725	Water	1200	3	3	104900	126900	17.5	56.0	255
1725	Water	1000	2	2	149800	171300	12.5	40.4	340
1725	Water	1100	2	2	137900	158300	15.5	50.6	302
1725	Water	1200	2	2	135000	154900	15.5	49.2	269
1725	Water	1000	1	1	164800	182300	12.5	44.4	351
1725	Water	1100	1	1	138900	156000	15.0	51.9	302
1725	Water	1200	1	1	120900	139300	20.0	58.5	269
				Type H					
1725	Water	1000	4	4	102300	117900	19.5	54.7	277
1725	Water	1100	4	4	93900	110000	19.5	58.5	228
1725	Water	1200	4	4	82800	103900	20.0	61.0	217
1725	Water	1000	3	3	103900	119800	18.0	54.7	269
1725	Water	1100	3	3	96800	114800	18.0	57.3	255
1725	Water	1200	3	3	83500	104300	21.0	61.0	223
1725	Water	1000	2	2	124900	141900	14.5	49.2	302
1725	Water	1100	2	2	104900	121300	18.0	59.8	262
1725	Water	1200	2	2	94900	112000	19.0	63.6	228
1725	Water	1000	1	1	130900	146300	14.0	53.3	332
1725	Water	1100	1	1	112300	126800	15.5	58.5	293
1725	Water	1200	1	1	104900	122300	18.0	64.8	255

TABLE 3 PHYSICAL PROPERTIES

Tensile and Izod lengths were water- and oil-quenched from 1550 deg. fahr. and drawn from 1000 deg. fahr. to 1375 deg. fahr.

Analysis

C	Mn	S	P	Si	Cr	Mo	Al
0.55	0.55	0.019	0.014	0.15	1.10	0.20	1.14

Test	Draws	Yields	Tensile	Elong.	Red. area	Izod	Tensile Izod Brinell
			Oil-Quenched—1550 deg. fahr.				
1	1000	182700	206200	11.0	35.7	20–17	425–477
2	1100	160700	178200	13.5	43.4	28–29	375–402
3	1200	137300	158700	16.0	49.2	39–42	321–340
4	1300	106800	129800	21.0	56.0	60–58	269–286
5	1375	93300	116300	23.0	56.0	62–65	241–255
			Water-Quenched—1550 deg. fahr.				
6	1000	183200	208200	11.0	37.3	15–12	425–477
7	1100	161700	179700	13.5	44.9	28–28	375–444
8	1200	139800	158300	16.0	49.2	40–44	325–364
9	1300	107300	129800	20.5	54.7	54–58	262–302
10	1375	94800	115300	23.0	57.3	65–65	228–269

charts showing tensile, Izod and Charpy impact values were made using 1-in. round stock. These data in chart form are shown in Figs. 5 and 6, for types G and H respectively. However, as these analyses are being used in larger sections, data have been obtained to show the effect of size in heat-treatment. Specimens of types G and H were heat-treated in 4-, 3-, 2-, and 1-in. rounds and their tensile properties determined. Table 2 shows the values obtained, and Fig. 7 shows values in chart

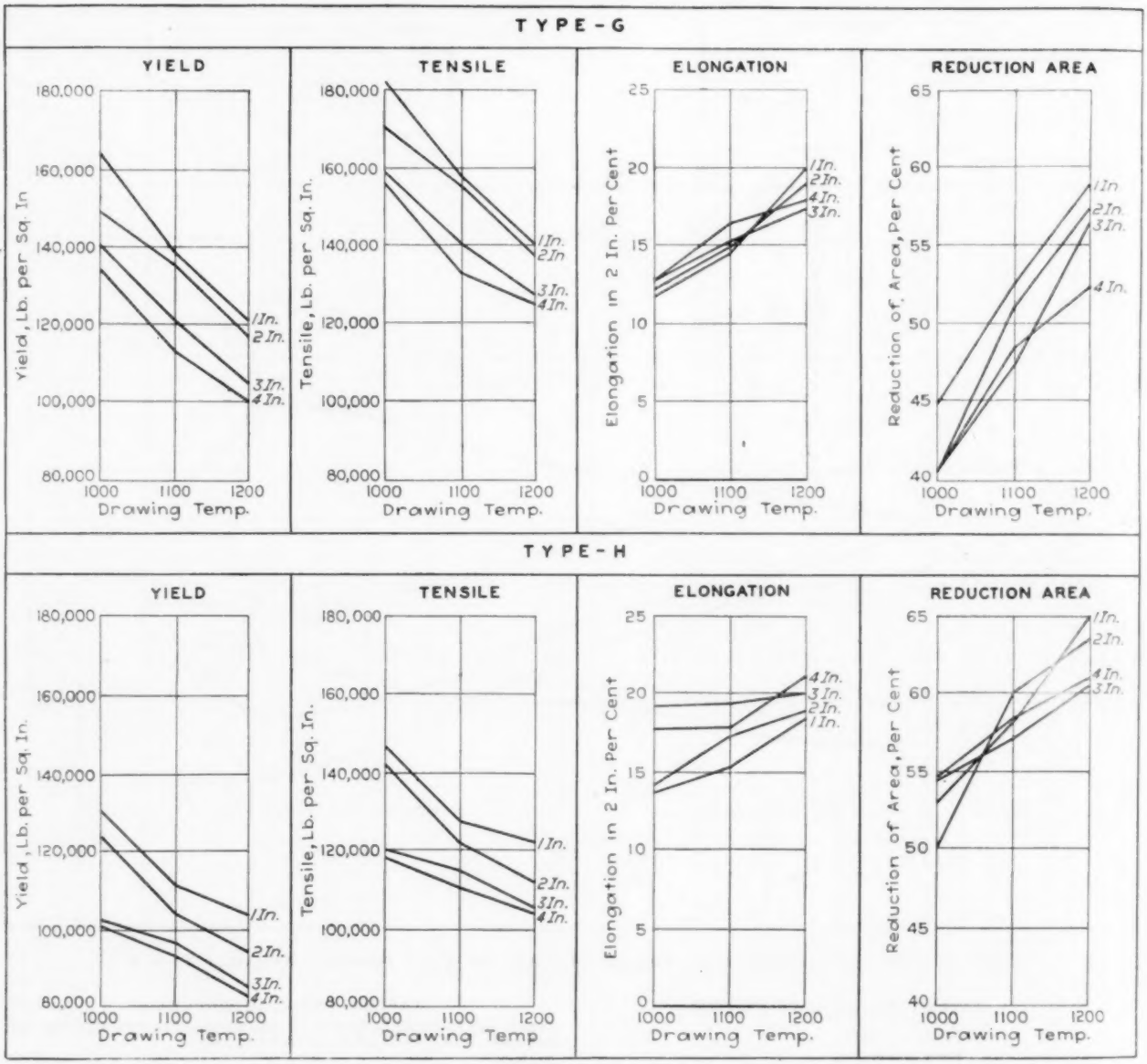


FIG. 7 YIELD POINT, TENSILE STRENGTH, ELONGATION, AND REDUCTION AREA FOR TYPES G AND H NITRALLOYS

form. These values were from tests taken longitudinally as near halfway center to edge as possible. They show clearly a marked decrease in yield point and tensile strength with increase in diameter, while the elongation and reduction of area are less affected.

TABLE 4 VALUES OBTAINED BY HEAT-TREATING

Tensile lengths were water-quenched from 1750 deg. fahr. and drawn from 1000 deg. fahr. to 1400 deg. fahr.

Analysis

C	Mn	S	P	Si	Cr	Al	Mo
0.14	0.49	0.021	0.017	0.24	1.01	....	0.21

Draw, deg. fahr.	Yield	Tensile	Elong.	Red. area	Izod	Brinell
1000	95500	111000	20.0	67.5	86	241
1100	91000	106000	20.5	70.0	105	228
1200	75000	95000	23.5	73.0	110	202
1300	60900	82000	28.0	75.5	114	172
1400	47300	89000	29.0	62.4	48	179
Normalized 1750 deg. fahr. and drawn as shown						
1000	46000	78000	29.0	65	90	153
1200	49000	74000	30.5	69	93	149
1400	42000	79000	30.0	65	75	140

FATIGUE PROPERTIES OF NITRALLOY G

While the authors do not have complete fatigue data, the following test it is believed will be of interest.

Nitralloy G stock 0.34 C, size $1^1/_2$ in. round, heat-treated, was used for tests. Tensile and fatigue (cantilever type) specimens were machined. The specimens were tested as heat-treated and as nitrided to give $^1/_{32}$-in. case. The results were as follows:

TENSILE TESTS

Treatment	Yield point, lb. per sq. in.	Ultimate strength	Per cent elong. in 2 in.	Per cent red. of area
As heat-treated.....	74,650	103,300	25.5	62.9
As heat-treated.....	74,550	102,000	25.0	63.4
As heat-treated and nitrided	83,200	105,700	10.0	12.2
As heat-treated and nitrided	83,500	108,700	10.5	12.5

FATIGUE TESTS

Treatment	Stress, lb. per sq. in.	Number of reversals	Remarks
Heat-treated	36,000	33,000,000	Did not break
Heat-treated	75,000	40,800	Bar broke
Heat-treated and nitrided	75,000	28,000,000	Did not break
Heat-treated and nitrided	85,000	1,411,000	Bar broke

It is quite evident that, at least in small sections, the nitrided case actually increases the fatigue limit of the steel.

Tensile and Impact Values of High-Carbon Nitralloy

A preliminary quenching cycle run on this type determined 1550 deg. fahr. as the proper quenching temperature with oil as a medium for quenching light sections and water for heavy sections (2-in. round or larger). One-inch round sections were heat-treated, being quenched both in water and in oil.

Table 3 gives data showing that little or no difference results due to the quenching medium when treated in the 1-in. round section.

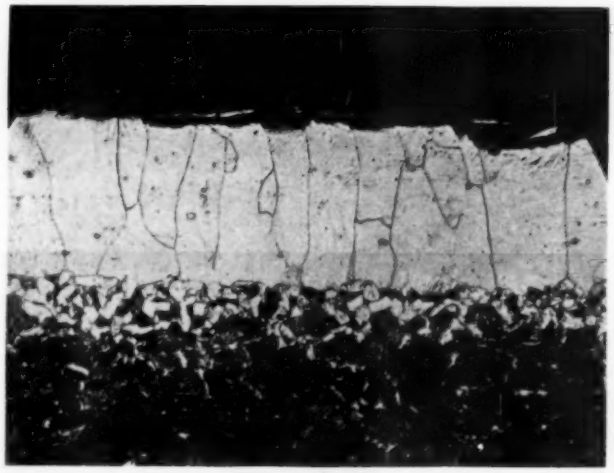


Fig. 8 Decarburized Surface—HNO_3 × 100

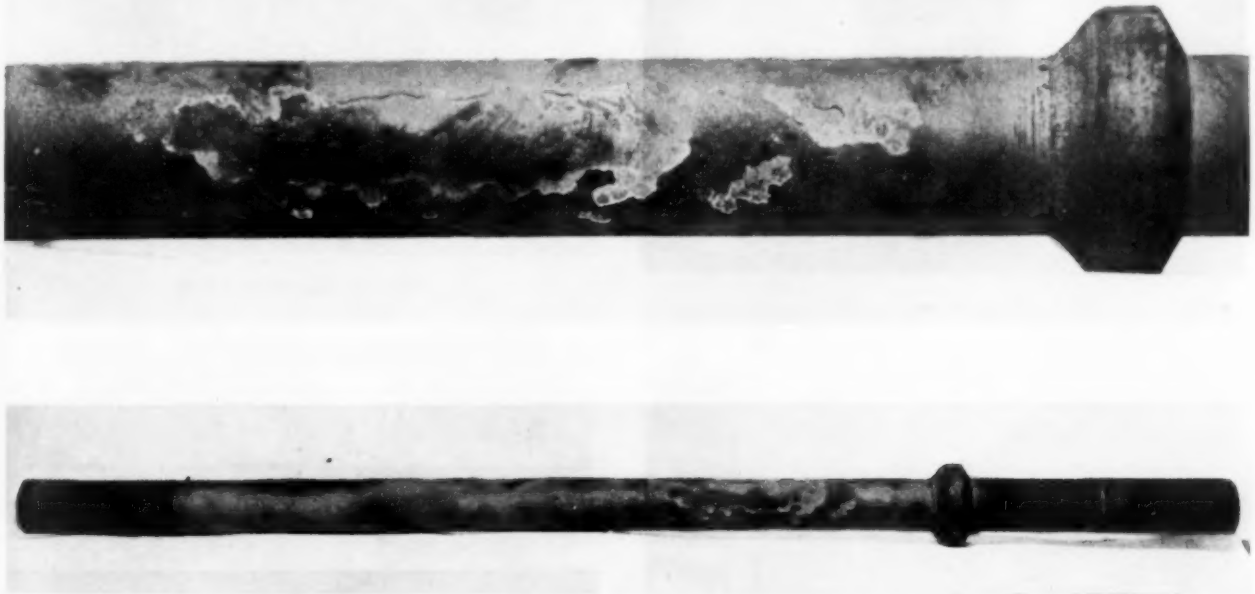


Fig. 9 Pumpshafts With Decarburized Surface—Nitriding Has Caused Spalling

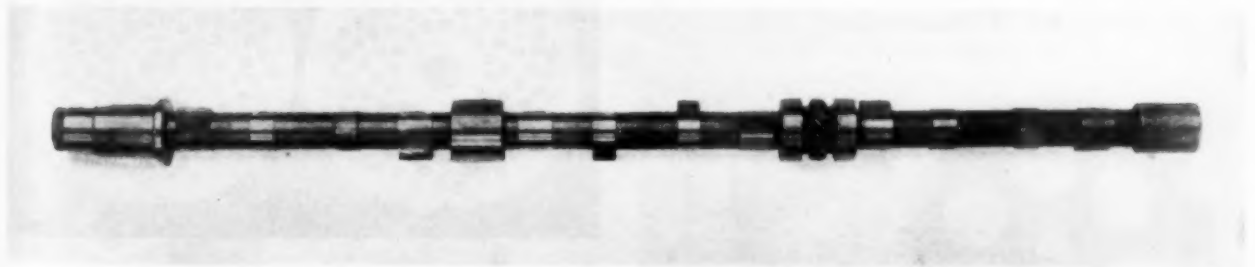


Fig. 10 Bus Motor Camshaft of Type G Nitralloy

Tensile Values and Impact Values of Type I

A preliminary quenching cycle on this type determined 1700–1750 deg. fahr. as the proper quenching temperature. Sections 1-in. round were heat-treated and water-quenched. Table 4 gives the values obtained by heat-treating. This product also lends itself to the making of flat products, e.g., sheet and strip, in which case water-quenching is difficult. Hence such products are normalized and drawn. Physical data on normalized and drawn stock are also included in Table 4. The normalized and drawn stock is applied to stamping and forming operations.

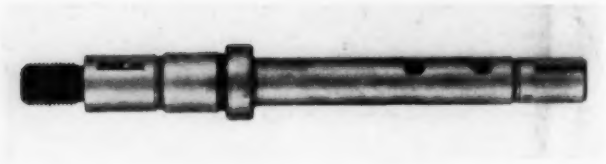


FIG. 11 PUMPSHAFT OF TYPE H NITRALLOY

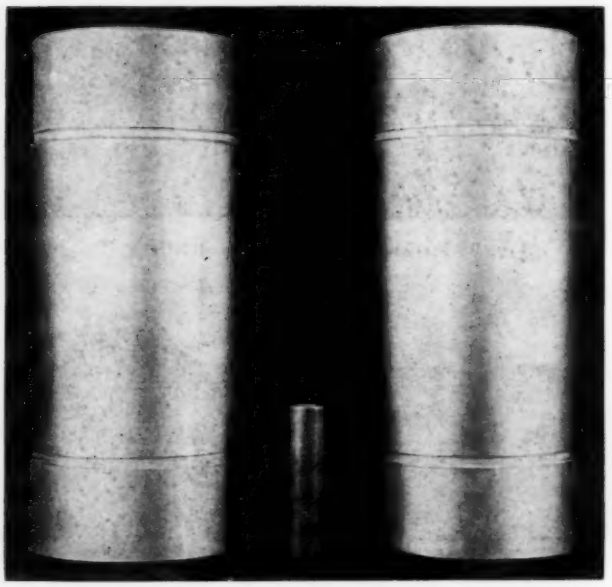


FIG. 12 GAS ENGINE AND AUTOMOBILE ENGINE WRIST PINS

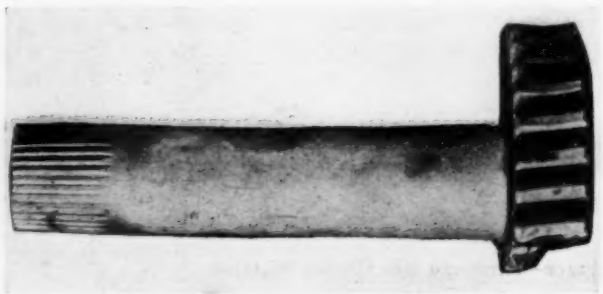


FIG. 13 STEERING SECTOR

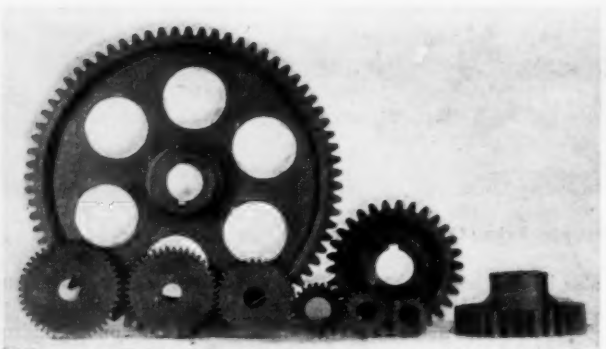


FIG. 14 TIMING GEARS

ADAPTABILITY

The experience of the authors as to the adaptability or selection of these nitriding analyses to the machine, automotive, and aviation fields dates back but little over two years. In that time there have been many successful applications, many more are still under test, and some few were failures. To give a clearer understanding, the various applications are listed and

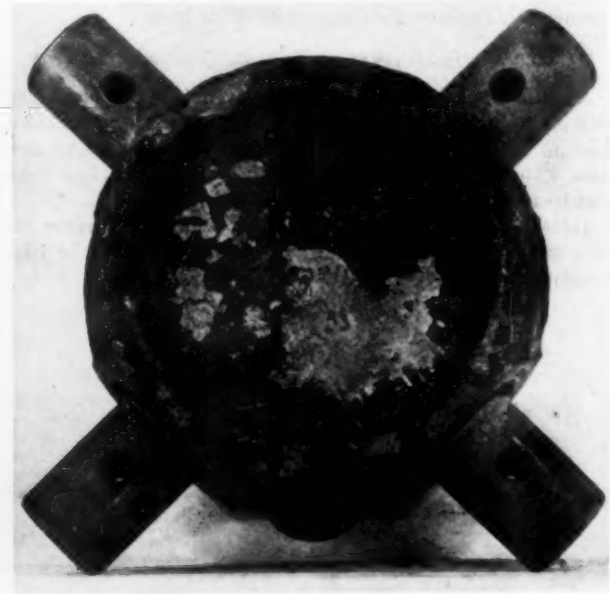


FIG. 15 UNIVERSAL CROSS



FIG. 16 BAR FOR CRANKSHAFT TURNING MACHINE

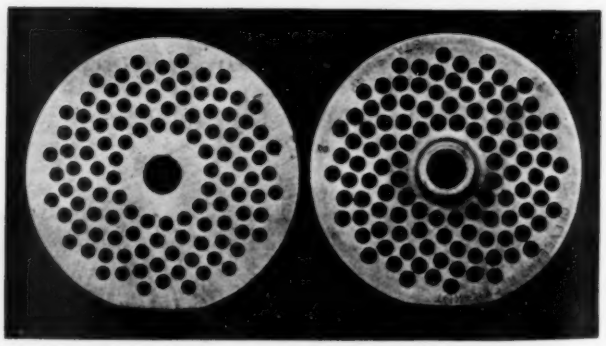


FIG. 17 MEAT-CUTTER DISKS

illustrations of many are shown. For best results, these analyses should be used in the heat-treated state, being fully machined before being subjected to the nitriding treatment. Although it has been previously stressed, it is again to be pointed out that, after heat-treating, the decarburized surface must be completely removed by machining. In general, a removal of $^{1}/_{16}$

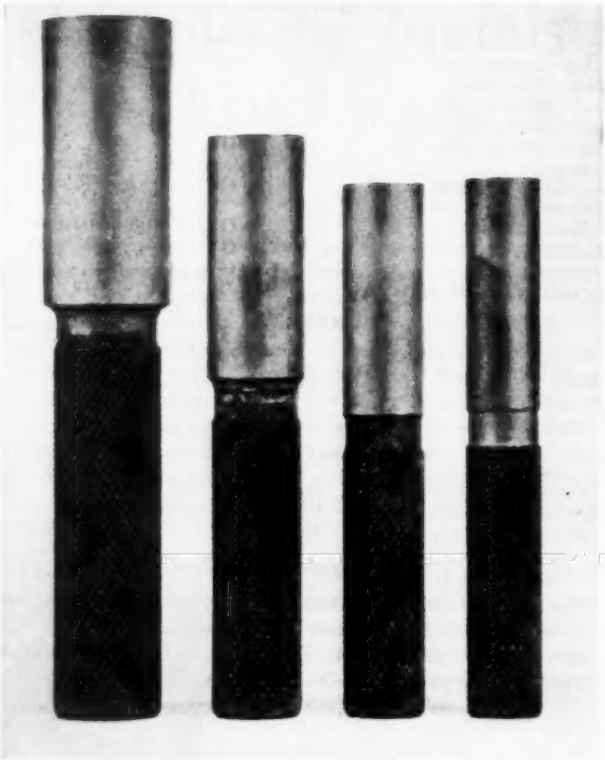


FIG. 18 PLUG GAGES



FIG. 20 DIE INSERT FORGING AND CROSSHEAD GUIDES

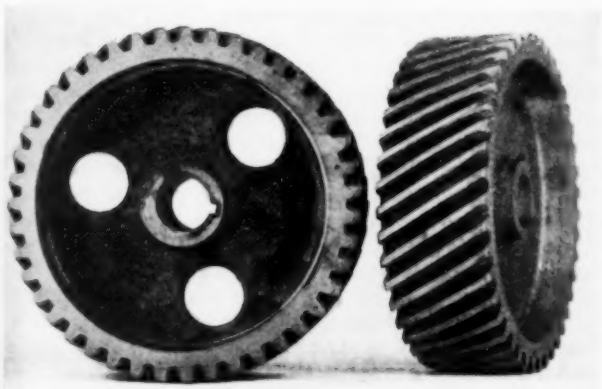


FIG. 22 AIRPLANE GEARS



FIG. 21 BRAKE DRUM, TYPE 1

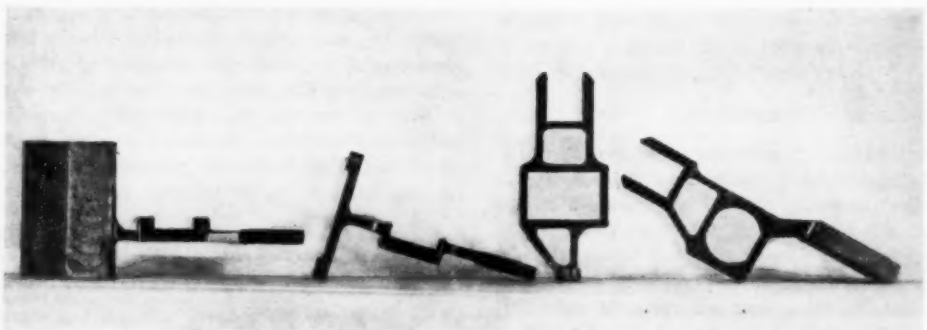


FIG. 19 MOVIETONE CAMERA PARTS

in. to $^3/_{32}$ in. is sufficient to remove all decarburized area. Fig. 8 shows a typical decarburized surface, and Fig. 9 illustrates the effect of nitriding a decarburized surface. It will be noted how the surface of the pumpshaft has spalled.

Applications in the following fields are listed. Unless mentioned otherwise, application is satisfactory.

AUTOMOTIVE FIELD

	Type	Fig. No.
Crankshafts	G	..
Camshafts	G	10
Pumpshafts	H	11
Piston pins	G or H	12
Steering sectors	G	13
Timing gears	H	14
Universal cross	G	15
Brake drums (under test)	I	21

All of the above applications are giving excellent service.

AVIATION FIELD

	Type	Fig. No.
Wristpins	G	..
Gears	G	22
Crankshafts	G	Still under test

LOCOMOTIVE

	Type	Fig. No.
Valve-motion pins and bushings	G or H	..
Crosshead guides	G	20
Piston rods	G	Still under test

MACHINE PARTS

	Type	Fig. No.
Large machine parts	G	16
Meat-cutter disks	G	17
Paper rolls	G	..
Paper corrugating rolls	G	..
Radial-drill spindles	G	..
Plug gages	H or G	18
Gas-engine wristpins	G	12
Hydraulic-pump cams	G	..
Roller-leveler rolls	G	Still under test
Shafting	G	..
Bushings	G	..
Movietone camera parts	H	19

VALVE

	Type
Steam	G or H
Oil and tar	H
Refrigerator	H

MISCELLANEOUS

Die-insert drop forging	High carbon type	Fig. 20
Upsetting-die inserts	High carbon type	
Clay-forming dies	Type G or H	

ACKNOWLEDGMENTS

The authors wish to acknowledge and thank P. R. Letz, of the Copper Bessemer Corporation, Mt. Vernon, Ohio, for the excellent fatigue results and R. B. Russell, of the Republic Steel Corporation, for the very good microphotographs.

MSP-53-3

Replacing Metals by Synthetic Resins—Phenol-Resinoid Molding Technique

By LEON V. QUIGLEY,[1] NEW YORK, N. Y.

It is the purpose of this review to assist the engineer and mechanic whose experience has been chiefly with metals to get his bearings with regard to the synthetic resin plastics, which are becoming increasingly important in the metal trades. The paper deals particularly with that type of plastic or synthetic resin materials known as phenol resinoids. The review has involved a study of the literature of America, England, and the Continent covering the past five years, and relies also upon the technical archives of the American company which, during the past 20 years, since the inception of these materials, has been engaged in their manufacture.



WHEN a new class of material begins to rank with metal, stone, and wood as one of the four principal materials of construction, it becomes properly a subject for thoughtful consideration by engineers. Properties of the new material are to be evaluated, and not only in the absolute sense, but in relation to the properties of materials which they may replace. It is the purpose of this review to assist the engineer and mechanic whose experience has been chiefly with metals to get his bearings with regard to the synthetic resin plastics, which are becoming increasingly important in the metal trades. The paper deals particularly with that type of plastic or synthetic resin materials known as phenol resinoids. The review has involved a study of the literature of America, England, and the Continent covering the past five years, and relies also upon the technical archives of the American company which, during the past 20 years, since the inception of these materials, has been engaged in their manufacture.

The current issue of *The Industrial Bulletin*, published by Arthur D. Little, points out that the aggregate value of the products of the plastics industry is already more than one-tenth that of the products from both the lumber and wood-working industries of the United States. Yet the wood industry has been long established, while the plastics industry might be considered as still in the industrial nursery.

Considering the relation of phenol-resinoid materials to the metal industry, it should be remarked that they have, for specific purposes, replaced practically every known form of metal. For the heaviest duty in buildings and machines, it would appear that resinoids will not supplant metal. It is conservative to assume that for many less strenuous services they will replace metal entirely. The new material, technically described as phenol resinoid and popularly associated with the trade-mark name of the company founded by its inventor, should not be hastily rated as inferior or superior to metal, wood, or stone. In contrast to each one of these older substances it possesses inherent advantages and limitations. It is something distinct, and therefore can stand alone, not as a questionable substitute, but as an auxiliary new material having a technical personality of its own. Phenol-resinoid materials have attained their present widespread adoption because they are possessed of a remarkable assortment of properties and because they possess remarkable ease of adaptability. It is these two factors which chiefly govern the adoption and industrial position of any material.

The importance of manufacturing processes, based on plasticity of the raw materials, is that they make possible conversion directly to the desired final shape, with practically no loss of material, and at an expenditure of time and labor which is usually only a fraction of that which would be required if the part were shaped from rough stock by machining or cutting operations.

The most important group of industrial plastics, which, however, appears to be still in its infancy so far as varieties are concerned, is the class in which synthetic resins are used as a binder. The earliest of these and by all odds still the most important are the phenolic-resinoid materials. They possess extreme plasticity in the formative period. The finished product is resistant to practically all of the normal conditions tending to structural breakdown. It is characterized by a high degree of electrical and thermal insulation, and presents attractive surface, requiring practically no polishing as it comes from the mold. Structurally, the material possesses a degree of toughness and strength unusual in organic substance, and where this is not adequate, as in the case of fine machine threads, it is possible to mold metal threaded inserts into the finished piece.

The earliest plastics, such as cellulose nitrate, as well as the majority of the natural gums and resins, soften to such an extent under heat as to seriously limit their field of usefulness. One of the major factors which has brought about the rapid extension in the use of the phenolic-resin plastics is the extent to which they resist not only heat, but also other conditions, such as moisture, and mild acids and alkalies, which tend to break down the appearance and frequently the structure of many materials.

The history of man has been written in the "ages" of the materials from which his handiwork has been largely fashioned. If we may gage the future by even current progress, we appear to be rapidly entering the plastic era.

Replacement by Resinoid

In reference to the adoption of resinoid materials in new fields, "replacement" is a better word than "substitute." Substitution, according to Fred C. Bowman, writing in *Factory and Industrial Management*, is as old as the days of Tubal-Cain, who persuaded his neighbors to buy his iron tools to replace their stone hatchets, and adds that it probably took all of his 700 years to overcome their sales resistance.

Generally, the bad reputation of substitutes has been deserved. Much effort was spent it making a substitute look like the

---

[1] Technical Editor, Bakelite Corporation. Mem. A.S.M.E.

Presented at the Fourth National Machine Shop Practice Meeting, Chicago, Ill., Sept. 22 to 24, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

TABLE 1 PHYSICAL PROPERTIES OF FABRICATED PHENOL-RESINOID PRODUCTS, MOLDED AND LAMINATED

	Molded			Laminated	
	Wood-flour filler	Fabric filler	Asbestos filler	Paper	Fabric
Specific gravity	1.3–1.4		1.8–2.0	1.3–1.4	
Tensile strength, lb. per sq. in.	Figure-eight test piece, 3500–6500; New A.S.T.M. test piece 6000–12,000	New A.S.T.M. test piece, 4500–9000	Figure-eight test piece, 3500–5000; New A.S.T.M. test piece, 5000–10,000	8000–24,000	8000–12,000
Impact strength (notched Izod specimen), ft-lb. per sq. in.	1–2.5	10–25	0.5–4	5–25	10–65
Modulus of elasticity (transverse), lb. per sq. in. $\times 10^6$	25	8–12	10–25	10–20	5–15
Modulus of rupture (transverse), lb. per sq. in.	10,000–20,000	8000–15,000	8000–20,000	Flatwise or edgewise 15,000–30,000	Flatwise or edgewise 15,000–25,000
Electrical resistivity (volume), ohms per cm. cube at 30° C.	10^{10}–10^{11}		10^8–10^9	10^{10}–10^{11}	10^9–10^{10}
Breakdown voltage (volts per mil), instantaneous at 60 cycles	300–500	200–500	150–400 Mica Filler 450–600	500–1300	200–500
Power factor, 1,000,000 cycles, per cent	1–10	5–10	10–20 Mica Filler 0.7–2.5	1.5–5	2–7
Dielectric constant	4.5–8	4.5–7	5–20	4.5–6	4.5–7
Thermal conductivity, cal. per sec. cm. ° C. $\times 10^{-4}$	4–6		12–20	5–8	
Specific heat	0.30–0.40				
Effect of heat	Tends to harden and shrink slightly with improvement of electrical properties				
Dry	Withstands 250° F. Does not fuse		Withstands 475° F. for short periods Does not fuse	Withstands 250° F. Does not fuse	
Moist	Slight deterioration of electrical properties				
Effect of aging	No deterioration. Tends to improve in mechanical and electrical properties				

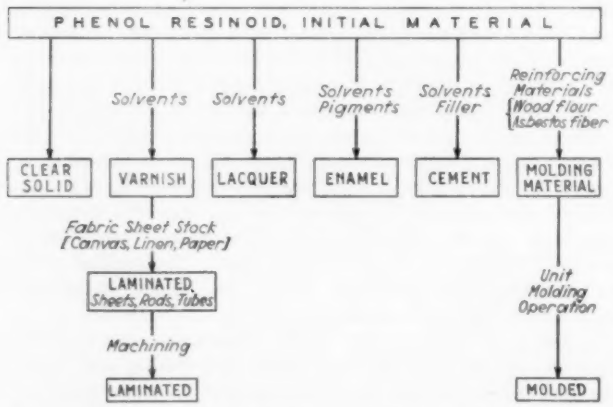


FIG. 1 CHART SHOWING RELATIONSHIP OF PRINCIPAL PRODUCTS FABRICATED FROM INITIAL PHENOL-RESINOID MATERIAL
(For structural purposes, the molded and laminated divisions are of chief importance.)

original and little thought given to making it behave right.

Within the last few years a new kind of substitution has come. Now a substitute must be better than the original. There is no attempt to deceive, and often none to produce resemblance. A mere saving of money is usually not enough to put over a substitute. It must have property merit.

Bakelite materials have many times been adopted for applications in which cast iron, steel, brass, tin, aluminum, lead, and a wide range of alloys were formerly employed. Usually this change of material of construction is based on decision by the designing engineer that the properties of the phenol-resinoid material justify its adoption as superior to the metal formerly employed. This does not mean, of course, that the metals are not highly satisfactory and preferable for many other applications. Block tin is still one of the world's metal utilities, but molded resinoid is fast replacing it for use as the caps for shaving-cream and tooth-paste tubes. This does not indicate that resinoid is a substitute for tin in the way most people would understand the meaning of the word.

The following, in order of importance, are the chief industrial fields using resinoid molding material:

1 Electrical manufacturing
2 Radio
3 Automotive
4 Mechanical
5 Gifts and novelties
6 Electrical, central station
7 Bottle and tube caps, packages
8 Building and hardware
9 Telephone
10 Textile
11 Chemical
12 Miscellaneous
13 Aviation

PROPERTIES

One important phase of mechanical engineering pertains to the adaptation of materials to new form, new properties, and new applications. It is the properties of a material which guarantee or jeopardize its engineering service. In the selection of a material of construction, engineering and economics must be simultaneously considered. Chief factors influencing selection are availability, cost, and properties.

The problem looks simple, but actually it is extremely complex. For instance, there are present properties and future properties. There are characteristics which will justify the application from the standpoint of safety and reasonable cost, but these may be far from the optimum characteristics available. Engineering ethics, and the honor of good craftsmanship, demand that we find for a given structure, machine, or commodity the material of construction which will be, during the life of the unit, the best.

There is only one good word for such a material or condition and that is optimum.

It is useful mechanical practice to seek to resolve, mentally, any object, structure, or machine into its component materials, then to ponder whether the best materials have been employed. Designing engineers apply this reasoning; all engineers will find it interesting and valuable. To recognize resinoid is not always easy. To recommend its adoption in many design problems requires knowing a wide range of properties and appreciating a specialized molding technique. The technical press has devoted some attention to this problem during the past five years. Knowledge of relative advantages and limitations will save metals from being unwisely replaced. With equal understanding, the resinoid materials should be conserved for optimum application.

Materials of Construction

In judging the suitability of a given material of construction, the engineer usually relies on current tables of properties, and adds to this actual service records of the material in typical range of application. In the case of metal, stone, or wood, the problem is easier. Resinoid materials, on the other hand, are made in the laboratory. There is seemingly no limit to the degree to which they can be improved by refinement and chemical alteration. Extensive research is being applied, and within the past two years new molding materials have been developed with shock resistance many times better than that of the best material previously made. There is therefore the problem of keeping one's information up to date. A second factor of difficulty is that it is next to impossible for any one man to know the complete record and service. New uses are being reported every day. The materials are used in literally every type of industry, and sometimes there are as many as a thousand applications in one small field. Because of the unusually large range of uses to which the materials have been found suited, a great many subdivisions of the materials have developed. It is possible to modify resinoid materials with respect to some 30 different properties. The result is that a single company may manufacture as many as 500 distinctive resinoid products. The properties of the resinoid product are also modified by the addition of such materials as wood flour, asbestos, graphite, linen, canvas, and paper. The chart, Fig. 1, will show the principal classifications of products in the resinoid industry.

Because of their relatively great importance in mechanical engineering, the laminated and molding materials will receive chief attention. (See properties chart, Table 1.) It will be the purpose in the remaining discussion to show that phenol-resinoid materials are properly to be ranked with metal, stone, and wood as a class of general-utility materials. They are in a sense a laboratory descendant of the former three. It is because they are somewhat akin to metal, stone, and wood that their combination of properties is so favorable and that their range of application is so broad. They are referred to as metal substitutes, synthetic lumber, and synthetic stone. Such designations are ill advised. They are true in a sense, but they lack conservatism. One should think of phenol resinoid as an individual material always ready to defer to metal, stone, and wood when these materials are better suited to the purpose at hand, and yet frequently able to solve in an optimum manner a question of design where each of the others has experienced difficulty.

Strength of Materials

At this point, there should be before one a rather definite idea of the ability of resinoid material to replace metal on a strength basis. George H. Dowty, in the June issue of *British Plastics and Molded Products Trader*, discusses the resinoid materials as applicable to over 100 airplane parts. He says, "Other things being equal, the material with the highest specific strength will be the most favorable for use in aircraft construction." His figure of merit, namely, the specific strength, indicates the relative load carried for a given weight of the material. It is obtained by dividing the tensile strength expressed in tons per square inch by the specific gravity. On this basis of comparison, he finds that the laminated resinoid materials are second only to steel, as in Table 2.

TABLE 2

Material	Tensile strength, tons per sq. in.	Specific gravity	Specific strength
Steel	60.0	7.85	7.6
Laminated resinoid material	10.0	1.35	7.4
Duralumin	18.0	2.75	6.5
Spruce	2.5	0.48	5.2
Resinoid material (molded)	2.5	1.35	1.85

On a weight basis, resinoid is six times lighter than steel and only half the weight of duralumin. It therefore follows that, weight for weight, the member fabricated in laminated material will be six times thicker than the corresponding steel member and double the thickness of the duralumin member. In airplane construction this design characteristic is considered an advantage, since the additional thickness gives greater rigidity and stability.

While in airplane design the property of tensile strength is considered especially important, designers in most industrial fields find it necessary to regard with equal care the property of shock resistance, or impact strength. The impact strength of laminated resinoid material, such as is used for gear stock, has a maximum value of 65 ft. lb. per sq. in. The average molded resinoid material was in its early years relatively low in impact strength. During the past few years new molding materials have been developed which yield products whose shock resistance is 10 to 20 times greater than the best previous grades, the maximum value now being 25 ft. lb. per sq. in. Any idea that resinoid molded materials are brittle or fragile is erroneous. It is possible to mold a resinoid part of fairly thin section which can be dropped 50 ft. to a concrete floor and suffer no fracture.

The tensile strength of molded resinoid materials is, at maximum, 8000, while that of the laminated form reaches 24,000. It will be seen on the basis of these figures that replacement of steel elements in the transportation field is impossible in installations which require tensile strength in the range of 60,000 to 120,000 lb. per sq. in. For the structure of bridges, rolling-stock carriages, and building frame, likewise for the construction of heavy machinery, steel in its various alloy forms, cast and wrought iron, also aluminum, hold the field well for metals. It is unfortunate that some writers waste time in predicting the immediate entry of synthetic resins as replacement in these heavy applications. The synthetic resins merit consideration in a very large field apart from heavy construction. It is quite true that they have replaced, and seemingly on a permanent basis, and with optimum effect, thousands of light machine elements and commodity parts. Up to the present time resinoid molded units have ranged in weight from a very small fraction of an ounce to 25 lb. The laminated resinoid can be produced in units weighing several hundred pounds. While these figures by no means indicate an upper limit, it will be seen that the resinoid replacement of steel is a factor for consideration by the machine shop and tool room rather than by the rolling mill. By the same token, it must be appreciated that resinoid properties and technique are of interest to men engaged in the production of die-cast metal parts and the machining of all types of metals.

As could be said for almost any material, phenol resinoid is seldom adopted because of any one property; rather for the sum total of its effect in reference to the requirements of a new design.

Typical Applications

A specific example of resinoid is in replacing cast iron. There is a recording machine known as the Egry register. It has two sides with area approximately 7 by 18 in. A depressed, decorative surface is desired; also, embossed reproduction of the firm name. The design must incorporate suitable provision for assembly, also several holes, bosses, flanges, etc., to accommodate transverse spindles and other parts of the operating mechanism. Based on production of the company in question, manufacture of these parts with resinoid made it possible to conserve 5000 sq. ft. of factory floor space which would have been required to accommodate the machinery essential to 23 grinding, drilling, and finishing operations which would be required with cast iron. Likewise, there was a saving of labor cost. In addition, the

weight of the finished machine was appreciably decreased, a factor of importance in the design of office and store machinery. Decrease in weight effected a saving in shipping cost, an economic factor which must also be considered.

Dental chair arms made of pressed steel required a subsequent enameling process. The chair arm was cold to the touch and the enamel too easily chipped. Molded from resinoid material and by means of a single unit molding operation, a strong, mahogany-finish part is produced with metal attachment inserts solidly embedded. The organic nature and low thermal conductivity of resinoid material make it warm to the touch.

Contrasting with the self-lubricating dashpot made from a molded mixture of resinoid and graphite is the grinding wheel application. In this field resinoid is used to bond the abrasive particles. The new abrasive wheel has come along with the demand for rapid accurate grinding. The Bakelite type of wheel is operated at 9000 surface ft. per min. as against 6000 for safe operation of vitreous wheels. Wheels may be produced of open structure especially suitable for snagging and cutting.



FIG. 2 TYPICAL EXAMPLES OF THE LAMINATED RESINOID PRODUCT
(Simple shapes can be molded, but detailed parts like the pinion are produced from disk blanks by hobbing machines or shapers. Process block, lower left, shows resinoid-impregnated canvas layers before and after press operation.)

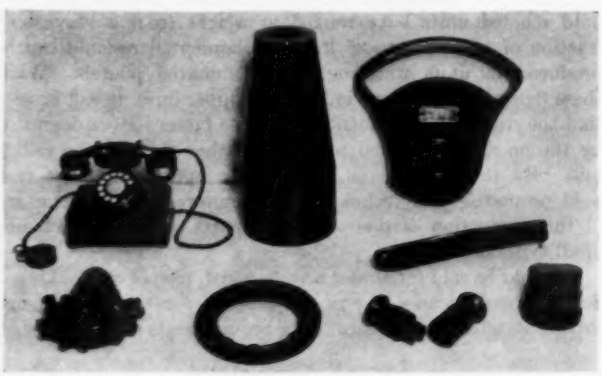


FIG. 3 TYPICAL EXAMPLES OF THE MOLDED RESINOID PRODUCT
(Insulator, center, weighs about 25 lb.)

They cut granite as rapidly as vitreous wheels cut marble, and wear away slower in the process.

There may be doubts as to the strength of molded articles, but the following should dispel them. The Philadelphia subway has adopted a car coupler made from a phenolic molding material. The main body of the coupler is of metal, and has two metal wings, each containing a molded part, which has embedded in it 14 metallic inserts. The inserts are used to make electrical connection, from coach to coach, of the various electrical circuits, and are so spaced that when the coaches are coupled the various electric circuits are completed. The couplings are of the jaw type. These molded parts have been found to stand up to the mechanical shocks of the train assembling yard.

In the textile industry, a cloth tentering machine is equipped with a large number of metal clamps, which are automatically opened and closed. These clamps are actuated by the sliding or rubbing action of the opener. Cast iron, and later, brass, was used for the opener, but both metals caused rapid wear of the clamps. This was regarded as serious because the clamps cost $4.75 each. During the past two years, resinoid laminated

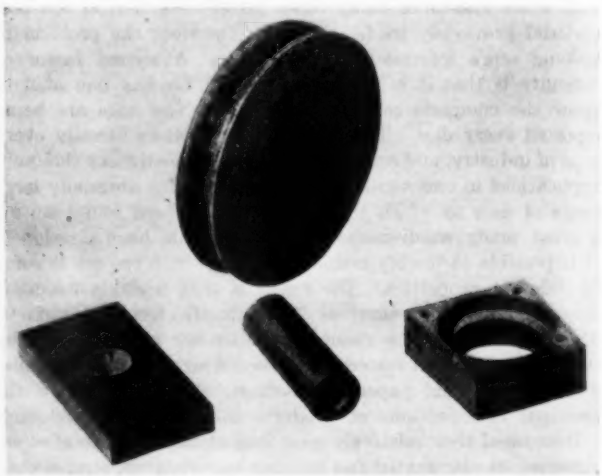


FIG. 4 AIRPLANE ACCESSORIES MADE FROM LAMINATED STOCK
(Guide pulley, control cable bushings, and engine mount shim.)

material has been used for the tenter clamp opener. There has been no discernible wear of the brass clamps. This is a nice illustration of resinoid, not only as a replacement, but as an ally of metal. A similar case is in automotive timing gear transmission, where a resinoid gear is meshed with two steel gears to promote the factor of silent operation. Recent improvement in canvas laminated gear blanks has increased the life of resinoid gears from 15,000 miles to almost that of the car.

Two of the most interesting replacements of brass have involved electric light fixtures of the ceiling type used in office buildings and switch plates for electrical switches and outlets. Both of these replacements are by the resinoid molding material. Resinoid laminated stock frequently replaces brass, a specific instance being the blades of electric fans. Replacement of brass for electric light fixtures eliminated the tarnish and the polish problem. Five years ago manufacturers were incredulous when the assertion was made that brass switch plates would be replaced. Now many millions of resinoid plates are made annually, and practically every new building is equipped with them. They are inert, non-corrosive, and require no polishing. Moreover, while a switch plate may not need to be an insulator, it is thoroughly sensible to make it of an insulating rather than of a conducting material.

Formerly the stem of an industrial thermometer was made of hard rubber reinforced by steel tubing. By molding the device completely of resinoid, six assembly operations are eliminated. The manufacturing cost is reduced 36 per cent. Steel inserts are molded integral with the resinoid in the molding operation. There were additional factors which influenced the choice of material, namely, heat and moisture resistance, electrical insulating ability, and mechanical strength.

For a special type of player harmonica an assembly of cold-rolled steel parts was replaced. In a resinoid only two parts

were required and 40 operations were eliminated. The part now resembles mahogany of high luster and embodies several lines of embossed lettering.

A paper napkin container and dispenser for restaurant use is essentially an end-open rectangular box with dimensions approx-

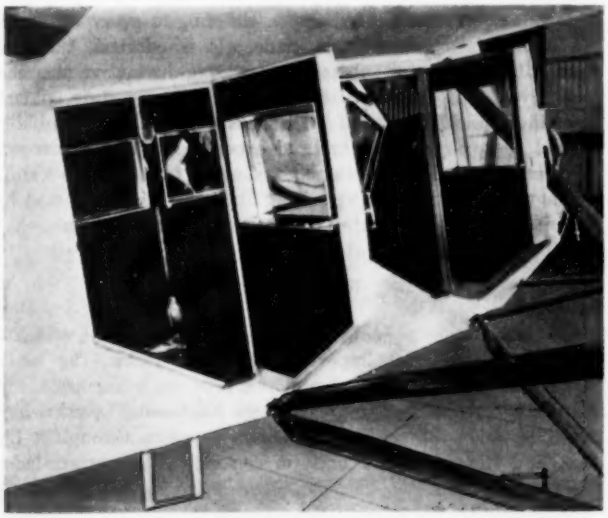


FIG. 5 RESINOID LAMINATED MATERIAL FOR AIRPLANE-CABIN WALL FINISH

imately 5 by 5 by 8 in. There are two large side apertures and a central partition. Resinoid materials in attractive colors have replaced metal because of their ability to yield such a device in a single operation (including the finish), also because there is no need of repeated daily polishing.

In the making of a radio speaker, resinoid, in molded and laminated form, respectively, replaced brass for bobbins and end plate. The metal finishing operations eliminated were, in the two cases, four and seven. In one instance the cost saving was 50 per cent; in the other, 60 per cent.

In a mechanical counting device there is a small ratchet wheel which operates at over 1600 r.p.m. In this service resinoid is reported to have been preferable to die-cast brass. After 30,000,000 revolutions it showed no perceptible wear.

The operating hand wheel of a letter-addressing machine, formerly of enameled steel, was replaced with resinoid rim molded integral with a two-spoke metal spider. The chief reason for the design change was to eliminate the chipping difficulty of the porcelain coating of the steel. A saving of 45 per cent was effected.

Not always is the competition with metal. On a burglar-alarm control box a $^3/_{\text{c}}$-in. slate panel was replaced by $^3/_{16}$-in. resinoid laminated stock. Each slate panel was drilled separately. Three resinoid laminated panels are drilled at once, eliminating 40 drilling operations. There is an appreciable saving in labor and tools. Additionally there is no breakage, and more uniform dielectric strength follows.

Surveying instruments are used in all climates—in the arctic regions at 60 deg. below zero, and in the tropics at 110 deg. and higher. Brass thumb nuts become very hot or cold, and fingers consequently freeze to the metal, or are scorched. Resinoid material, a non-conductor, overcame both difficulties. Operations eliminated on the different parts ranged from 9 to 32, and there was a saving of 40 per cent.

These instances of replacement with resulting design and property improvement, and with production economies of 40 and 50 per cent, are not unusual.

The remaining portion of this paper is intended to answer two questions which the mechanical engineer and mechanic are asking most frequently; namely, "Just what is this resinoid stuff, and what does one need to know about its technique or shop practice?"

WHAT IS RESINOID?

Phenol-resinoid material receives its name because chemicals known as phenols are used in its manufacture and because the

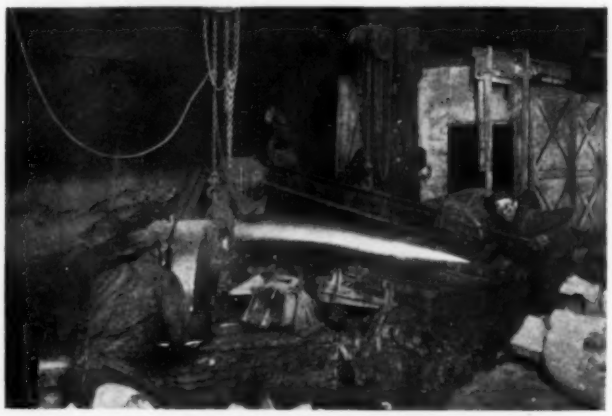


FIG. 6 HIGH-SPEED GRINDING EQUIPMENT
(Resinoid bonding wheels are being safely run at a speed of 9500 surface feet per minute.)

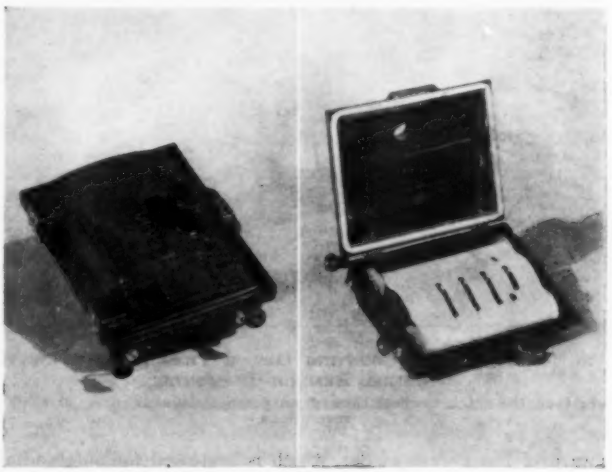


FIG. 7 EXAMPLE OF MECHANICAL ADVANTAGES WHICH RESINOID MATERIALS AFFORD
(The unit molding operation provides not only the channels, ridges, and apertures characteristic of the case interior, but yields also the grain leather exterior finish, and the embossed lettering.)

essential resulting products are somewhat like resins. It should be understood that a resinoid is not one of the natural resins like rosin, copal, or dammar. Resinoids are made in the laboratory by the building-up process of organic chemistry. This building-up process is the opposite of analysis. It is called synthesis. Synthetic chemistry is perhaps the highest form of chemical science because it is the most creative. Just as one develops an alloy steel which will have certain properties, so one can build in the laboratory a super-resin which will meet specifications better than any such material previously known.

Natural resins are useful for many purposes, notably in varnish making. The phenol-resinoid materials go beyond this to become materials of construction. Chemists in the latter part of

the Nineteenth Century knew that there was some possibility of making a synthetic resin from the reaction of aldehydes and phenols. Until the work of Dr. L. H. Baekeland, however, the world waited for an industrially successful super-resin which could serve as a material of construction. Announcement of the invention was made in 1909. When it was recognized by the

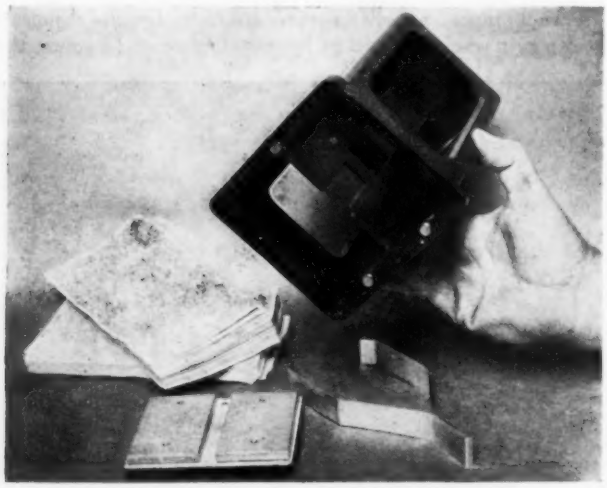


FIG. 8 MOLDED RESINOID NAPKIN DISPENSER
(This retains its luster with no necessity for polishing. The device is well designed from the standpoint of ease of manufacture.)

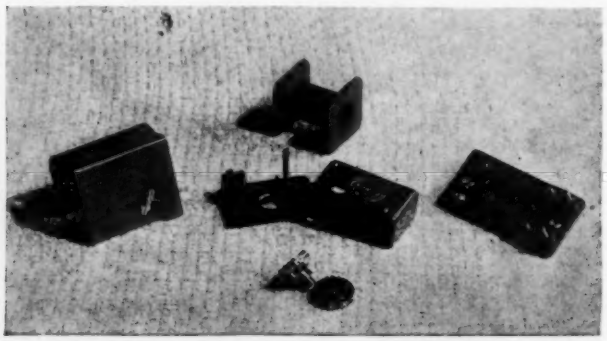


FIG. 9 MECHANICAL COUNTING DEVICE THAT INVOLVES SEVEN MOLDED RESINOID ELEMENTS
(Test on the ratchet wheel showed no perceptible wear after 30,000,000 revolutions.)

award of the Perkin medal, which is bestowed for outstanding industrial achievement, Dr. Charles F. Chandler, of Columbia University, epitomized the significance of the invention as follows: "Before Baekeland had published the results of his work, there existed not a single industry, nor a single application of any importance based on these condensation products. Since then, there have been started here and in Europe numerous factories where these processes are used for the most varied purposes, ranging from the manufacture of a billiard ball to that of a wireless telegraphic apparatus; the manufacture of a self-starter for automobiles to a wide range of uses, embracing such articles as switchboards for battleships, moldings for kodaks, armatures and commutators for dynamos and motors, telephone receivers, railroad signals, grinding wheels, etc."

In its physical appearance, Baekeland's product resembled the natural resins. Likewise in its initial state, it could be melted readily and dissolved. But, in a later state, it became quite different in properties. The agency of this change was heat; the process is described by the chemist as *polymerization*. When this change has been completed, the resinoid is radically different from natural resins. In careful appreciation of the physical resemblance and chemical dissimilarity, Baekeland adopted a distinctive term for his new material. This word is resinoid.

Under certain well-defined conditions two odoriferous, unpromising, liquid solutions of phenol and formaldehyde combined to form a solid resin-like substance, differing in every way from the liquids from which it is made. It is odorless, tasteless, inert—an entirely new substance possessing distinctive chemical and physical properties. In its raw or primary state, phenol resinoid is quickly softened by moderate heat and is soluble in alcohol or acetone. A very brief exposure to greater heat sets up a further chemical reaction and causes it to solidify to a state which will not permit of remelting and which is unaffected by solvents. Thus the material is thermo-plastic. More than this it is heat-reactive and heat-transformed.

THE LAMINATED TYPE

To manufacture phenol-resinoid laminated material, which is one of the two most important industrial forms, the initial resinoid is dissolved in solvents to provide a varnish. The varnish is then used to impregnate the laminating fabric, which is paper, linen, or canvas. The fabric passes through a dip tank and thus secures a uniform coating. The varnish-laden sheet passes from the dipping operation to drying apparatus. After being dried of solvent, the processed web is cut into sheets 100 in. sq., or of other dimension, as desired. A number of these sheets are then superimposed and placed in an hydraulic press operating with 350 deg. fahr. and 1500 lb. per sq. in. Under this action of heat and pressure a tough, dense, rigid slab is produced. If one of these slabs is examined it will show relationships and differences compared to wood, metal, and stone. Aside from its application for gears and pinions, this material is used for thousands of electrical and mechanical accessory purposes. Most recently it is being adopted for wall-paneling and in furniture manufacture. Tables and desks in modernistic design are composed of steel tubing and sheets of resinoid laminated material. Laminated pinions, adopted chiefly for the factor of silence or to resist atmospheres containing corrosive chemicals or abrasive particles, are used on motors rating from 1/4 to 100 hp. The drive is applied to practically all types of

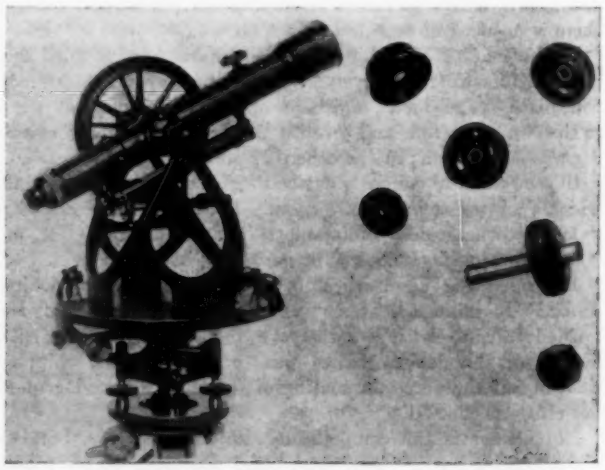


FIG. 10 EYEPIECE AND ADJUSTMENT WHEELS ON SURVEYOR'S TRANSIT MOLDED OF RESINOID MATERIAL
(Low thermal conductivity, in distinction to brass, is considered an advantage. Moreover, a unit molding process replaces 9 to 32 operations required with metal.)

shop equipment such as planers, milling machines, high-pressure pumps, broaching machines, etc.

By reason of its layer structure, the laminated material is generally stronger and tougher than the molded product. Moreover, the layer structure prevents its being molded to intricate shape. When applied to applications which are essentially flat, it requires very little processing. Otherwise it requires the usual machining operations which are applied to wood, metal, and stone. It will be understood, therefore, that this material replaces the older materials of construction on the basis of its chemical or physical properties, rather than from the standpoint of production economy.

The Molding Material

With the molding material the case is quite different. It offers all the properties of the laminated material with the exception that it is less strong mechanically. Its greatest advantage lies in the fact that it requires no machining, being converted by a single rapid molding operation into finished product form.

Molding materials are usually furnished in powdered form. This powder is supplied ready for molding in steel dies. It requires no mixing or modification by the consumer. Molding

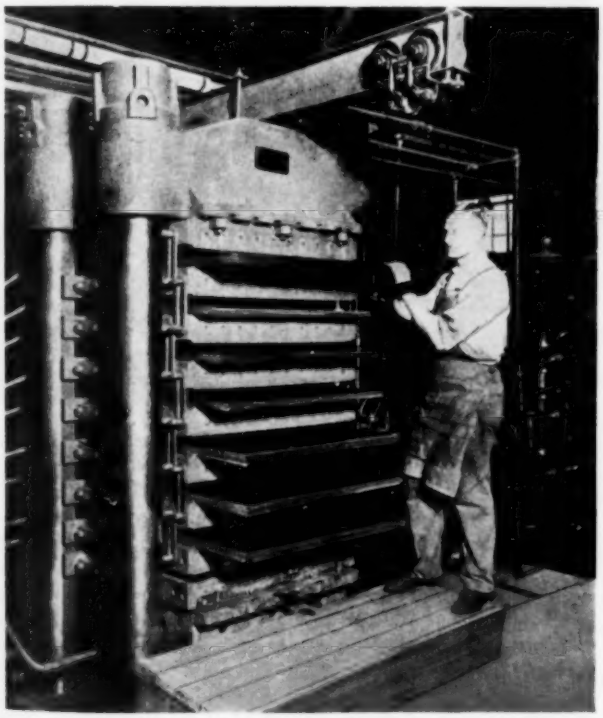


Fig. 11 Multi-Platen Press at 1500 Lb. per Sq. In. Bonds Resinoid-Impregnated Sheets of Canvas, Linen, or Paper Into Slabs of Bakelite Laminated

(The finished stock is adapted to thousands of uses, ranging from silent timing gears in the automobile to wainscoting in modern buildings.)

materials embody the resinoid material—in such state that it will be further converted during the molding operation—plus reinforcing material and color pigment. Reinforcing material is usually wood flour or asbestos fiber. It is not added particularly to cheapen the commodity, but rather to provide better molding qualities, greater toughness and strength, or additional heat resistance. For manufacture of dashpots, the addition of graphite imparts a marked degree of self-lubrication.

Die-Molding Production

As applied to metals, the term "die casting" is correctly defined as the art of casting molten metal into dies, these dies usually being constructed of special irons and steels. Hence in a broad sense this term embodies the processes of both "gravity" and "pressure" casting. The gravity method relies upon the force of gravity to carry the molten metal into the various cavities of the dies and therefore must be used in connection with metals of rather high fluidity. The pressure method employs the application of external pressure, either air or mechanical, or a combination of both.

From what has been said, it will be understood that resinoid molding is not a "die casting" operation in the usual sense of the term. There may be gravity effect after the start of the



Fig. 12 Bakelite Laminated Sheet Stock Used as Wall Finish

(The paneled enclosure of this filing department is an example. The ceiling light fixtures are an application of the molded product.)

molding operation, depending on the mold, but essentially the operation relies upon mechanical pressure. The charge is solid rather than liquid. Usually it is dry powder, though for purposes of convenience in loading, this powder may have been preformed into tablets or sheets. Mechanically, the molding process suggests the molding of hard rubber, but the process analogy is only partial.

Resinoid molding material might be defined as a general utility, partially converted, intermediate, raw material adapted to final conversion in dies operated with heat and pressure. It is not extruded, nor cast, nor is it "die cast" in the same sense that these terms apply in the molding of metal.

By the resinoid hot-molding operation, mechanically, the material is not merely set up in a new form, but its nature is changed chemically. In other words, as the molding operation progresses, the charge not only becomes different in form, but it is converted to a new material. The importance of this point cannot receive too much emphasis. Upon the understanding of it, depends success in resinoid molding technique and satisfactory product performance.

Hardened steel dies are used in molding resinoid materials. The die cavities are produced by hobbing or by the usual machining operations. Molding presses are usually hydraulic. Normal operating pressure is 1 ton per square inch. Temperature is normally 350 deg. fahr. These figures are not arbitrary; special material may require higher pressure or lower temperature. Steam is the popular heating medium, but gas and electrically heated platens are also common. The molding time for different resinoid materials covers an appreciable range. There are

hundreds of molding materials, hence a wide range of flow rates, color, and properties in general. Principally, however, molding time depends on weight and structure of product. Dies may be "hand," or mounted on the press, "semi-automatic." They may have one cavity or many. Dies are further classified as positive, semi-positive, and overflow.

The overflow mold is made so that the upper and lower portions rest one on the other as they are brought together under pressure. In the positive mold, the top plunger telescopes within the lower portion of the mold. In the overflow mold, the molding powder, or a preformed biscuit of it, may be used, provision being made for the excess material, or flash, to be squeezed out as the die members contact. In the positive mold, having no provision for overflow, the mold charge must be still more accurately apportioned.

The powder materials compress, in the molding process, to about 40 per cent of their original volume. In other words, to produce a flat tablet $^1/_2$ in. thick, the charge depth will be $1^1/_2$ in. The molding material suffers no appreciable loss in weight during the molding process. With well-designed molds and proper technique, molding accuracy within 0.002 in. is obtainable. For most purposes, tolerance of 0.005 in. is found adequate.

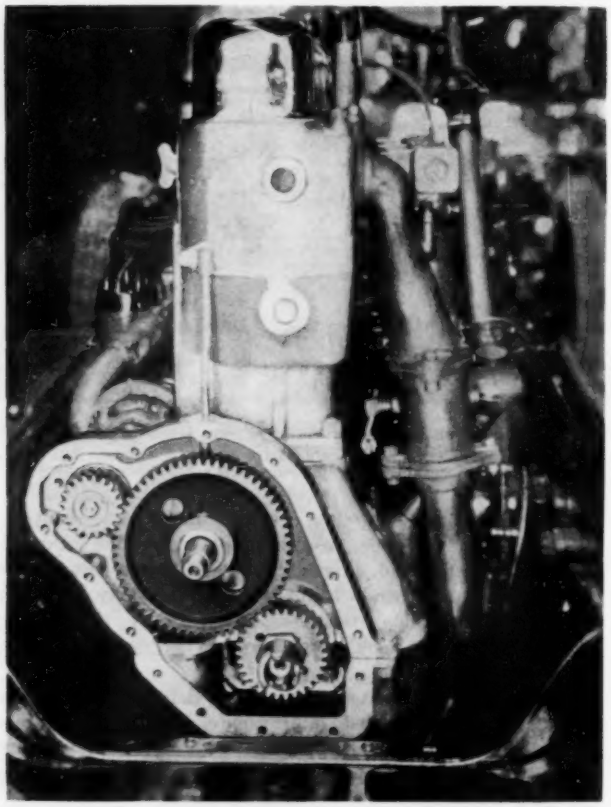


FIG. 13 THE MAJORITY OF AMERICAN AUTOMOBILES CONTAIN AT LEAST ONE LAMINATED RESINOID GEAR
(Most frequently it is in the timing mechanism and is installed intermediate with two metal gears.)

Knurling, embossed effects, bosses, grooves, holes, and screw threads are nicely obtainable in the molding process. Where special thread strength or bearing reinforcement is desired, metal bushings are incorporated. The embedding portion of the metal bushing is given a knurled grooved surface to promote adhesion.

SHOP PRACTICE SUGGESTIONS

A notebook record of service engineering experiences during the past five years contains many interesting incidents. Typical entries are the following:

(1) The premature hardening of a portion of a mold charge to the point where it is no longer plastic under heat, is usually called pre-curing. Pre-curing results in molded parts of poor strength and finish. It is caused by allowing the charge to stand in the mold for several minutes before being placed in the press. It is a simple matter to avoid pre-curing. The molder should put the mold in the press as quickly as possible before loading. He should also close the average mold within 15 to 30 sec. from the time it is placed between the platens of the press.

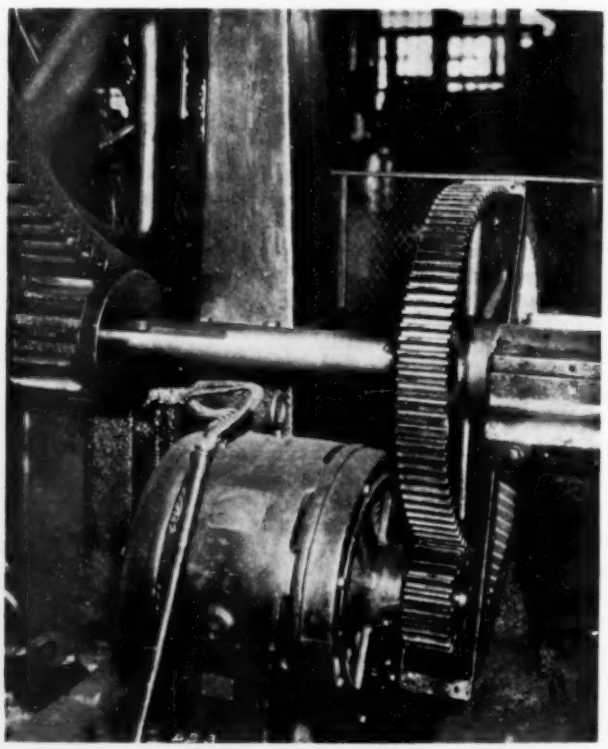


FIG. 14 SMALL BAKELITE GEARS USED FOR LIGHT-LOAD SERVICE
(The Bakelite pinion shown is mounted on a 100-hp. motor which drives a boring mill.)

(2) A molder who was producing jar covers was obtaining production of but 130 per day. A stopwatch study of loading and curing time made by a service engineer revealed that it was taking $7^1/_2$ min. for each cycle. The press platen was too high and the arbor press too far from the molding press for easy and efficient handling of the molds. A platform was built in front of the presses and the distance between arbor press and molding press was reduced. The result was that production was increased to 190 covers per day, a gain of 45 per cent.

(3) In molding phenol-resinoid materials gases are given off under the heat and pressure of the molding operation. If a ready escape is not provided for these gases, the molded piece is apt to have interior gas pockets and surface blisters. This trouble can be overcome readily by the practice known to the trade as "breathing" the mold; that is, releasing the pressure for a period of 3 or 4 sec. after the mold is first closed. Preheating the molding tablets for 10 or 15 min. at a temperature of 180 to 190 deg. fahr. will also be found a very effective means for checking the tendency to blister. An "air-bound" mold is

another fruitful cause of blistering. Such a mold is one in which the top force fits into the chase so snugly that there is no means of escape for the heated air or gases given off in molding. This occurs only in positive molds of the "straight-draw" type. To avoid air binding, the mold force should be made with a free sliding fit in the chase and should be provided with sprue grooves to allow for the escape of gases.

(4) In 1926 a company producing 5000 tube bases per day was using the following cycle: (*a*) Load, discharge, 1 min. 20 sec. (*b*) Closing mold, 40 sec. (*c*) Cure, 2 min. A service engineer demonstrated that the 2-min. cure could be reduced to $1^1/_4$ min. by puffing. Thus they included in their operation a slight opening, or release, of the mold about 10 sec. after the first closing, and standardized on a cure of $1^1/_2$ min. In 1930 the cure requires less than 1 min.

(5) Whenever possible the practice of using a small mold in a large press should be avoided. The excessive pressure applied on the small surface of the mold may drive the mold into the platen of the press and score it badly. When it is absolutely necessary to use a large press with a small mold, it is a wise precaution to place at the sides of the mold steel blocks of approximately the same height.

(6) One of the best speed records for hand-mold operation is 600 pieces per hour from a single press, using two nine-cavity molds. Time in the press is 40 sec.; transfer, 5 sec. Wall thickness of molded part is $^1/_{16}$ in. In considering molding speed records, it should always be remembered that excessively short curing time does not impart properties claimed for phenol-resinoid material.

(7) In an instance of alleged shrinkage, with finished piece undersize, it was found that the gas-heated presses were operated at too high a temperature. The alleged shrinkage was really warpage. The molded part had a sizeable central slot. Use of shrink plugs eliminated the trouble.

(8) The use of hard water will cause coating of the steam channels in the mold, and this, by interfering with the thermal conductivity of the metal, will influence the curing and chilling effect, and alteration in these factors will cause surface defects on the molded piece.

(9) In molding a typewriter space bar, poor surface, due to insufficient cure, caused by poor conduction, was traced to the fact that the mold had warped. A thin film of resinoid material had become located between the bed plate and the mold. This acted as a heat insulator. The die was machined, and relocated with good contact surface, and the trouble was eliminated.

(10) In molding a coil cap, a void was formed in the center of the molded section. Material was not flowing properly into place because the force of the mold having long pins in it was allowed to rest near an open window for several minutes during the loading operation; consequently it became so cooled that flow of material was retarded. The trouble was corrected by keeping both top and bottom mold parts hot. Proper molding was facilitated by heating preformed disks for one hour at 210 deg. fahr.

(11) It is good practice to use an air hose to clean off the mold between cycles. Care must be taken, however, not to blow impurities into the supply of molding material and not to blow contamination into recesses of the die. It is sometimes found that the pump supplying the air pressure has its intake located near the floor. An air filter should be included in the air supply line.

(12) In the molding of an electrical instrument case blistering occurred, but only on one side of the part. The material was blamed. A service engineer found that a small amount of snow was drifting through a ventilator in the roof and settling in the open cavities of a tilt-head press mold. When the ventilator was closed, the blistering trouble disappeared entirely.

(13) On a 12-cavity mold for a cylindrical automobile part, four cavities had been left empty because of broken pins. Operators were equipped with five sets of shrink plugs for each cavity, and they stated that on taking from a given cavity five successive parts, differences after shrinking on the plugs until cold were between 0.003 and 0.007 in. The die cavities differed slightly; hence the different sets of marked shrink plugs. In spite of marked indication, the operator was not taking any particular care to put the part from a given cavity on the right plug. Proper use of shrink plugs removed trouble.

(14) The manufacturer of an electrical device requiring exact dimension and high polish complained of steel particles in one drum of molding material. It was found that a workman had been using a steel screwdriver to clean mold cavities. Three cavities in the mold have been badly scored, and would require repolishing. Having ruined the finish the workmen had reported that the molding material contained metallic impurities.

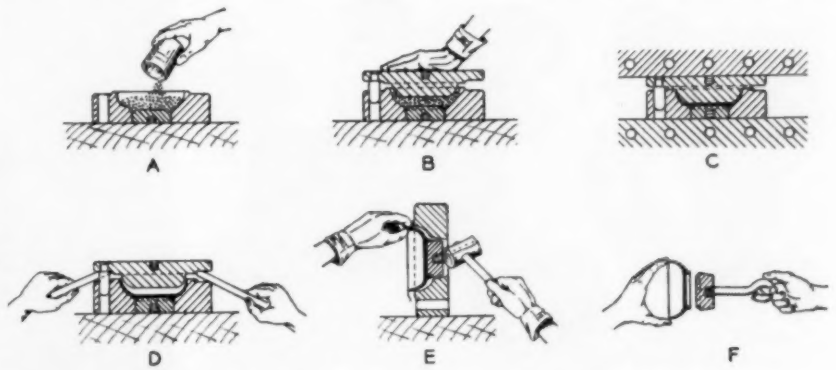


FIG. 15 THE MOLDING OF A RESINOID PART OF SIMPLE DESIGN

(The operations shown are analogous in principle with the machine methods involved in quantity production. *A*, fill mold with requisite amount of molding material; *B*, press down with hand; *C*, cure in hot-plate press for about 43 min.; *D*, remove the top force; *E*, tap the ejecting plug and the molding from the mold; *F*, remove molding from the ejecting plug.)

(15) Large silver-gray spots, appearing without apparent cause on surface of molded parts, were attributed to oil or water having passed through the air hose.

(16) A large three-cavity positive mold for the production of parts with thin, deep-wall section requires a "soft" material with quick flowing properties. In the instance reported, sticking occurred in the middle cavity. On checking with a pyrometer, it was found that after a period of 45 min. under steam, the end cavities were 300 to 325 deg. fahr. The center cavity registered 250 to 270 deg. fahr. These molds were being operated with chill, so the production temperatures would be still lower. It is essential that molds be designed so as to maintain the same temperature throughout all cavities.

(17) A fast-curing molding material was found to take an unusually long time to cure. Investigations showed that the drum was shipped and stored during a blizzard. When the material was put in the mold, a long time was required to heat it through. The material being a good heat-insulating medium, it was naturally slow in reaching room temperature before molding.

(18) Misaligned mold pins are always a source of molded part cracking or breaking.

(19) A molded tool handle was found to be breaking in service. Examination showed that the resinoid material was under-

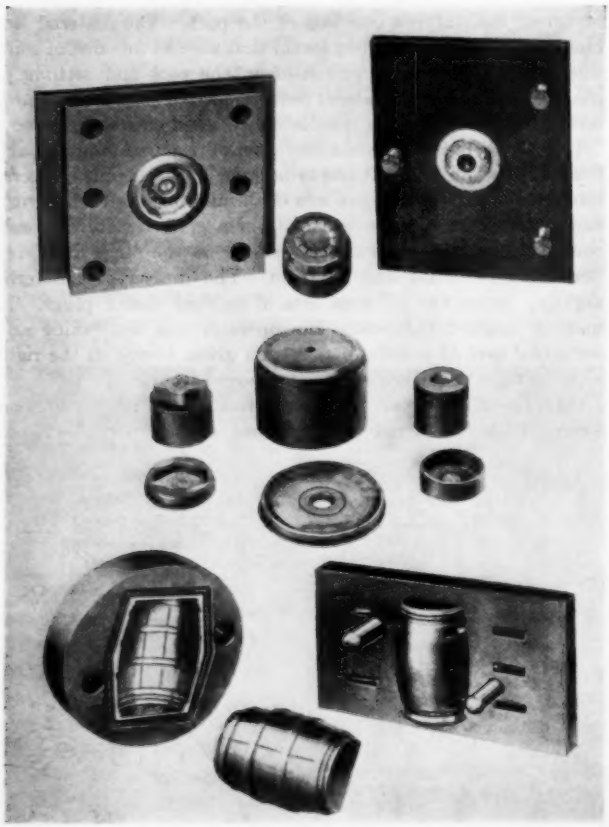


FIG. 16 GROUP OF MOLDS AND HOBS USED IN FORMING THEM

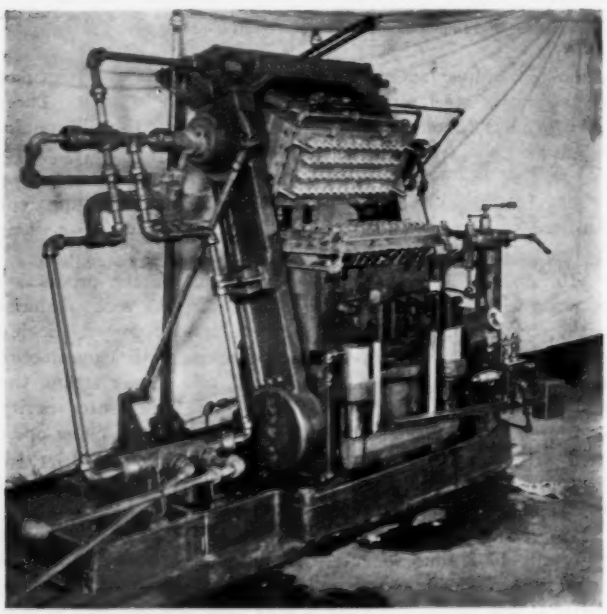


FIG. 17 TILTING-HEAD MOLDING PRESS

cured. Moreover, considerable material had escaped from the flash mold instead of being compressed into the finished handle. The shape of the preform was changed, and the time of cure was increased to 6 min. The tablets were pre-warmed, and the mold was closed slower to prevent escape of material. A handle, thus made, could be pounded on the floor until the metal part was seriously deformed, and the handle was unimpaired.

IMPROVED METHODS AND MATERIALS

Rather than discuss equipment and technique of the resinoid

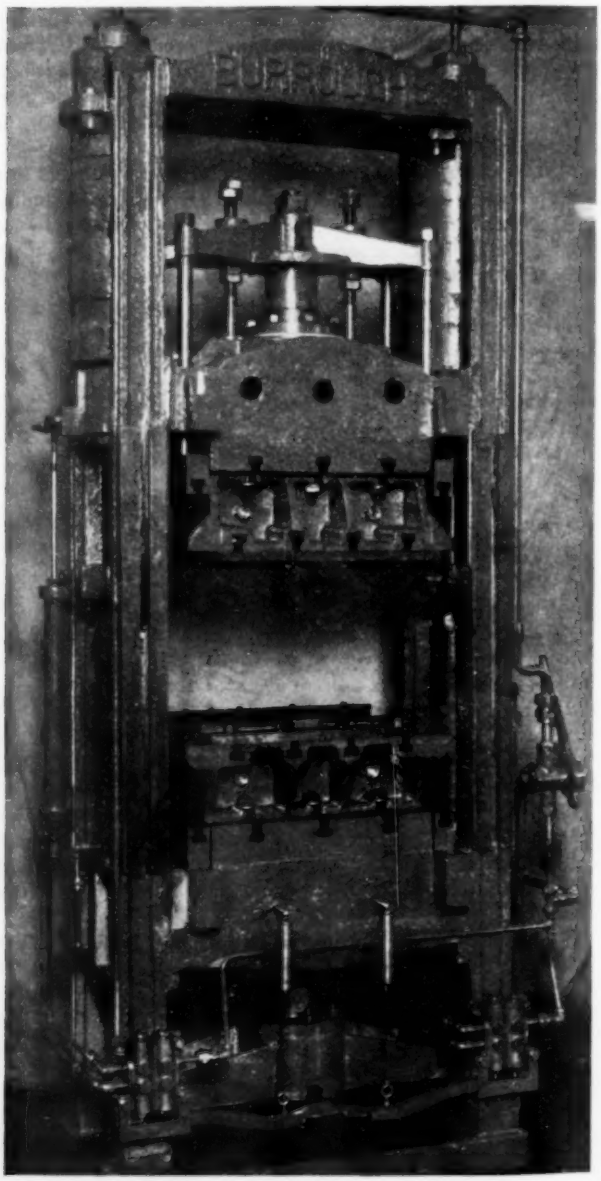


FIG. 18 HYDRAULIC SEMI-AUTOMATIC MOLDING PRESS

molding plant as if it were a new subject, some of the technical improvements which have been responsible for the remarkable advance in production and application during the past five years are listed.

1 Presses:
Automatic—increasing production speed; compared with hand molding.
Tilting head—permitting time saving and easier loading.
Rodless—minimizing danger of platen misalignment.
Mechanical (molding machines)—eliminating hydraulic system.
Duo—Turntable design permitting heating and chilling operations

quickly continuous without hand transfer. Hot and cold operation confined to respective press units.
Mechanisms, control—eliminating hand operation of press cycle.

2 Molds:
Multiple—molding a hundred parts instead of one in equal time.
Chromium plated—protecting surface, facilitating release of complex parts; also an aid to hobbing.
Hobbed—extending application for mass production of multiple cavities.
Design improvement—reducing flash to a minimum.

3 Preforming equipment:
Rotary—notably advancing tablet production per minute output.
Percussion—heavy construction eliminating frame breakage and producing tablets of larger size.

4 Plant:
Production line—sequence improved, and conveyor systems increasing speed and convenience.
Press tables—at same height with press, keeping work on same level.
Loading frames—permitting rapid charging of multiple molds.
Line connections—maintenance of standardized temperature and pressure minimizing leaks; copper pipe spiral connections sometimes preferable to swing joints.

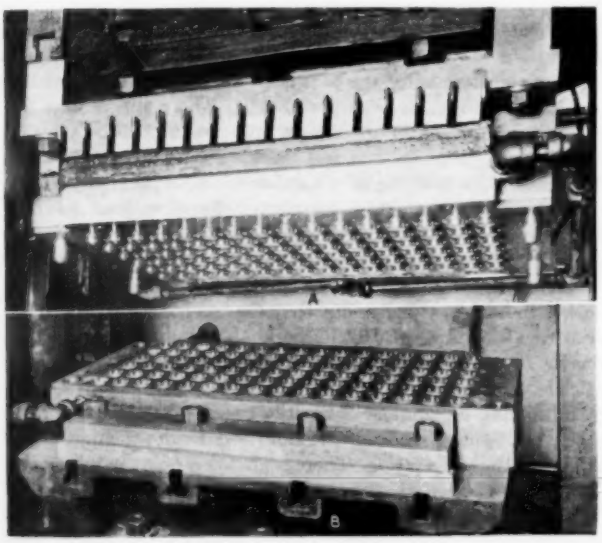


FIG. 19 UPPER AND LOWER DIES FOR PRODUCTION OF A BOTTLE CORK COVER

(*A*, upper die; *B*, lower die. These are mounted in a 100-ton semi-automatic press. Work is stripped from forces of the upper die by a stripper plate moved downward by the ejector cylinder. The mold is direct-heated and contains 96 cavities. The cup-shaped moldings produced are 1 in. diameter and 9/8 in. depth. Production is 23,000 pieces per day.)

Instruments—increasing use of indicating and recording types.
Measuring equipment—electric scales for accurate charge weighing; rotary measuring machines for preparation of identical charges of powder material.
Material-handling methods—involving segregation of colored materials, and the protection of molding material, generally, from contamination.
Safety devices—automatic press guards, etc., preventing injury of operators.
Air hose—slender neck type permitting insertion to reach die cavities; installation of dust filters.

5 Materials:
Development of a material possessing increased shock resistance—twelve times that of the regular molded product, itself a good shock-resisting material. Airplane pulleys and camera cases are typical applications.
Development of materials which can be drawn hot. Equal number of cold presses eliminated, doubling plant capacity. Time saving results because a 9-min. cycle may now be two. This latter point means that a molding plant of given size is capable of at least four times its former output.
Development of an entire color series supplementing the blacks and browns which long prevailed.
Development of an improved heat resistant material by which the heat resistant range of the molded piece has been raised from 250 deg. fahr. to 400 deg. fahr.
Development of material with increased resistance to the carbonizing effect of the electric arc. Endurance aircraft have used it for their magnetos.
Development of improved "low loss" material for radio. This yields a molded piece with very low power factor, and high resistance; suitable, therefore, as insulating parts of high frequency apparatus.
Development of a material offering increased flexibility. This seems to solve the problem arising from difference in expansion between a reinforcing metal core and its molded resinoid covering. An important use is found in the automobile steering wheel. Integrally molded, a resinoid steering wheel with metal spider is reported to be three times as strong as the rubber rim wheels, and from two to three times as strong as one made from wood.

6 Material application trend:
Coupled with shop-practice improvement and material development is the rapid extension of molded resinoid parts to mechanical uses involving relatively large forms. Instances include soda fountain ice-cream cabinet covers, lunch-counter stool tops, office-machine sides, electric-fan bases, subway-train coupler parts.

CONCLUSION

In this review an effort has been made to show the origin, comparative properties, and present industrial position of phenol-resinoid material. This resinoid class of substance which is a product of the research laboratory has been considered as a major class among materials of construction. It has been ranked, by reason of its versatility, with wood, stone, and metal. Resinoid has been so nominated, after consideration of its rapid advance, permanence of achievement, and future promise—particularly with regard to its unusual combination of properties adapting it to every field of engineering and in consideration of its thousands of important industrial applications.

Emphasis of the spectacular has been avoided. The best services of resinoid have been found to supplement metal and the older materials of construction. Every effort has been made

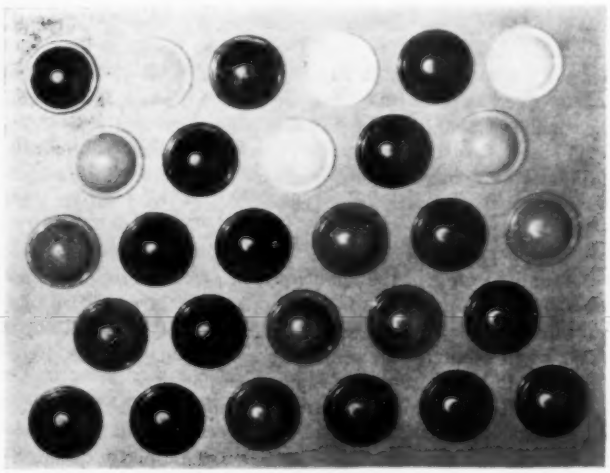


FIG. 20 RESINOID MOLDINGS IN A WIDE RANGE OF COLOR

(In addition to standard one-tone shades, wood finish mottles such as walnut and mahogany are available.)

to prevent overestimation or incorrect interpretation of the rôle of replacement. Resinoid is not a universal cure-all, but it is one of the most useful material allies of modern progress. Effort has been made to indicate and to discuss the engineering factors which govern its adoption in new fields.

One of the chief problems confronting twentieth century organization is that of coordinating the complex activities produced by the past generation of industrialism. Resinoid material, by reason of increasing use, is creating new departments in mechanical shop practice. Its fabrication, especially the unique process of molding, is a natural supplement to older operations of the metal trades.

The history of humanity is a record of transition no less in materials than in mankind. Particularly is this generation replete with change. There is unparalleled attention to research, which is deemed necessary to survival. Research is responsible for new materials and their application. The total effect of material transition, either upon industry or upon the workers, is impossible to assay. Sometimes resinoid replaces steel. Again conjoined with sheet steel, it replaces stone, and opens a new metal market. Gleaming corridors of mammoth buildings bear evidence. Optimum application of material must involve conservation as well as efficiency. A study of

the engineering factors involved in material replacement becomes, therefore, increasingly important.

Discussion

W. S. Huson.[2] This paper marks an unusual presentation of a very live subject, in that it enters so fully into intimate detail of manufacture and application and, too, because it is a material which at first glance would seem remote from a place usually assigned to the metals.

However, as the author has pointed out at length, where pieces formerly considered only from a metal base can now be made of a material so different and yet adaptable to many used heretofore considered as of metallic possibility only, the paper will undoubtedly broaden concept and use as study is given the material and its economic value in the industries.

There is a phase of production on which the author touches rather lightly, and that is the machining of resinoid materials and the effect on cutting tools. The material is very abrasive in its action on cutting edges, and while not seriously affecting machining, nevertheless retards it and is a factor in cost consideration. Here steps in another laboratory accomplishment, cemented-tungsten-carbide tools, which are so hard that the abrasive nature of resinoid has small detrimental effect on the cutting edges, so little that parts are produced much faster than when made of metal. It is well this paper has been presented, for it emphasizes how materials other than those customarily considered as being the only available are coming into larger use and in many ways better service.

W. F. Lent.[3] The paper is an able and comprehensive review of the fundamentals of the application of synthetic resins. The writer has some comments on the application of molded products which apply as well to the phenol-resinoid type mainly discussed as to other types of molding compounds such as urea, celluloid, cellulose acetate, cold molded, and others.

As the author brings out so clearly, in application engineering one must simultaneously consider both engineering and economics. And when these two phases are properly balanced there is truly an optimum application. The solution of the economic phase, which is perhaps the least understood and appreciated, will remove the most serious obstacle to more rapid expansion of applications.

The fundamental nature of the molding art demands a relatively high preparation cost (molds and fixtures) prior to the initiation of economical manufacture. The typical case therefore presents a problem of economic balance between high tool cost and savings in production. This is generally a complex balance and involves many considerations of apparently trivial details of design of molded parts as well as of the tools for making these parts.

First, regarding the design of molded parts, it should be realized that it is often very easy to produce a part by machine methods which it may be very difficult to mold. This involves a consideration of under-cuts, side holes, use of inserts molded in, dimensions of sections, location of holes, proportions, etc. These are the every-day problems of the molder and have been treated in detail in current literature. They are extremely important for successful and economical manufacture. The designer should realize that apparently slight concessions in these details have a powerful bearing on cost.

Second, regarding the tool equipment, there is always the economic balance, here stressed again, of the amount to be expended on tools versus a counterbalancing lower product cost. Here there are two considerations—how the mold unit is to be laid out most advantageously and how many units (or cavities) will be most economical over all.

The layout of the unit determines whether a positive, semi-positive, or flash type of construction will be used, in what position the part will be molded and how the fins will appear. Of course this problem must be considered in connection with the detailed design of the molded part, and it is mainly for this reason that the detailed design has so important a bearing on cost. Flash molds are cheap to construct and easy to operate, but they do not necessarily produce the best parts. The fins appear laterally, entail a slow operation for removal, and are apt to leave objectionable marks when removed. Contrast this to the vertical fin from a semi-positive or positive mold which may be removed by a simple rub on an abrasive surface. The cost is by no means all there after the part has been molded, although the most skillful work has been performed. There still remain the fin-removing (cleaning) operations, and possibly drilling, reaming, tapping, grinding, and inspecting—all classed as finishing operations. An analysis of a number of small molded parts in large quantity manufacture for electrical applications discloses aggregate cost of finishing operations ranging from 20 per cent below to 20 per cent above the molding cost. A prominent hot molder recently said that he had been employing about one finisher to one press hand, but that the tendency seemed to increase the ratio toward two finishers. The point to be noted is that the layout of the mold has a vital influence on finishing cost, which is an appreciable item in total cost.

Multiplying the number of cavities in a mold at least approximately multiplies the cost of the mold and divides the unit molding cost in the same ratio. There is always a point in number of mold cavities above which or below which the cost of total parts to be made plus tool cost is greater. More effort should be put into the correct determination of this most economical number of cavities. It is indeed surprising in studying proposals from different sources working with the same information, to note the great variation (often as much as three or four times) in number of cavities in molds.

Serious thought is apparently necessary in preparation to manufacture molded parts. The design of the part must be suitable and the layout of the tools correct. Once committed to a design and layout, a change is costly, as it may necessitate reconstruction of tools.

The problem of those interested in the sound expansion in the use of molded products is to get designers and development engineers to think in terms of the possibilities of the application of such parts. Effort is necessary to educate them in the physical properties of the materials and in the general methods of fabrication. The latter is not an easy matter and probably will never be wholly accomplished, as economical design is dependent on many fine points of technique, a complete knowledge of which can be had only by those in intimate contact with the design and production of molded parts.

The writer would say that the recommended procedure is to call in the molder in the early stages of the design. Give him all details of what is to be accomplished, particularly the relation of the molded parts to the complete assembly. Work the problem out with him frankly and do not withhold "secrets" for fear of trade information or patent data leaks. There is a well-defined code of ethics to be followed in such cases. Failure to be frank surely limits the possibility of material savings in adaptation of molded product design to specific conditions.

[2] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.
[3] Cutler-Hammer, Inc., Milwaukee, Wis.

MSP-53-4

Repair of Worn Parts by Electrodeposition of Iron

By T. P. THOMAS,[1] EAST PITTSBURGH, PA.

In this paper a commercial method is described for salvaging worn or undersized steel parts, such as thread and plug gages, arbors, reamers, motor shafts, gear centers, etc., by electrodeposited iron. Details of the process have been worked out which provide a practicable set of instructions for shop use. The high current efficiency and rate of deposit of the warm concentrated sulphate bath will enable the plater to complete practically any salvage job in one working day.

IN THE past few years considerable attention has been given to a means of salvaging valuable machine parts in which certain critical dimensions were undersize, due either to wear in service or to errors in manufacture. The reclamation of such material by applying an adherent coating of electrolytic iron began in England and France during the recent World War. Those most concerned in this development were B. H. Thomas[2] and W. A. Macfadyen.[3] The British were the first to make practical application of the electrodeposition of iron for this purpose, applying it to the repair of worn automobile and aeroplane parts.

The Westinghouse Research Laboratory became interested in this problem and work has extended over a period of several years. Experimenting with different solutions, addition agents, current densities, etc., there has been developed a standard method of procedure using commercial salts and sufficiently high current density so that all ordinary repairs will not require more than two or three hours in the plating bath. The cost of building up and machining are thus kept low enough so that it will generally pay to reclaim a piece rather than use a new one.

For general repair work a plating bath made up in the following proportions is satisfactory: Commercial ferrous ammonium sulphate, or the proportional amount of ferrous sulphate and ammonium sulphate, 2.5 pounds per gallon of water plus five ounces of ferrous carbonate freshly precipitated and kept under water, plus a handful of powdered charcoal. The carbonate "mud" keeps the solution practically neutral. The charcoal prevents pitting at the cathode surface. The sulphate salts must be free from sulphocyanates.

All small pieces are plated in earthenware crocks and large pieces in lead-lined wood tanks or stoneware tanks. Difficulty is experienced in using the plating solution in a wood tank. Organic materials from the wood enter the plating solution and cause excessive pitting of the deposit. The anodes are made of Armco iron, cylindrical in shape. Micarta disks with a hole cut in the center are fitted in each end of the anodes. The pieces to be plated are made the cathode and held stationary as they extend into the plating solution through the holes in the micarta disks. The anodes are attached to a suitable device designed to move them up and down, thus keeping the plating solution agitated and the ferrous carbonate and charcoal in suspension. The temperature of the bath is held at approximately 60 deg. cent. A current density of 60 to 65 amperes per sq. ft. is used. This deposits metal at the rate of 0.006 in. in diameter per hour. This method of procedure applies to pieces such as shafts, plug gages, reamers, pins, etc. For pieces such as gear centers, where the inside diameter is to be reduced, the anode passes up and down through the center and micarta rings are attached to it to produce stirring.

Probably the most essential part of the process and the step which requires the most careful attention is the cleaning of the piece before plating. After trying different methods of cleaning to obtain good adherence of the deposited metal, the electrolytic process was adopted as follows: The piece to be plated is made the cathode in an alkaline solution consisting of 5–10 per cent of caustic soda and 5–10 per cent of commercial sodium carbonate, using a current density of 3 amperes per sq. in., for a period of three minutes. It is then washed in running water and transferred to a 30 per cent commercial sulphuric acid solution and given a high enough current density to make it passive. This treatment is continued for a period of three minutes or longer, according to the condition of the piece. If it is free of rust in starting, then three minutes is sufficient. The piece being cleaned should come out of the acid-cleaning solution bright and silvery except in the case of cast iron which will be black due to graphite. When taken from the acid-cleaning solution the piece is washed in running water and immediately transferred to the plating bath without the cleaned surface being touched. It is very essential that the current density be held high enough, at least 3 amperes per sq. in. in the acid solution, so that the piece being cleaned remains passive. When passive there is a rapid evolution of gas with the piece remaining bright, but if allowed to become active it has a dull black appearance. When a piece is coated with oil or grease it is cleaned with gasoline or benzene prior to cleaning electrolytically.

Portions of the work which are not to be plated, but which will come in contact with the solution, are brushed over with melted hydroline of a melting point sufficiently high to stand immersion at 60 deg. cent. without undue softening. After the piece is plated the protective coat can be readily removed by washing with benzene.

It is impossible to build up a piece accurately to size; therefore it is necessary to plate a few thousandths over and then grind to finish size. This method of recovery can be used with steel, cast-iron or case-hardened material. Press fit, severe bending, and forging tests with deposits $^1/_{16}$ in. or more thick, all indicate that platings produced in this manner are satisfactorily anchored to the base metal. Plug and thread gages have been repaired and have given satisfaction, although they are not as hard as heat-treated tool steel. It is, of course, possible to carburize such pieces, when they will compare favorably with the original gages. One of our prize exhibits is the shaft from a 3-hp. motor which was used on a six-spindle drill press with a rather short, very tight belt. This shaft was ground undersize to simulate wear, then plated oversize, ground, and reassembled. It ran steadily

---

[1] Research Laboratories, Westinghouse Electric & Manufacturing Co.

[2] Thomas, *Automotive Industries*, August 26, 1920.

[3] Macfayden, *Trans. Far. Soc.*, 1920, vol. xv, 98.

Presented at the Fourth National Machine Shop Practice Meeting, Chicago, Ill., September 22–24, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

on the day shift for eight months and was then taken out for inspection. No wear could be detected. As this shaft had had more than a thousand hours of actual service, it is quite evident that this material is all that can be desired for bearing service.

The limiting factor in the size of a plating plant is the amount of current needed for cleaning. At the rate of 3 amperes per sq. in., a 1000-ampere generator set would clean a piece having 333 sq. in. surface, the equivalent of a shaft 4 in. in diameter and 26 in. of plating length, or a shaft 6 in. in diameter and 17 in. of plating length. After a repair shop of this kind has been equipped the chief operating expense is labor. The plating baths should run 6 months at the least without any attention or addition of salts with the exception of adding water to replace that lost by evaporation and the addition of a small amount of ferrous carbonate and charcoal once a week. The cost of the current used in cleaning and plating is small. The anodes are gradually used up and have to be replaced by new ones; however, the initial cost of an anode is distributed over a number of repair jobs, making the amount added to an individual piece small. The laborer is chiefly concerned in getting pieces ready to be plated and cleaning the ones that have already been plated. As they require no attention while in the plating bath one man can handle enough pieces in a day to make such a shop a paying investment.

Discussion

H. A. Rohrich.[4] Have worn cast-iron parts been successfully salvaged by this process? There are a large number of slides and other parts in automatic bottle machines which wear rapidly and are now thrown away. Has there been enough experience with cast-iron parts salvaged in this way to show that they wear sufficiently well to justify the cost of salvaging as against replacement?

Our bottle machines are complicated and have a great many cast-iron parts, some of which pass through a series of machining operations. If local wear can be restored by electroplating, it will save considerable money, as the parts will last indefinitely otherwise. There must be many machines where local wear of expensive cast-iron pieces can be restored, and if the author can give additional information, it will be of great value to the industry. We expect to do some experimenting with Mr. Thomas' process and will report results.

[4] Mechanical Engineer, Materials Research, Owens-Illinois Glass Company, Alton, Ill. Mem. A.S.M.E.

MSP-53-5

Arc-Welded Jigs, Fixtures, and Machine Tools

By J. R. WEAVER,[1] EAST PITTSBURGH, PA.

Among the advantages of jigs and fixtures welded up from structural-steel shapes over those of cast construction, the author cites production on short notice, strength combined with comparative lightness in weight, ease of alteration, etc. He outlines the methods used in their manufacture, and gives examples of a wide variety of jigs and fixtures and of special machine tools embodying this form of construction.



THE application of welding to the building of jigs, fixtures, and machine tools, which the author proposes to discuss here, was practically forced upon the company with which he is connected about a year ago. Its business increased very rapidly about that time, so that in order to obtain all advantages possible in manufacturing, it was necessary to obtain jigs, fixtures, and special machine tools on very short notice.

It was quite noticeable in regard to new machine tools purchased from the machine-tool manufacturers that delay in receiving them was due in all cases to the time necessary to make the jigs and fixtures. Naturally, the machine tool was more or less of a standard product and could be delivered on comparatively short time.

When the idea was conceived of manufacturing jigs and fixtures from structural shapes welded together, several such jigs were built and put through a very thorough test to ascertain whether they would be suitable for the company's work. After testing them for all the inaccuracies that are liable to occur due to warpage, distortion, etc., a definite program of manufacture was then decided upon which has given extremely satisfactory results. The thought was advanced that the jigs and fixtures would not maintain their shape after a period of time, but they have been used for about a year and have required less maintenance than did similar cast-iron jigs.

As mentioned above, it became necessary to obtain jigs, fixtures, and machine tools on very short notice, and in checking deliveries of cast jigs versus structural jigs, it was found that when putting the orders through for cast jigs in the regular way, it required about four weeks to obtain castings. Of course, it was necessary to make patterns, core boxes, etc., and the castings had to be made, cleaned, etc., before being sent to the shop. In the case of welded structures, however, we can obtain the most complicated jig in not more than a week's time. The average time is about four days and it will readily be appreciated what a decided advantage this is in manufacture of any kind of product.

Construction of Welded Jigs

These jigs are made from hot-rolled steel of whatever shapes and sizes are necessary and welded together. Where hardened pieces are required, these are made from tool steel, hardened and attached to the jigs by means of screws, bolts, etc., or in the case of bushings, pressed in holes provided for them. Some trouble was experienced owing to the fact that after the jig had been annealed the feet were too soft and chips on machine tools would imbed themselves in them. In order to overcome this, the feet were made from tool steel and heat treated before being welded to the jig. They were then drawn to a scleroscope hardness of 45 to 50 and were of sufficient hardness but could still be machined. The results have been very satisfactory.

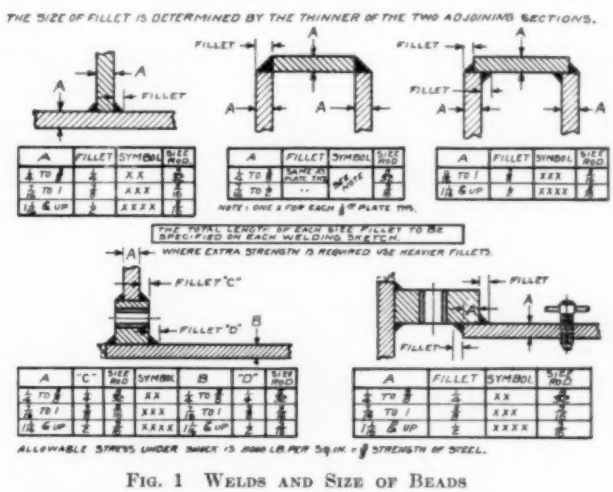


FIG. 1 WELDS AND SIZE OF BEADS

Of course, when welded jigs and fixtures were adopted, it was necessary to change the entire method of manufacturing jigs, fixtures, and machine tools. This not only included the actual manufacture of these jigs and fixtures in the shop but also their design. In order to have the proper information on the drawings and sketches which were sent to the shop, it was necessary to train the tool designers in the art of welding to a certain extent. To do this, the company incorporated in its standard book several sheets of instructions which, in addi-

[1] Superintendent, Manufacturing Equipment Department, Westinghouse Electric & Manufacturing Co. Assoc-Mem. A.S.M.E. The author was born in 1895 near Pittsburgh, Pa. Mr. Weaver spent all of his time since the age of 16, with the exception of the World War, when he served in the Navy as Ensign aboard the *U.S.S. Connecticut*, with Westinghouse, either engaging in the actual operations of manufacture or in their supervision. After serving his apprenticeship of four years at the Westinghouse Air Brake Company, he remained two years as master machinist. In 1919, he was made assistant superintendent of tools and later had charge of moving all tool equipment from East Pittsburgh to the South Philadelphia Works. In 1921 he became associated with the Westinghouse Electric and Manufacturing Company, of East Pittsburgh, in the capacity of mechanical engineer. In 1926, he was promoted to superintendent of machine tool equipment, and in 1928 he was again promoted to superintendent of manufacturing equipment, which position he now holds.

Presented at a meeting of the Metropolitan Section of the A.S.M.E., New York, N. Y., January 7, 1930, and at a meeting of the Cincinnati Section, Cincinnati, Ohio, January 23, 1930.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

tion to supplying the workmen in the shop with information, also standardized the manufacture of welded jigs and fixtures.

Fig. 1 shows the various types of welds and the size of bead which is used on welded jigs and fixtures. This is a standard throughout the shop, so that the workman is familiar with it and any reference to it is readily understood.

In order to avoid delays due to material being out of stock, there is included in the standard book a sheet showing the standard stock sizes of steel that are being used for this particular method of manufacture.

Fig. 2 shows such a sheet and gives the sizes of bar and plate stock. When bar stock is not suitable, the part is cut to size from a plate by means of an acetylene torch.

When the designer makes his drawings of a jig, he lays it out without showing any particular method of construction. Fig. 3 shows a milling fixture, and it will be noted that no reference is made to welding in it. Therefore it is necessary for the tool designer to make a separate sketch, which is shown in Fig. 4 and is called a welding and machining sketch. This gives the detail construction and indicates what material is to be ordered, and from the drawing can be obtained the information as to the shape desired. This material is then cut from the proper-sized stock with a saw in the case of bars, or with an acetylene torch in the case of a plate. The parts are then assembled and welded together. The title on this sketch states the total length of a certain-size weld necessary to completely weld this jig. This is done for the convenience of the time-study supervisor, who estimates and sets a value on the time it should take to manufacture the jig.

Figs. 5 and 6 show more in detail the various parts of a jig and the method of assembling and machining, and give a better idea of the extent to which information is given to the shop. In Fig. 6 the various sections are shown separately so that the shop will have no trouble to obtain the material and cut it to the proper shape and size. Fig. 5 also is used for machining purposes, giving the necessary dimensions, etc. that are required to properly machine this jig.

Of course, it is not possible to test a weld without destroying it, and the workman must be relied upon for good workmanship. In order to control this phase of the welding, all welders are required to stamp their work with a stencil. This stencil identifies the particular welder who performed the work and, in case there is any question, it can be readily traced back to him.

Periodically, a test is run on welds, and if any welding is dis-

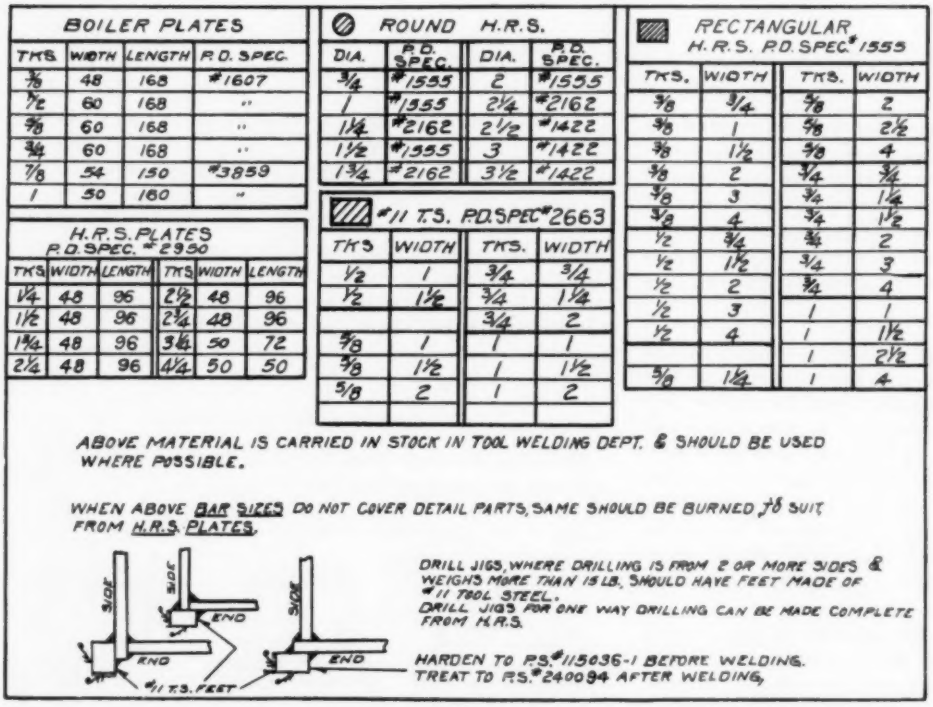


FIG. 2 MATERIAL TO BE USED IN WELDED JIGS AND FIXTURES

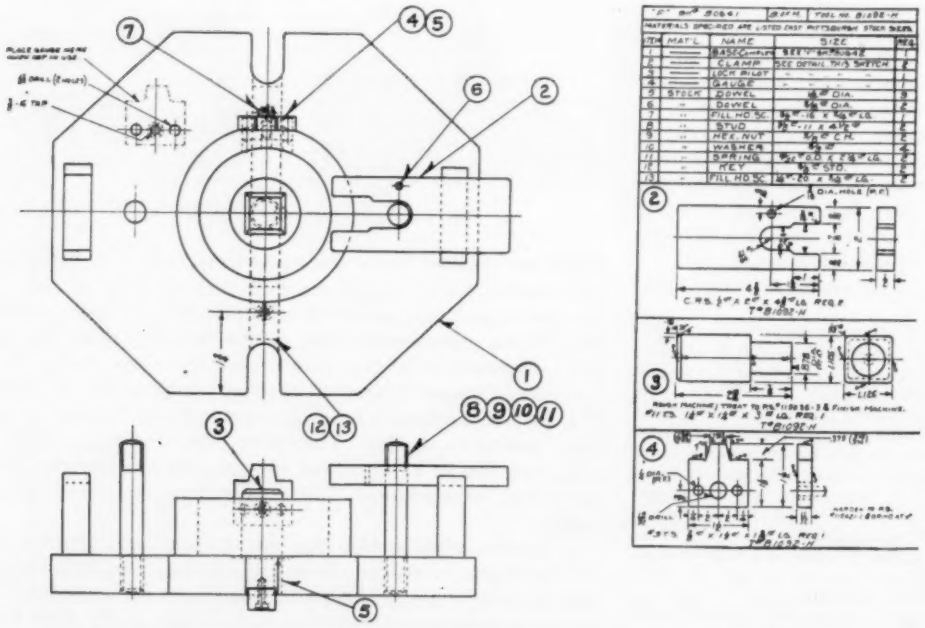


FIG. 3 MILLING FIXTURE FOR CLUTCH-TYPE OIL-IMMERSED CAM CONTROLLER

covered that is not satisfactory, the workman is required to take additional instruction; or, if he proves entirely unsatisfactory, he is taken off the job. The type of stencils used is shown in Fig. 7.

Fig. 8 shows a cutting machine which is used for cutting from plate stock the various parts that are required to make a jig. There are several different types of these machines in this department for cutting any shape or size necessary.

Fig. 9 shows a number of parts cut to size to make a jig. It will be noted that all the necessary large holes are cut in the plates by means of an acetylene torch before the parts are assembled into a jig.

These various detail parts are assembled and tack welded just sufficiently to hold them together (Fig. 10). This is done by a mechanic so that the parts will be properly located, also so that the jig will be squared up and be a good job when it is finally finished. After the tack welding the jig is passed to a welder, who completes the job by welding the jig as it should be.

Fig. 11 shows a jig completely welded. While this jig is fairly large, it weighs only 120 lb. It is used for drilling the bracket which is shown at the right of the jig.

Removing Strains From Jig Due to Welding

The fact that these jigs are made from hot-rolled steel, as well as the welding, detracts somewhat from the appearance of the completed jig. Certain strains due to the welding must be removed from the jig so that it will maintain its shape and size. This is done by heating the jigs in a furnace to a temperature of 1000 deg. fahr., after which they are allowed to cool in the air. Owing to the use of hot-rolled steel and this normalizing treatment, a certain amount of scale develops on these jigs, which is removed by means of a sand-blast machine as shown in Fig. 12. After the sand-blasting operation, which really improves the appearance of the jig, the jig is spray painted as shown in Fig. 13. It is sent to the shop for machining, which is no different from what it would be for a cast jig except that not as much material is allowed for finish as is ordinarily allowed on castings.

To give an idea of the extent to which welding has been applied in the manufacture of jigs and fixtures, Figs. 14 and 15 show a jig weighing 1300 lb., the overall dimensions being $39\frac{1}{2}$ in. $\times$ $51\frac{1}{2}$ in. $\times$ 21 in. This complicated jig is built entirely of steel, using hot-rolled bar stock and plates. The bosses shown in Fig. 14 are made from hot-rolled round steel cut off with a saw and welded in place.

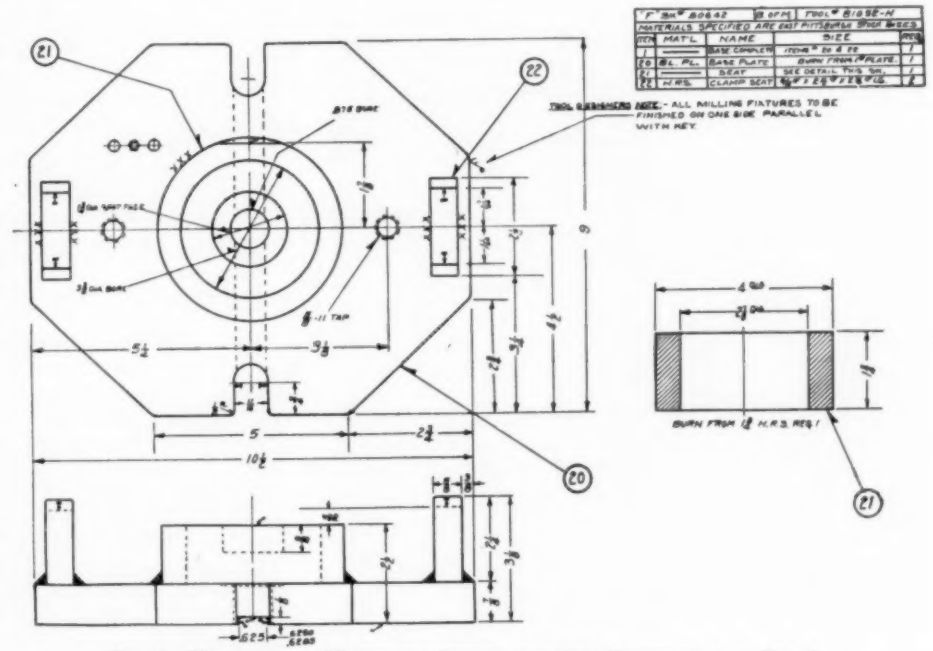


Fig. 4 Welding and Machining Sketch for Cam Controller of Fig. 3

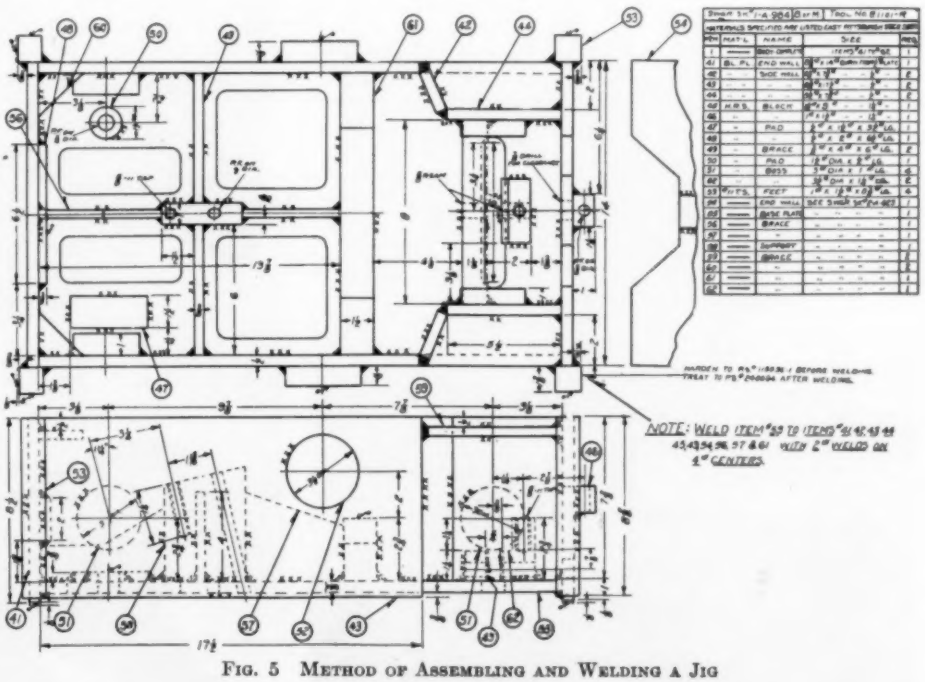


Fig. 5 Method of Assembling and Welding a Jig

Fig. 16 shows welded jig being used on a drill press. No difficulty has been experienced with these jigs due to inaccuracies, although they have been in use for approximately a year.

Figs. 17, 18, 19, and 20 show several other types of jigs that have been manufactured, and indicate that it is feasible to make any kind of a jig that is necessary by this method.

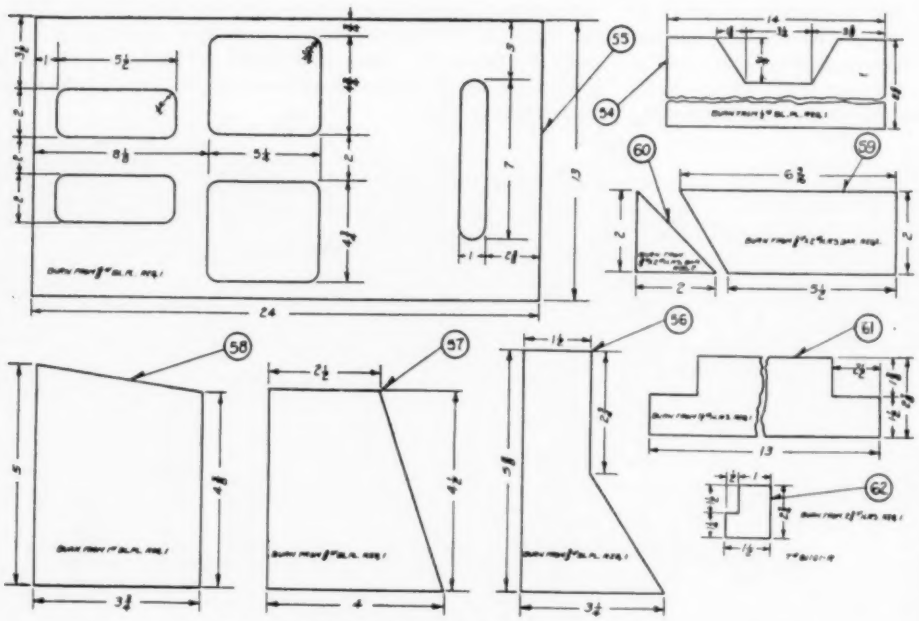


FIG. 6 SEPARATE PARTS OF JIG SHOWN IN FIG. 5

SEVERAL ADVANTAGES OF WELDED JIGS

In addition to other advantages set forth, jigs made from steel have considerable more strength and weigh less than cast jigs. This is because thinner material can be used and because it is not necessary to increase the sections to reduce casting difficulties, etc. It is also possible to place metal where it is most desired in order to make a stronger jig with the least amount of weight.

Also in the event that it should become necessary to make a change in the jig owing to the products being changed or for any other reason, it is very easy to remove any interference by means of an acetylene torch and reweld the parts in new locations if desired.

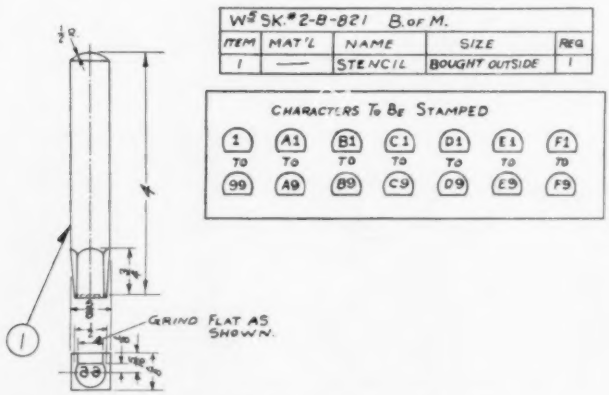


FIG. 7 WELDERS' IDENTIFICATION STAMP

MACHINE TOOLS OF WELDED CONSTRUCTION

Attention is called to another application and that is the design of special machine tools employing the same type of construction. The Westinghouse Company has applied this method of construction only to special machine tools such as it designs and builds itself, but the author believes it will be applied to all types of machines in the future. For example, the machine shown in Figs. 21 and 22 is a double-head oxygen-acetylene

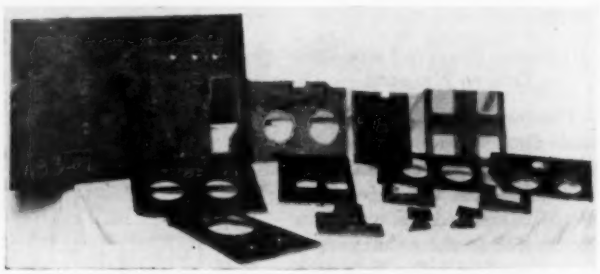


FIG. 9 VARIOUS DETAILS AS CUT WITH TORCH, SHOWING ALSO ONE SIDE MARKED OFF, AND DRILLED HOLES FOR STARTING THE CUT

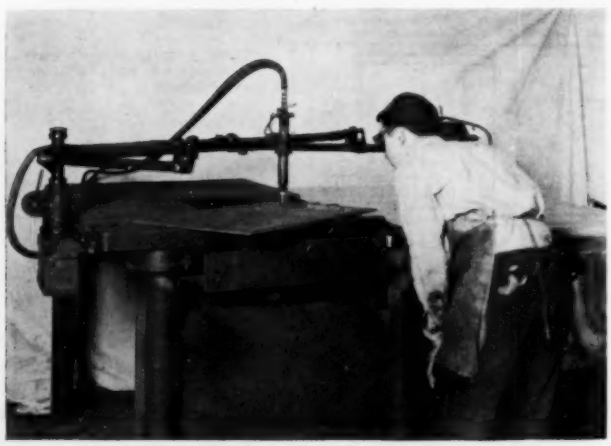


FIG. 8 CUTTING OUT THE PARTS WITH A TORCH

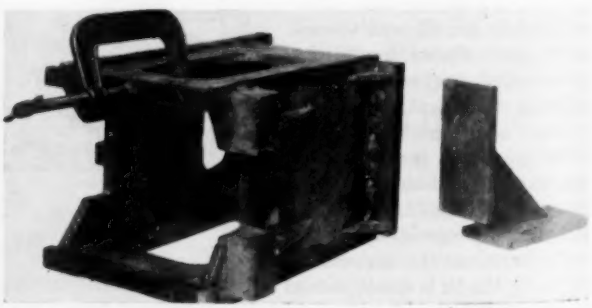


FIG. 10 SHOWING CLAMP HOLDING TOOL-STEEL FOOT IN PLACE FOR TACK WELDING. A SQUARING OR SETTING-UP ANGLE IS SHOWN AT RIGHT

[Feet made from tool steel S.A.E. Spec. 6150(0.4 5–0.55 carbon); hardened before welding and annealed by heating to 1000° F. after welding.]

cutting machine. It was necessary to design a machine that would cut either straight or bevel edges of various thicknesses of steel. The machine is so constructed that one man can operate both heads, as the controls are so arranged to be convenient

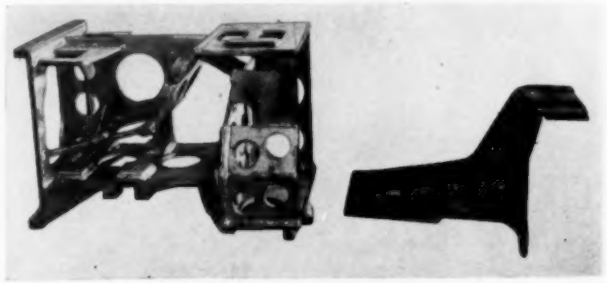


FIG. 11 BODY OF DRILL JIG FOR BRACKET FOR TALKING MOVIE MACHINE
(This jig is 15 in. × 23 in. × 12½ in. high and weighs only 120 lb. Bracket shown at right.)

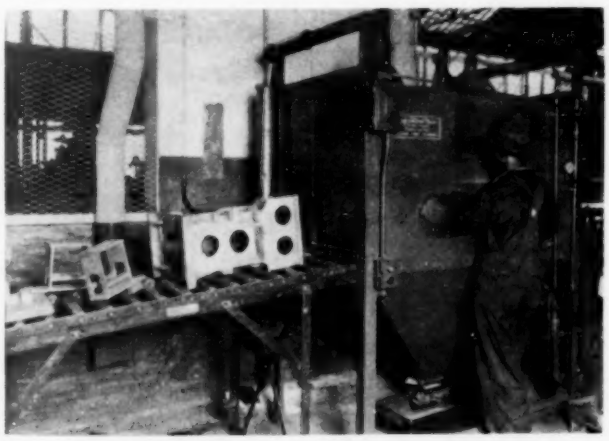


FIG. 12 REMOVING SCALE BY SHOT BLAST



FIG. 13 SPRAY-GUN PAINTING OF JIG WITH OIL-RESISTING PAINT TO MAKE SMOOTH SURFACES AND PREVENT CHIPS FROM STICKING TO JIG

from one position. This particular structure worked out very satisfactorily, and no troubles developed due to misalignment, vibration, or any of the things that may be found in the machines which are not properly designed or built.

FIG. 15 (LOWER RIGHT-HAND COLUMN) REAR VIEW OF DRILL JIG SHOWN IN FIG. 14 BREAKER TYPE "B-32"



FIG. 14 FRONT VIEW OF DRILL JIG FOR MAIN BASE OF OIL CIRCUIT BREAKER
(Weight, 1304 lb.; size, 39½ in. × 51½ in. × 21 in.)

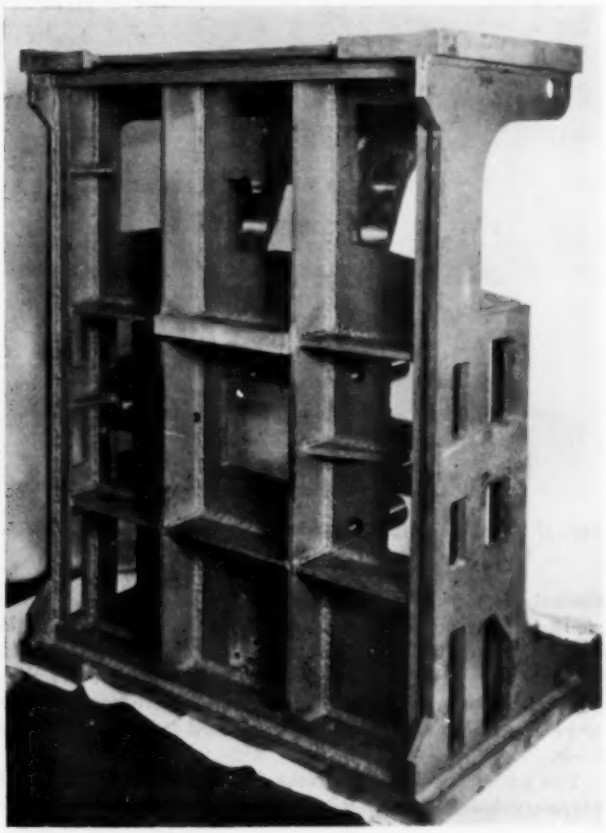


Fig. 23 shows a machine for testing mica sheets. With the exception of the worm-gear reduction unit, this is built entirely from structural steel and welded together.

Fig. 24 shows winding machines for winding mush-type coils for a.c. motors. These machines are entirely fabricated from structural steel and welded together, and attention is particularly called to the construction of the tailstock. This is similar to the general design of tailstocks, with the exception that this is welded from structural steel. It functions satisfactorily and, although the requirements here are not as rigid as they would be on metal-cutting machines, there does not seem to be any reason why it could not be used successfully on such tools.

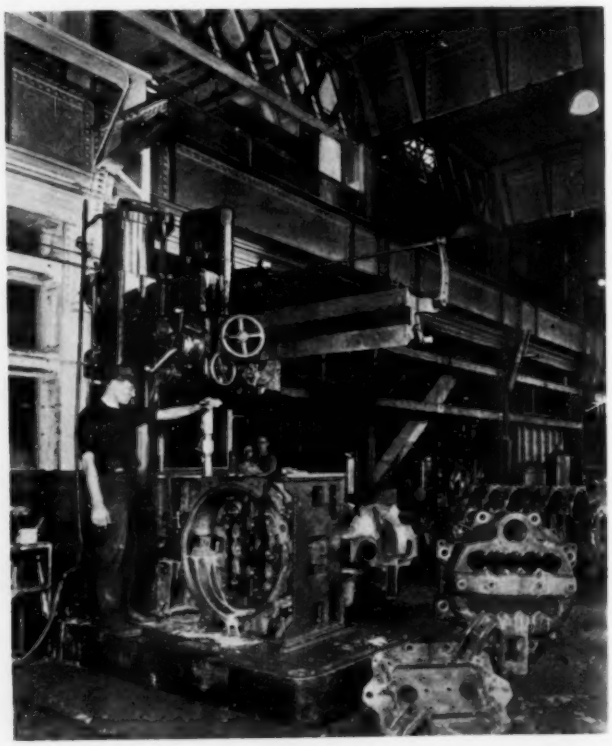


FIG. 16 DRILL JIG OF FIGS. 14 AND 15 IN USE

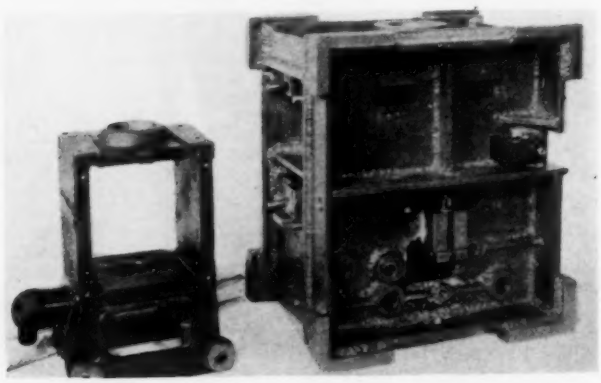


FIG. 17 SOLENOID MECHANISM FRAME AND 7-DRILL JIG FOR SAME
(Heavy boring is done in this jig.)

The author realizes that some further development will be necessary before this particular type of construction can be applied to general machine-tool construction, but he believes the facts he has brought out successfully demonstrate the practicability of welded jigs, fixtures, and machine tools. He is building from 200 to 400 welded jigs and from 3 to 6 special machines per month, and has been able to build them in a shorter time, to reduce the average cost approximately 25 per cent, and to produce jigs and fixtures that are more desirable due to the fact that they are lighter and at the same time of at least equal strength. The jigs and fixtures which have been built in this way have required less maintenance, and it is not believed that anything has been sacrificed that would result in misalign-

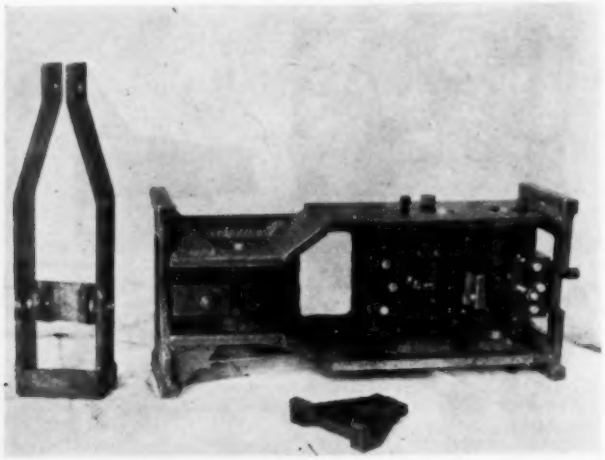


FIG. 18 FABRICATED LEVER AND DRILL JIG FOR SAME



FIG. 20 REAR VIEW OF FIXTURE SHOWN IN FIG. 19
(16½ in. sq. base × 32 in. high.)

FIG. 19 FRONT VIEW OF DRILLING AND LINE-REAMING FIXTURE FOR BEARINGS ON CROSS BAR FOR LARGE OIL CIRCUIT BREAKER

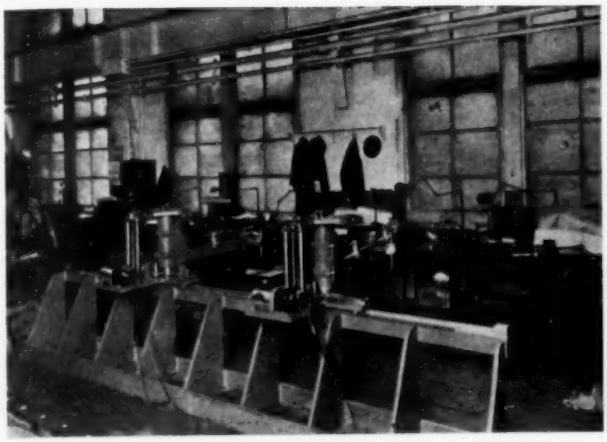


Fig. 21

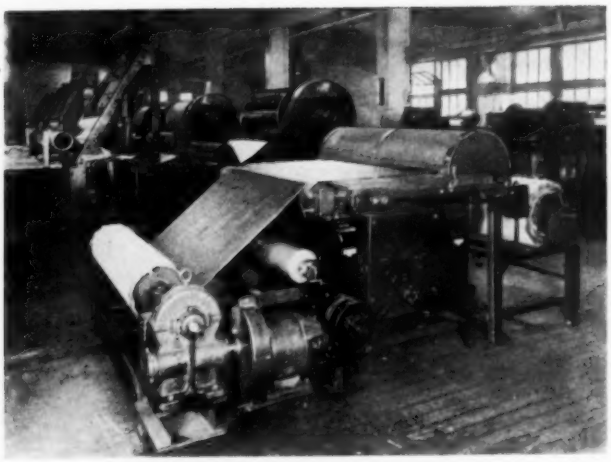
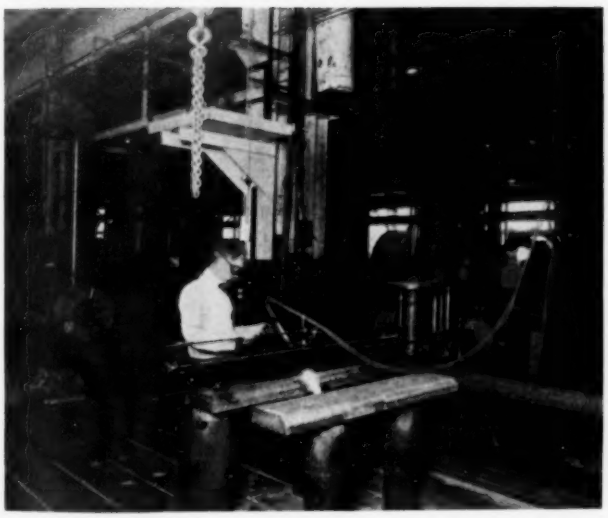


FIG. 23 MACHINE FOR TESTING MICA SHEETS



FIGS. 21 AND 22 DOUBLE-HEAD OXYGEN-ACETYLENE CUTTING MACHINE



FIG. 24 WINDING MACHINES FOR WINDING MUSH-TYPE COILS FOR A-C. MOTORS

ment, warpage, distortion, vibration, etc., or detract from the appearance on this particular type of equipment. It is believed to be a practical idea and, owing to the advantages which have been set forth, it should be given serious consideration.

MSP-53-6

Machining Properties of Some Cold-Drawn Steels

By O. W. BOSTON,[1] ANN ARBOR, MICH.

This paper gives the results of a series of machining tests on 13 cold-drawn screw-stock steels and 2 cold-drawn stainless steels. Machinability values for standardized conditions are given for each steel as follows: The force required to remove a chip by a planer tool; the energy required to remove a chip by a single-tooth milling cutter; the torque and thrust developed by a 3/4 in. diameter drill; the penetration of a 1/4 in. diameter drill under a constant feeding load. An attempt is made to correlate the results of these various machining tests with the Brinell, Rockwell, and scleroscope hardness values, as well as the analysis and structure of each bar. Values representing the power required to cut 1 cu. in. of metal per minute under specific standardized conditions are given for planing, milling, and drilling the various steels.



THE object of this paper is to show the results of various metal-cutting tests on a variety of cold-drawn steels so as to indicate, if possible, the best method for determining whether the material is suitable for machining purposes and the relative advantages of one material over another from the standpoint of machining.

In a paper before the American Society for Steel Treating,[2] the author gave an outline of the various test methods which were used in determining machining properties and comparative values between 18 ferrous metals and 21 non-ferrous metals. This paper is a continuation of the tests made on cold-drawn steels.

The Tests

Each specimen was subjected to the following tests:

(1) The Brinell, Rockwell, and Shore scleroscope hardness readings were determined on a ground surface approximately 1/16 in. below the cylindrical surface, as follows:

In the Brinell test a 3000-kg. load and a 10-mm. ball were used. The readings were taken with the grain; that is, across the diameter of the impression lengthwise of the bar. The time of duration of the load was 30 sec., and measurements of the impressions on the ground surface were taken with a Brinell microscope.

The Rockwell reading was read on the B scale, using a 1/16 in. diameter ball and the 100-kg. load.

The scleroscope readings were obtained with a Shore scleroscope.

(2) The force developed in the direction of cut by a planer tool was measured. The tool was of the end-cutting type, 0.500 in. wide and provided with a 15-deg. front-rake, but no side-rake angle. The corners of the tool were beveled off, making a 1/64-in. flat at a 45-deg. angle, to insure greater endurance of the geometric shape of the high-speed-steel tool. After grinding to shape and size, the tool was honed. A depth of cut of 0.010 in. as measured on a dial gage was taken in all cases. An average was taken of from 6 to 12 consecutive readings, when cutting dry at a speed of 20 ft. per min.

(3) The torque and thrust developed by a 3/4 in. diameter high-speed-steel twist drill was determined. The drill had a helix angle of 30 deg. and a clearance angle of 3 deg. and 48 min. as measured at its periphery. The web thickness at the point was 0.095 in. and the outside diameter was 0.750 in. The drill was ground on a drill grinder with a drill overhang from the jaws of 3/8 in. The grinder forms the drill point so that it has a conical surface back of the cutting edge. The drill was measured on a special measuring instrument to make sure that the cutting edges were of the same length and position to cut chips of equal thickness. This drill was rotated at 153 r.p.m. and was given a feed of 0.012 in. per revolution.

(4) The drill penetration or the distance traveled by a 1/4 in. diameter twist drill for 100 revolutions when rotating at 504 r.p.m. under a dead load, including the weight of the drill and drill spindle of 93.75 lb., was determined. The twist drill had an outside diameter of 0.250 in., a web thickness at the point of 0.0269 in., and a helix angle of 24 deg. The clearance angle was about 6 deg. and 30 min. at the periphery of the drill. This drill was ground on a drill grinder which formed a cylindrical surface back of the cutting edge.

(5) The energy absorbed by a single-tooth milling cutter of the side-cutting type—that is, similar to a cutting-off tool—was determined. The tool had a width of 0.250 in., a 15-deg. front-rake, but no side-rake angle, and a 4-deg. clearance angle on the end and sides. The edges of the cutter were beveled to 1/64-in. flat at an angle of 45 deg., and the tool point finally was honed all over. The cut taken had a radius of 1 3/4 in., a depth of 0.125 in., and a feed of 0.010 in. per chip.

(6) The influence as shown by photographs of the cut surface as made by a shaper tool when having various front- and side-rake angles and when operating at light and heavy depths of cut and feeds, as well as two different speeds, are described below in connection with the tests.

It is realized that the influence of the material on the "tool life" under standardized conditions is of real commercial value. Such tests were not included with those described, as there was

---

[1] Director, Department of Engineering Shops, Colleges of Engineering and Architecture, University of Michigan. Mem. A.S.M.E. Mr. Boston was graduated from the University of Michigan Engineering College in 1913, received a Master's degree in 1917, and the degree of Mechanical Engineer in 1926. After graduation he was engaged at the University of Michigan as instructor in engineering mechanics and mechanical engineering for four years. In October, 1917, he was commissioned in the U. S. Navy and assigned to duty in the Bureau of Ordnance on design and manufacture of submarine mines used in the North Sea blockade. From 1919 to 1921 he was engaged in industrial engineering work for the Cleveland Tractor Company, in Cleveland, as assistant to the vice-president and works manager. In the fall of 1921 he returned to the University of Michigan.

[2] O. W. Boston, "Methods of Tests for Determining the Machinability of Metals in General, With Results," Trans. A.S.S.T., vol. 16, Nov., 1929, pp. 659 to 710.

Presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

insufficient material of each type on hand to make it possible to obtain any information of value.

It is further realized that, when various metals are compared, the metals themselves should be the only variables. In commercial practice a tool usually is ground to such a shape that satisfactory tool life is obtained in conjunction with finish and size. If the tool shape is changed, the tool life will be changed and also the finish on the cut surface will be different. The different degrees of roughness will in themselves alter the size of the finished piece or the size may be varied by changing tool shape, which affects pressures on the tool and thereby causes it to spring more under one condition than under another. No attempt is made in this paper to present the optimum tool form for the materials studied in order that maximum tool life, least power, and best finish may be obtained simultaneously.

The Testing Machines

The testing machines used in conducting the tests for machinability, as outlined, may be described briefly as follows: A specially designed dynamometer for measuring the force on a planer tool in the direction of cut was mounted on the 30-in.

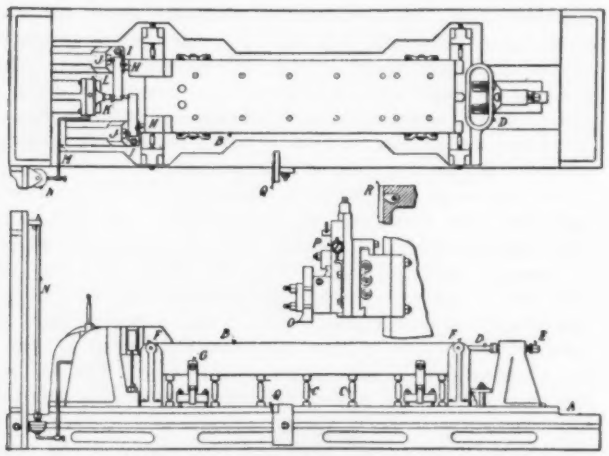


Fig. 1 Line Diagram of Planer Dynamometer

by 36-in. by 8-ft. planer. Power was furnished to the planer from a main drive shaft, a satisfactory arrangement in that the horsepower output of the machine was measured rather than the input. The cutting speed for all of these tests was 20 f.p.m. No cutting fluid was used.

Fig. 1 shows a line diagram of the dynamometer which consists essentially of a cast-iron bed *B*, weighing about 1300 lb., mounted on seven 14 in. long horizontal knife edges *C*. The bed is prevented from moving sideways when a side-rake angle on the tool is used by the four horizontal floating pins shown in the plan view, supported by the brackets *F*. The bed is kept from being lifted by four knife edges shown at *G*. An initial load of 2000 lb., tending to force the bed to the left was put on the bed by compressing the loop spring *D*. This load keeps the whole mechanism in compression, thereby eliminating chatter during a cut, and also keeps the mercury-column gage under an initial load which is indicated as a zero point on the scale. The loop spring was made of $^3/_4$ in. square spring steel having major and minor axes to center of bar of $11^1/_2$ in. and $4^3/_4$ in., respectively. It had previously been calibrated on a Universal testing machine. This spring also served for calibrating the mercury-column gage *N*. For more detailed description of this equipment, see Trans. A.S.M.E., 1926, Vol. 48, p. 749. The depth of each cut in these tests was secured by lowering the planer head and tool vertically 0.010 in. as measured on the dial indicator *P* mounted on the right-hand side of the head. There were 6 to 12 consecutive cuts taken on each test piece, and an average was determined. This eliminated even very slight deviation from the 0.010-in. feed per cut which might occur for a single chip.

The torque and thrust developed by the $^3/_4$-in. drill were determined by means of a torque and thrust dynamometer mounted on a manufacturing-type drill press. For detailed description, see Trans. A.S.M.E., Paper No. MSP-52-2. The press was driven by a 5-hp., 220-volt, 3-phase, 60-cycle motor through a short belt. The motor was mounted directly on the press column. All tests were run without a coolant or lubricant on the drill.

The thrust on the drill was measured on a recording chart of a gage. This pressure was exerted directly on a steel plate bearing on a rubber diaphragm just above the dynamometer base. Water underneath this rubber diaphragm was compressed and the pressure was transmitted to the gage thrust pen through a $^1/_8$-in. pipe. The plate resting on the rubber diaphragm which carried the thrust load was made to form the lower race of a large ball bearing consisting of 64 $^1/_2$-in. balls, so that the table of the dynamometer was free to rotate except for an extending arm which had a bearing against a piston resting on a second rubber diaphragm. The horizontal distance from the center of the drill to the support on the end of this arm was just 1 ft. The pressure on this torque rubber diaphragm compressed water back of it. This pressure was transmitted through a $^1/_8$-in. pipe leading both to a mercury column and the gage. The mercury column calibrated against the gage gave a scale reading for check purposes somewhat larger than that obtained on the gage.

To register the travel of the spindle the chart of the recording gage was made to rotate by means of a fine wire which passed from the upper end of the drill-press spindle over pulleys to a position vertically above the recording gage, thence downward making one turn about a special pulley on the shaft of the gage to a counterweight located below.

For determining the drill penetration under a given load, a sensitive drill press was remodeled so as to carry on the upper end of the rotating spindle a weight of sufficient amount to cause the tool to dig in when the spindle was rotated. By trial and error, the weight of 93.75 lb., including the drill spindle and drill, was arrived at in order to cause a $^1/_4$-in. twist drill to cut freely in various metals. A lighter load on the spindle or a larger size drill did not seem to give satisfactory results, as the drill could not be made to bite on some of the harder materials. This drill press also was driven through a short belt by a 5-hp. motor, a tight and loose pulley being provided. After mounting the specimen to be tested on the drill-press table or in a vise, the drill was first allowed to penetrate about $^1/_8$-in. By means of a beam-type indicator attached to a vernier height gage, the height of a particular point on the spindle was measured. With a speed counter attached to the spindle, the belt next was shifted from the loose to the tight pulley, causing the spindle to rotate. After 100 revolutions plus or minus, the spindle was stopped, and the height of the point on the spindle was again measured. The difference in measurements gave the penetration of the drill for the actual number of revolutions shown on the speed indicator. This was then reduced to 100 r.p.m.

This method of measuring the drill penetration was rather tedious and required considerable time; so in some subsequent tests the following method was used: Two beam-type indicators were mounted on a vernier height gage so that the vertical distance between the two, as measured with Johansson blocks, was just 0.250 in. The slide on the vernier scale was

so adjusted that, after the drill had penetrated about 1/8 in. the upper beam indicator bearing on the end of the drill sleeve would indicate zero, at which time a stop-watch would be started. After drilling continuously until the second indicator showed zero, the stop-watch was stopped. This gave the lapsed time for the drill to penetrate the 1/4 in. These results checked remarkably well with those obtained by the first method outlined. Averages of four to eight readings were taken for each test specimen.

All of the drill-penetration tests related in this paper were obtained with the same drill and with one grinding, as it was found almost impossible to regrind the drill, even on a machine, to give results comparable with those of the first grinding. After all tests were completed, many duplicate tests were run to check the influence of the dulling of the drill. It was concluded, however, that the dulling from all of the tests was insufficient to affect materially the results.

The energy absorbed by the single-tooth milling cutter, side-cutting (end-cutting) type, was measured on the machinability tester. This was constructed by Carl Oxford[1] and is described in detail in the Trans. A.S.M.E., 1921, Vol. 43, p. 549. If the pendulum be allowed to fall freely from the top vertical position it will rise almost to the same height. A freely rotating pointer is carried forward with the pendulum and indicates on a steel disk the highest position to which the weight rises. This position is taken as the zero reading to allow for friction of the bearings. If the material is bolted to the bed and adjusted for depth of cut and feed, a subsequent fall of the pendulum will form a chip, and the pendulum will not swing as high after taking the chip as when taking no chip. This change in height is a measure of the energy in foot-pounds given up in forming the chip. It is the same principle as is used in impact testing machines. The face of the steel disk is graduated at the periphery, so that the reading of the energy absorbed by taking a chip of any given material is read directly. Ten readings for each test specimen were averaged for the final value.

[1] Chief Engineer, National Twist Drill and Tool Company, Detroit, Mich.

The Materials Tested

The materials tested consisted of a variety of cold-drawn steels for use primarily in automatic screw machines. The bar numbers, together with the sizes and chemical analyses, are listed in Table 1. Table 2 shows the Brinell, Rockwell, and Shore scleroscope hardness values of the bars tested.

The exact analysis of bar 414 is as given. The tensile strength of this particular specimen was 86,100 lb. per sq. in. This material is purchased in accordance with the following specifications:

"Cold-drawn super free-cutting steel for screw-machine work. Carbon not over 0.18 per cent, with 0.10 to 0.18 per cent preferred. Manganese not over 0.90 per cent, with 0.60 to 0.90 per cent preferred. Sulphur not under 0.17 per cent, with over 0.18 per cent preferred. Phosphorus not over 0.13 per cent, with 0.07 to 0.09 per cent preferred. Tensile strength to be not less than 95,000 lb. per sq. in. for sizes 3/8 in. and under; to be not less than 90,000 lb. per sq. in. for sizes 3/8 to 5/8 in.; to be not less than 85,000 lb. per sq. in. for sizes 5/8 to 1 in. The material must be sound with surface suitable for screw-machine manufacturing operations."

Bar 415 also has the exact chemical analysis given, but has additional physical properties as follows: Proportional limit 65,600 lb. per sq. in.; tensile strength 92,050 lb. per sq. in., elongation in 2 in. 11.5 per cent, and reduction of area 44.8 per cent. This steel was found by a large user to stand machining at 100 per cent faster speed than the usual commercial stock. An attempt was made to have steel made to this analysis, but its machining properties were greatly inferior to the imported steel.

The carbon content of bars 422 and 424 appeared, from the photomicrographs, to be questionable, so that carbon analyses of these two bars were determined and found to be 0.336 per cent for bar 422 and 0.365 for bar 424, which conforms with the carbon range specified in Table 1.

Photomicrographs at 100 magnifications of steel bars Nos. 414 to 426, inclusive, are shown in Figs. 2 to 16, inclusive, except that Figs. 11 and 15 are of bars 422 and 424, respectively, at 1000 magnifications. This higher magnification was used in these two instances better to show the structure involved, which was not clear at 100 magnifications. All of these specimens were etched in a 4 per cent nital solution. All photomicrographs were taken on a surface approximately 3/16 in. from the outside, running parallel to the length of the bar. These photomicrographs represent fairly well the structure of the metal from

TABLE 1 COLD-DRAWN STEEL BARS TESTED FOR MACHINING PROPERTIES, WITH CHEMICAL ANALYSES

Bar No.	Material	Chemical analysis, per cent: Carbon	Manganese	Phosphorus	Sulphur[a]	Chrome or nickel	Vanadium
414	1 1/16-in. round super-free cutting screw-stock steel	0.10	0.66	0.098	0.195	.....	..
415	1-in. round free-cutting screw-stock steel (imported)	0.05	0.48	0.066	0.192	.....	..
416	1 1/4-in. square SAE 1112 cold-drawn, 1/16-in. draft	0.09–0.13	0.70–0.90	0.08–0.11	0.085–0.12	.....	..
417	3/4-in. square SAE 1120 cold-drawn, 1/16-in. draft	0.18–0.22	0.60–0.80	0.050	0.075–0.12	.....	..
418	1 1/2-in. square SAE 1120 cold-drawn, 1/16-in. draft	0.18–0.22	0.60–0.80	0.050	0.075–0.12	.....	..
419	3/4-in. square SAE 1112 cold-drawn, 1/16-in. draft	0.09–0.13	0.70–0.90	0.08–0.11	0.085–0.12	.....	..
420	1 1/8 round manganese screw-stock cold-drawn, 1/16-in. draft (1)	0.10–0.20	1.00–1.30	0.05	0.09–0.13	.....	..
421	1-in. round SAE 1315 cold-drawn, 1/16-in. draft (2)	0.10–0.20	1.25–1.65	0.05	0.10	.....	..
422	1 3/16-in. round SAE 6140 cold-drawn, 1/16-in. draft	0.35–0.45[b]	0.50–0.80	0.04	0.04	0.80–1.04	0.15
423	1 1/8-in. round SAE 2320 cold-drawn, 1/16-in. draft	0.15–0.25	0.50–0.80	0.04	0.045	3.25–3.75[e]	..
424	1 1/4-in. round manganese-carbon alloy cold-drawn, 1/16-in. draft	0.30–0.40[c]	1.60–1.90	0.04	0.045	.....	..
425	1-in. round special manganese-carbon alloy cold-drawn, 1/16-in. draft (4)	0.25–0.40	1.25–1.65	0.05	0.10	.....	..
426	1 1/2-in. round SAE 1320 cold-drawn, 1/16-in. draft (3)	0.15–0.25	1.25–1.65	0.05	0.10	.....	..

[a] The sulphur on all the high-sulphur steels may run above the limits indicated.
[b] Actual carbon content, 0.336 per cent.
[c] Actual carbon content, 0.365 per cent.
[d] Chrome.
[e] Nickel.

TABLE 2 HARDNESS VALUES OF BARS TESTED

Bar No.	Hardness number: Brinell	Rockwell B	Scleroscope
414	207	93	29
415	180	93	32
416	196	94.5	30
417	156	90	26
418	163	85	28
419	207	98	34
420	170	92	29
421	179	92	28
422	228	99.5	35
423	205	96	33
424	255	103	42
425	199	96.5	33
426	168	92	31

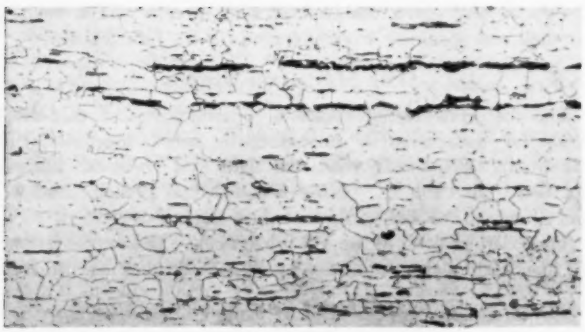


FIG. 2 BAR 414, 1 1/16-IN. ROUND SUPER-FREE CUTTING SCREW-STOCK STEEL × 100
(Shows low carbon; banded inclusions; moderate grain size.)

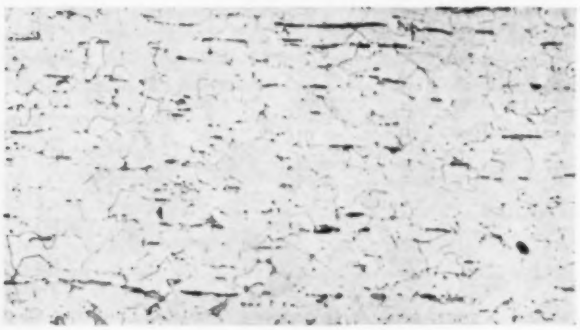


FIG. 3 BAR 415, 1-IN. ROUND IMPORTED FREE-CUTTING SCREW-STOCK STEEL × 100
(Medium grain size; slight banded structure.)

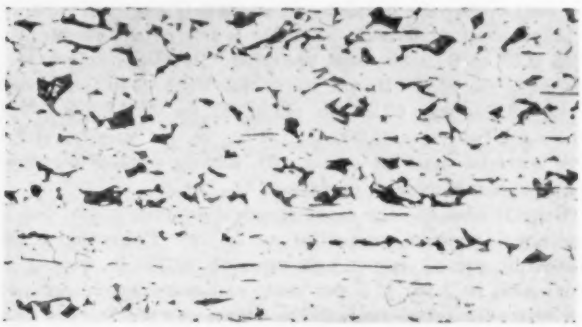


FIG. 4 BAR 416, 1 1/4-IN. SQUARE SAE 1112 × 100
(Medium grain size; shows considerable banded structure.)

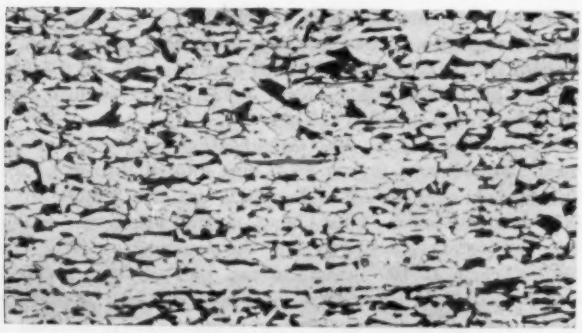


FIG. 5 BAR 417, 3/4-IN. SQUARE SAE 1120 × 100
(Small grain structure; pronounced cold-drawing structure.)

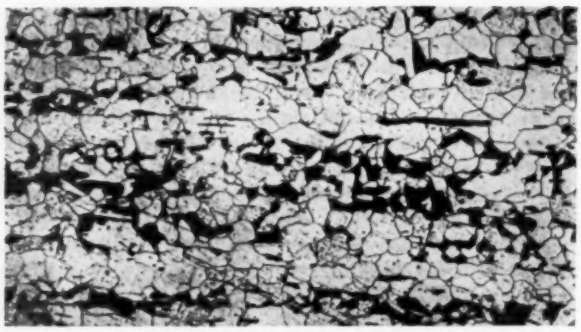


FIG. 6 BAR 418, 1 1/2-IN. SQUARE SAE 1120 × 100
(Medium grain size; banded structure.)

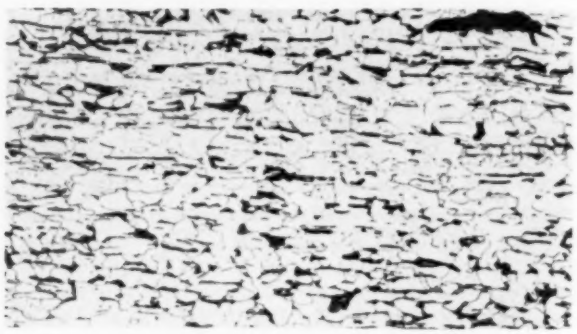


FIG. 7 BAR 419, 3/4-IN. SQUARE SAE 1112 × 100
(Small grain size; banded structure; pronounced cold-drawing structure.)

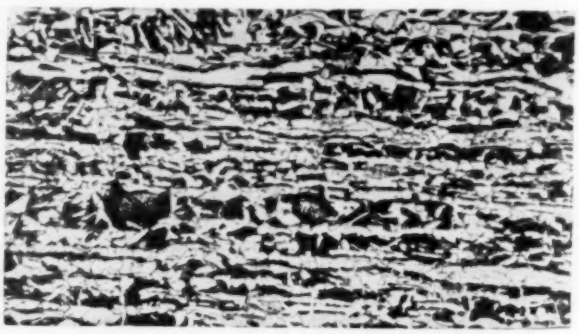


FIG. 8 BAR 420, 1 1/8-IN. ROUND MANGANESE SCREW-STOCK STEEL × 100
(Medium grain size, with large carbide areas.)

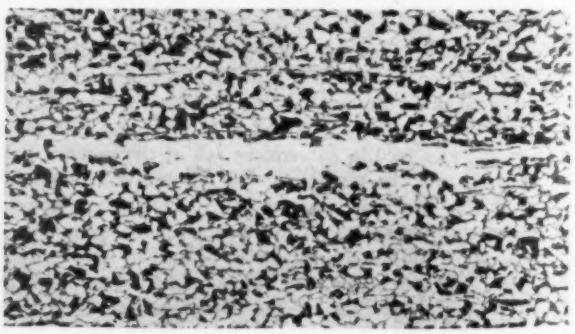


FIG. 9 BAR 421, 1-IN. ROUND SAE 1315 × 100
(Small grain size.)

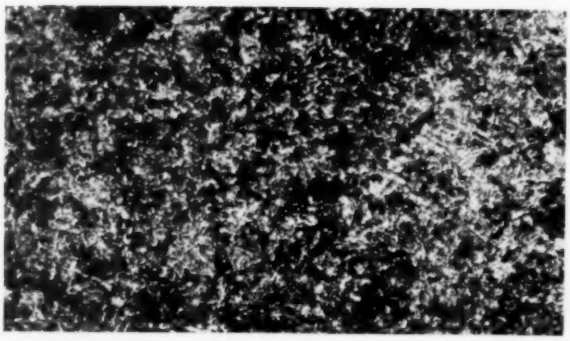


FIG. 10 BAR 422, 1 3/16-IN. ROUND SAE 6140 × 100
(Appears not fully annealed before cold drawing, little banding.)

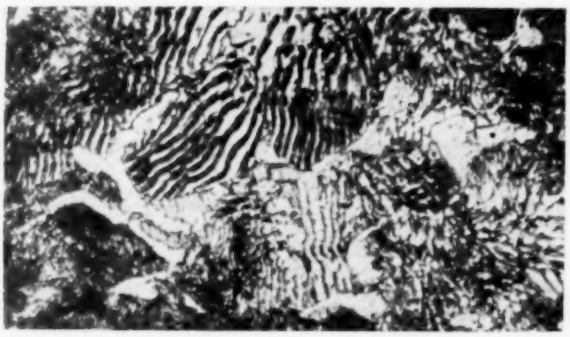


FIG. 11 BAR 422, 1 3/16-IN. ROUND SAE 6140 × 1000
(Shows laminated pearlite.)

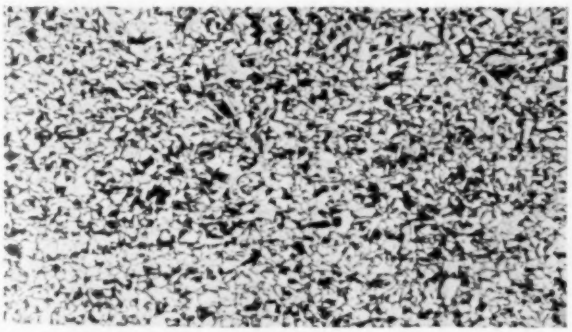


FIG. 12 BAR 423, 1 1/8-IN. ROUND SAE 2320 × 100
(Fine grain sizes well refined, remarkably free from banding for nickel steel.)

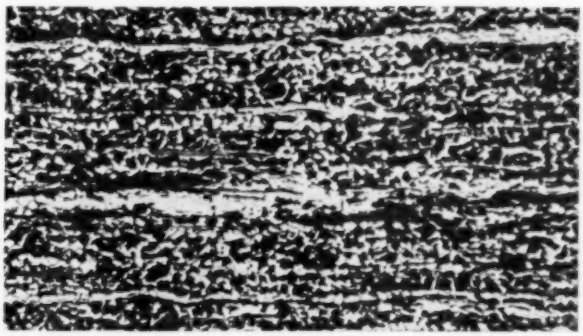


FIG. 13 BAR 425, 1-IN. SPECIAL MANGANESE CARBON STEEL × 100
(Fine-grain size; banded structure.)

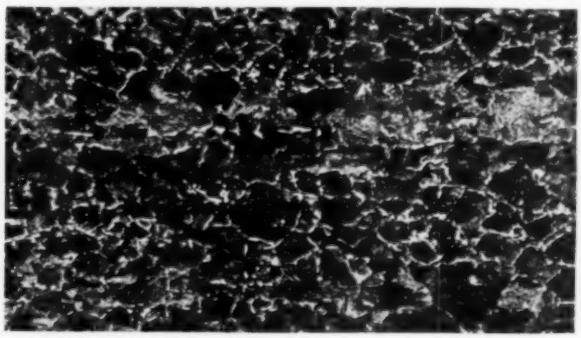


FIG. 14 BAR 424, 1 1/4-IN. ROUND MANGANESE CARBON STEEL × 100
(Appears not fully annealed before cold drawing.)

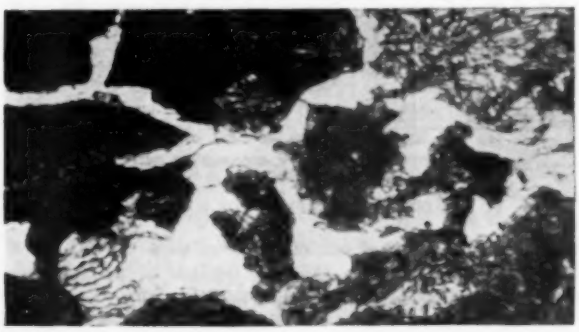


FIG. 15 BAR 424, 1 1/4-IN. ROUND MANGANESE CARBON STEEL × 1000
(Shows some laminated pearlite; little banding.)

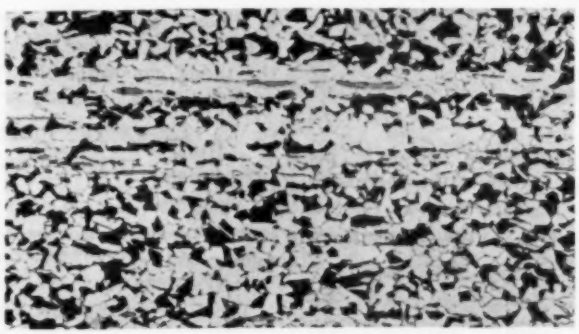


FIG. 16 BAR 426, 1 1/2-IN. ROUND SAE 1320 × 100
(Small grain size.)

which the majority of the experimental data was obtained. Comments as to each photomicrograph are given in the caption of each figure.

Machinability Data

The results of the various machinability tests are shown for the steel screw stock in Table 3. These data were determined by measuring the force on a planer tool, the torque and thrust of a $^3/_4$ in. diameter drill, the penetration in inches per 100 revolutions of a $^1/_4$ in. diameter drill, and the energy in foot-pounds required to remove a chip by milling, as described.

The total force F on a planer tool is the value in pounds actually exerted in the direction of cut on the planer tool $^1/_2$ in. wide

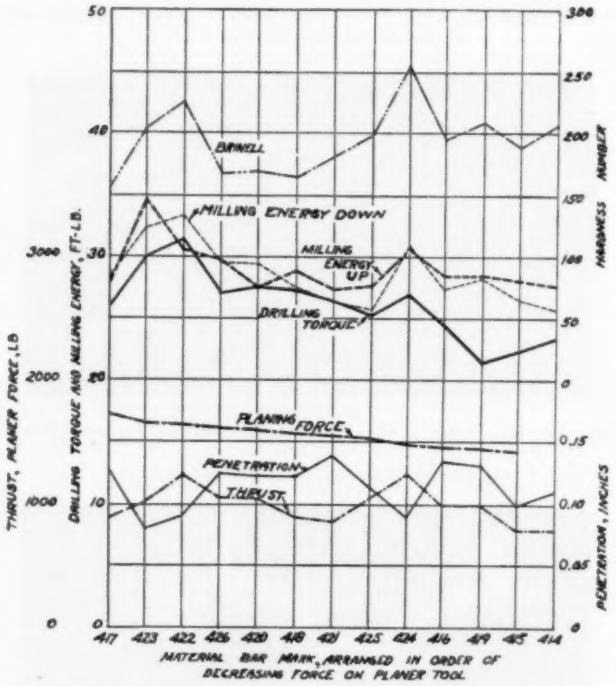


Fig. 17 Machining Properties of Screw-Stock Steels

when taking a depth of cut of 0.010 in. at a cutting speed of 20 f.p.m. The unit force U is the number of pounds exerted on 0.001 sq. in. of cross-sectional area of the chip.

$$U = \frac{F}{1000\ WD} = \frac{F}{5}$$

in which W = width of cut 0.5 in. and D the depth of cut 0.010 in.

For bar 419, SAE 1112,

$$U = \frac{1444}{5} = 288.8 \text{ lb.}$$

This unit force of 288.8 lb. is equivalent to 288,800 lb. per sq. in. of chip area. The horsepower developed by the planer tool equals $FS/33{,}000$, in which F is the total force on the tool in the direction of the cut in pounds and S is the cutting speed in feet per minute. As S equals 20,

$$\text{Hp.} = \frac{20F}{33{,}000} = \frac{F}{1650}$$

For SAE 1112, bar 419, the developed

$$\text{Hp.} = \frac{1444}{1650} = 0.875$$

The cubic inches of metal removed per minute = WDL = $0.5 \times 0.010 \times 20 \times 12 = 1.2$, in which L is the length of cut per minute in inches and equals $12S$. The horsepower per cubic inch of metal removed per minute equals the total power developed divided by the cubic inches of metal removed per minute, or

$$\frac{F}{1.2 \times 1650} = \frac{F}{1980}$$

For SAE 1112, bar 419, the horsepower per cubic inch of metal removed per minute is then 1444/1980, or 0.729.

To reduce the experimental data for the drill torque and thrust to net horsepower per cubic inch of metal removed per minute, the following method may be used. The total horsepower developed at the point of drill is the sum of that due to torque and thrust. The horsepower of the torque

$$\text{Hp.}_t = \frac{2\pi TN}{33{,}000} = \frac{2\pi \times 152T}{33{,}000} = 0.02894\ T$$

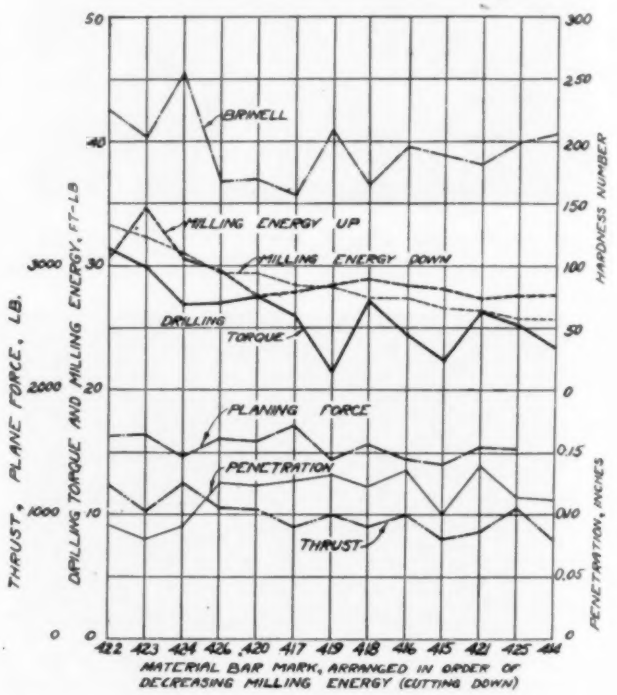


Fig. 17-a Machining Properties of Screw-Stock Steels

in which T equals the torque in foot-pounds and N equals the actual revolutions per minute of the drill. The horsepower of the thrust equals the thrust times the distance traveled per minute divided by 33,000, or

$$\text{Hp.}_{th} = \frac{fN\ Th}{33{,}000} = \frac{0.012 \times 152\ Th}{33{,}000 \times 12} = 0.000004606\ Th.$$

in which Th equals the thrust in pounds, f equals the feed in inches per revolution of the drill, and N equals the revolutions per minute of the drill. The total net horsepower Hp. = $0.02893\ T + 0.000004606\ Th$.

The cubic inches of metal removed per minute equal

$$\frac{\pi d^2}{4} fN = \frac{\pi(^3/_4)^2}{4} \times 0.012 \times 152 = 0.8068$$

in which d is the diameter in inches of the hole drilled, f is 0.012 the feed in inches per revolution, and N is 152 the revolutions per minute of the drill.

The net horsepower per cubic inch of metal removed per minute in drilling with a $^3/_4$-in. drill rotating at 152 r.p.m. and having a feed of 0.012 in. per revolution, for SAE 1112, bar 419, of Table 3, for which there is a torque of 21.5 ft-lb. and a thrust of 1000 lb., is

$$\frac{0.02894T + 0.000004606Th}{0.8068} = \frac{0.622 + 0.0046}{0.8068} = \frac{0.6266}{0.8068} = 0.777$$

The influence of the thrust on power is so small that it may be omitted.

For determining the horsepower per cubic inch of metal removed per minute in milling, using the experimental data, it is necessary to make the following assumptions, which include the conditions under which the experiments were run:

d = the depth of cut = 0.125 in.
w = the width of cut = 0.250 in.
f = the feed per tooth = 0.010 in.

A cutter is assumed to have 14 teeth and to rotate at 58.3 r.p.m., giving 816.2 chips per minute. The horsepower developed equals

$$\frac{816.2}{33{,}000} \times E = 0.024733\,E$$

in which E is the energy in foot-pounds required to remove one chip. For SAE 1112, bar 419, the horsepower developed is $0.024733 \times 28.5 = 0.705$.

The cubic inches of metal removed per minute equals $dwftN = 0.125 \times 0.250 \times 0.010 \times 14 \times 58.3 = 0.255$, in which t is the number of teeth in the cutter and N is the revolutions per minute of the cutter.

Therefore, the horsepower per cubic inch of metal removed per minute $= 0.024733E/0.255 = 0.097\,E$. For SAE 1119, bar 419, this becomes $0.097 \times 28.5 = 2.76$, as shown in Table 3.

Discussion of Machinability Data

Certain machinability data on the steel screw-stock bars tabulated in Table 3 are shown graphically in Figs. 17, 17*a*, and 18. In Fig. 17 the Brinell hardness numbers, the drilling torque and thrust, the planing force, the drill penetration, and two milling-energy values, for each bar, are plotted over that bar number. The bar numbers are arranged from left to right in order of decreasing force on the planer tool.

In Fig. 17*a* the same values as represented in Fig. 17 are shown plotted over the bar numbers arranged from left to right in order of reducing "milling energy down." The respective values have been joined by lines to indicate continuity.

At first only the "milling-energy-up" curve was prepared. The cut by the milling cutter was taken upward or against the feeding motion of the specimen. Subsequent tests on another project, however, showed that some materials yielded greater, equal, or smaller energy values when the direction of cut was reversed; that is, the cut taken down or with the feed of the material. It was thought desirable, therefore, to determine the values of milling energy on a down cut for these cold-drawn steels so as to make the information a little more complete. There are then two milling-energy curves, one

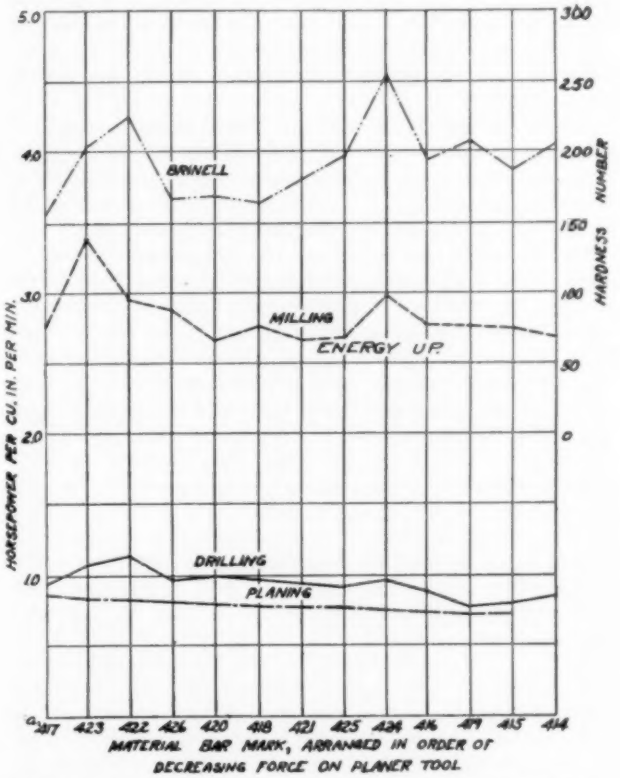


Fig. 18 Power Consumed in Cutting Screw-Stock Steels

TABLE 3—MACHINABILITY DATA ON STEEL SCREW-STOCK BARS

Bar No.	Material	Force of planer tool: "F," total	Force of planer tool: "U," unit	Force of planer tool: Hp. per cu. in. per min.	Drilling: Torque, ft-lb.	Drilling: Thrust, lb.	Drilling: Hp. per cu. in. per min.	Drill penetration, in. per 100 r.p.m.	Milling: "E," ft-lb. per chip	Milling: Hp. per cu. in. per min.	Brinell hardness
414	$1^1/_{16}$-in. rd. super-free cutting	...	...	...	23.4	790	0.844	0.111	27.63	2.680	207
415	1-in. rd. (imported)	1430	286.0	0.722	22.3	800	0.805	0.100	28.23	2.738	180
416	$1^1/_4$-in. sq. SAE 1112 cold-drawn, $^1/_{16}$-in. draft	1475	295.0	0.745	24.5	1000	0.887	0.135	28.53	2.762	196
417	$^3/_4$-in. sq. SAE 1120 cold-drawn, $^1/_{16}$-in. draft	1717	343.4	0.866	26.0	900	0.938	0.127	27.99	2.718	156
418	$1^1/_2$-in. sq. SAE 1120 cold-drawn, $^1/_{16}$-in. draft	1579	315.8	0.797	27.3	900	0.984	0.123	28.86	2.798	163
419	$^3/_4$-in. sq. SAE 1112 cold-drawn, $^1/_{16}$-in. draft	1444	288.8	0.729	21.5	1000	0.777	0.131	28.50	2.760	207
420	$1^1/_2$-in. rd. mang. screw stock cold-drawn, $^1/_{16}$-in. draft	1604	320.8	0.810	27.6	1050	0.995	0.123	27.51	2.670	170
421	1-in. rd. SAE 1315 cold-drawn, $^1/_{16}$-in. draft	1561	312.2	0.789	26.5	860	0.955	0.139	27.42	2.660	179
422	$1^3/_{16}$-in. rd. SAE 6140 cold-drawn $^1/_{16}$-in. draft	1646	329.2	0.831	31.5	1250	1.138	0.091	30.45	2.950	228
423	$1^1/_2$-in. rd. SAE 2320 cold-drawn, $^1/_{16}$-in. draft	1656	331.2	0.836	30.0	1020	1.082	0.081	34.83	3.380	205
424	$1^1/_4$-in. rd. mang. carbon alloy, $^1/_{16}$-in. draft	1497	299.4	0.756	27.0	1250	0.975	0.090	30.66	2.970	255
425	1-in. rd. special mang. carbon alloy $^1/_{16}$-in. draft	1534	306.8	0.775	25.2	1050	0.922	0.113	27.69	2.685	199
426	$1^1/_2$-in rd. SAE 1320 cold-drawn $^1/_{16}$-in. draft	1612	322.4	0.814	27.0	1050	0.975	0.124	29.73	2.880	168

indicated by "up" and the other by "down," indicating the method by which the values are obtained.

In connection with another investigation, it has been found that, by tests cutting dry, some metals require as much as 30 per cent less energy when the cut is taken down or with the feed than when against it. The power for feeding is not included. On the other hand, other metals require as much as 25 per cent more energy when the cut is down or with the feed. A wide variety of metals fill in the range between these two extremes and some show no difference in energy for the two types of cuts.

It also has been found that the introduction of a coolant or lubricant complicates this situation still further, as the saving in energy between dry and wet cutting accomplished on an up stroke frequently does not compare favorably with a similar saving between dry and wet cutting on the down stroke.

Fig. 17 shows that in general the Brinell curve slopes upward to the right, which indicates that the greater the Brinell number the lower is the planing force. The "milling-energy-down" curve slopes downward to the left to bar 424, for which there is a rise, after which the curve again slopes to the right. The "milling-energy-up" curve slopes downward to the right up to bar 425 and is practically horizontal to the right of bar 416. The drilling-torque curve, with a few inconsistent points, also slopes to the right. The thrust curve shows no particular trend, but might be considered as a horizontal line. The penetration curve shows perhaps a slight trend to increase to the right as might be expected in that the softer the metal the greater would be the drill penetration. In general, the various machinability and hardness curves of Fig. 17 do not agree favorably with the "planing-force" curve, as there are too many exceptional points on each curve.

In Fig. 17*a* the machinability and hardness values have been plotted over the bar numbers arranged from left to right in order of reducing "milling energy down." On this new basis a general curve, representing Brinell numbers, may take two forms: the first, excepting values for bars 426, 420, and 417, a straight line would be obtained sloping downward from left to right; the second, if values for bars 422, 423, and 424 are omitted, would give a straight line sloping from bar 426 upward to the right. The "milling-energy-up" curve slopes downward from left to right across the figure, except for the low values for bars 420, 417, and 419. Neglecting the first four points, however, would leave a horizontal line representing the values of bars 420 and those to the right.

Excepting points for bars 419 and 416, the drill-torque curve would slope gradually from left to right. The planing-force curve, excepting the values for bar 424 and 417, gives a satisfactory straight line sloping from left to right across the figure. The drill-thrust curve also gives a satisfactory general slope from left to right. The $^1/_4$ in. diameter drill-penetration curve, however, seems to follow more closely the Brinell curve in that two possible curved lines may be obtained as representing smoothed-out values: the first, for bars 422 to 421, inclusive, omitting the low values of bars 415, 424, and 423; or a second curve sloping from left to right for bars 426 to 414, inclusive, with values for bars 416, 415, and 421 omitted. This second curve of drill penetration, sloping from left to right for bars 426 to 414, inclusive, agrees favorably with a Brinell curve sloping upward from left to right for bars 426 to 414, inclusive. This is a satisfactory relation in that it naturally would be expected that the harder the material the less would be the penetration per hundred revolutions. This, then, indicates that bars 422, 423, and 424 are out of order.

Bar 422, SAE 6140, is shown in Fig. 10 to be not fully annealed before cold drawing. This structure undoubtedly plays some part in the odd results obtained. Bar 423 is the only $3^1/_2$ per cent nickel low-carbon steel bar in the group. This, too, may account for its unusual properties. Bar 424, a manganese-carbon steel bar, also appears not fully annealed before cold drawing, as shown in the photomicrographs in Figs. 14 and 15.

Fig. 17*a* shows an interesting relation between bars 417 and 418, each an SAE 1120 steel. It is seen that bar 417, which is cold-drawn to $^3/_4$ in. sq., has machining properties greater than 418, which is cold-drawn to $1^1/_2$ in. sq. The photomicrographs Figs. 5 and 6 indicate that bar 417 has received more cold drawing than bar 418. Similarly comparing bars 419 and 416, both SAE 1112 steels, it is seen that bar 419, cold drawn to $^3/_4$ in. sq., has higher machining properties than bar 416, which is cold-drawn to $1^1/_4$ in. sq. The drill torque, however, is higher for bar 416 than for bar 419. The same relation holds for bars 417 and 418, which might indicate that the drilling torque is less the finer the grain structure. The planing force for these two sets of bars is just the reverse, being higher for the smaller bars. The Brinell numbers are inversely proportional to the torque and planing force for these four bars and vary directly with the thrust and penetration.

Bar 417, SAE 1112, $^3/_4$ in. sq., is shown in Table 2 and in Figs. 17 and 17*a* as having the highest planing force. The value of 1717 lb. is high and apparently out of order. There may have been an error in copying this value from the data sheet, as it appears that 1517 lb. would be much more appropriate. There was not enough material left in the bar to run check tests, however, so that no further light can be gained on this point.

It is of interest to note the relative positions of bars 415 and 414, supposedly super-free-cutting screw-stock steel. Bar 415 has a low thrust, a low planing force, a low drill torque, an average "milling energy down" value, and an average Brinell for its position. The penetration, however, is low, whereas it should be high. Bar 414 has a Brinell number above the average for its position and has a low penetration.

Fig. 18 shows, more for commercial use, the value of horsepower per cubic inch of metal cut per minute by milling, drilling, and planing, of the various steel screw-stock bars. These curves show the same general tendency as the force or energy curves of Fig. 17.

It is realized that these tests have been performed somewhat contrary to actual commercial practice. Drilling for torque and thrust with a $^3/_4$-in. drill, for instance, has been done crosswise of the bars, whereas in most screw-machine work it is done lengthwise. The results of similar tests in drilling, however, have shown that the results in cross-drilling compare quite favorably with those of end-drilling. The force on the planer tool has been determined from a cut running lengthwise of the bar, whereas in turning the cuts are at right angles to the axis of the bar. No data are available to indicate whether one is higher than the other. All milling-energy values have been determined by taking cuts lengthwise of the bar. In several instances check tests have been made to show that the energy required to remove a chip lengthwise of the bar is equal almost exactly to that required when taking a cut at right angles to the bar. For example, Table 3 shows the milling-energy value for the cut lengthwise of bar 418 to be 28.86 ft-lb. per chip. Tests taken crosswise at the same time show the results to be 28.95 ft-lb. per chip, or about one-third of 1 per cent greater.

Chips From the $^3/_4$-In. Drill and Planer Tool. All chips from the cold-drawn steels numbered 416 to 425, inclusive, for the $^3/_4$-in. drill and the planer tool were retained and compared.

The torque chips for bars 422, SAE 6140 steel, 423, SAE

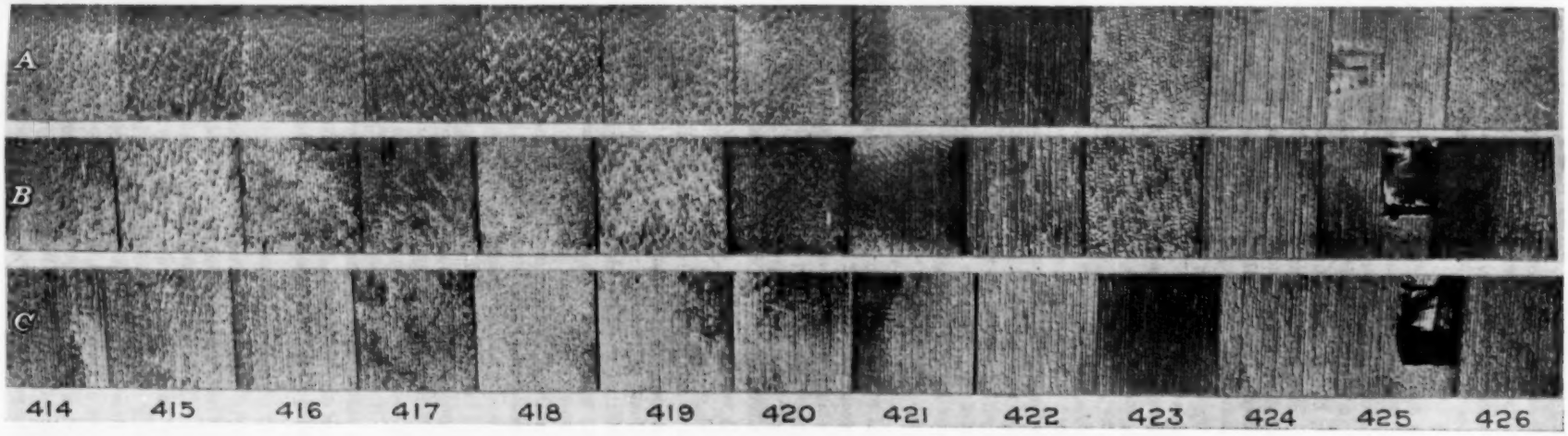


FIG. 19 FINISHED SURFACE ON RESPECTIVE SPECIMENS LEFT BY SHAPER TOOL × 1.8

(*A*, Tool 0 deg. front and 0 deg. side rake and 1/16 in. nose radius, the cut 0.005 in. deep, with 0.013 in. feed, 40 3-in. strokes per minute. *B*, same as *A*, but with 0.015 in. depth of cut. *C*, same as *A*, but tool 13 deg. front and 16 deg. side rake and 1/16 in. nose radius and 0.005 in. depth of cut.)

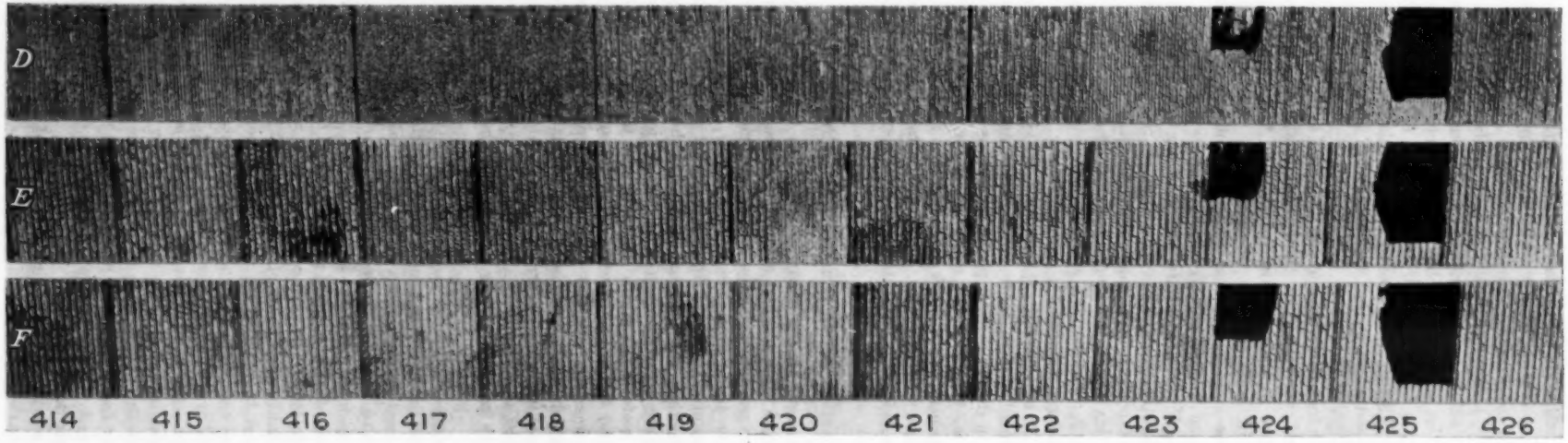


FIG. 20 FINISHED SURFACE ON RESPECTIVE SPECIMENS LEFT BY SHAPER TOOL × 1.8

(*D*, Tool 13 deg. front and 16 deg. side rake and 1/16 in. nose radius, the cut 0.015 in. deep, with 0.013 in. feed, 40 3-in. strokes per minute. *E*, same as *D*, but with 0.010 in. depth of cut and 0.027 in. feed. *F*, same as *E* except that speed of shaper was increased from 40 3-in. strokes to 82 3-in. strokes per minute.)

2320 steel, and 424, manganese-carbon, were in the form of coils from 1 to 3 in. in length. All three chips were somewhat blued, indicating the development of heat. While all tests were run dry, none of the other chips was blue. The drilling chip of bar 425, a special manganese-carbon steel, was well broken up, but some of the chip particles were slightly blued. The drill chips from bars 416, SAE 1120 (3/4 in. sq.), and 419, SAE 1112 (3/4 in. sq.), were coiled up to 1 in. in length, whereas the balance of the chips were well broken up into parts not to exceed 1/4 to 3/8 in. in length.

The planer tool gave chips on bar 422, SAE 6140, which were slightly blued, very rough on the bottom, and compact into a coil 3/4 inches in diameter and spiralled to 3/4 in. in length. The length of each cut was about 12 in. Bar 424, manganese-carbon steel, had planer chips not colored, rough and in the form of a small coil 5/8 in. in diameter and spiralled to 1 in. in length. The planer-tool chip of bar 423, SAE 2320, was very slightly blued and very rough on the bottom (next to the tool face). It was small and compact, a coil 7/8 in. in diameter. Bar 416, 1 1/4 in. sq. SAE 1112, and bar 419, 3/4 in. sq. SAE 1112, produced the smallest planer-tool chips of all, being in the form of a spiral 1/2 in. in diameter. Bar 417, 3/4 in. sq. SAE 1120, and bar 418, 1 1/2 in. sq. SAE 1120, gave planer-tool chips rougher than those for bars 416 and 419 and in a coil 7/8 in. in diameter. All planer-tool chips were continuous for the 12 in. length of cut, except those for bar 420, 1 1/8 in. round manganese screw stock. These were broken into two and sometimes three sections. In every instance there was a built-up nose left attached to the chip at the end of the cut for all bars.

Comparison of Surfaces Left by Shaper Tools

It was thought that a study of the finish left on each bar might prove of interest when a finish produced under a given condition on one bar was compared with that similarly produced on the other bars. Specimens of each bar, therefore, were prepared, approximately 1/2 in. thick and 2 in. long. These small sections then were clamped together in the vise of a shaper and a roughing cut was taken across all sections so as to present a flat surface, the width of which was the length of the specimens and the length of which was the sum of the widths of the specimens. The cuts were taken lengthwise of each specimen, which was lengthwise of the original bar. The bars were arranged in numerical order, with bar 414 at the left end and bar 426 at the right end. For each test cut the pieces were clamped with parallels on the bottom and the front to insure rigidity.

Fig. 19 shows the finished surface (× 1.8) on the various specimens left by a shaper tool. For the surfaces shown at *A* a tool was used having a zero degree front rake, a zero degree side rake, and a 1/16 in. nose radius. The cut was 0.005 in. deep and the feed was 0.013 in. per stroke. The shaper was set up to make 40 3-in. strokes per minute and the cut was made from right to left. The surfaces shown at *B* were made under the same conditions as those of *A*, except that the depth of cut was increased from 0.005 in. to 0.015 in. per stroke. There appears to be very little difference in the surfaces of cuts *A* and *B* for a single bar. Whether the surface *B* for bar 417 actually is smoother than that of *A* for the same bar is difficult to say. The photographs did indicate that *B* was the smoother. Also surface *B* of bar 418 appears to be smoother than surface *A*. On the other hand, surface *B* of bar 419 appears to be rougher than surface *A* for that bar, which might be expected because of the greater depth of cut.

It was noticed that, in taking the depth of cut of 0.005 in. with the zero degree front- and side-rake tool, the chips of bar 424 were of the curled type, being approximately 3/8 in. in diameter. The chips from all other bars, however, were straight and came off in one piece. When the depth of cut was increased to 0.015 in., using the same zero degree front- and side-rake tool, the chips from bars 422 and 424 were curled, while the rest were straight.

Cut *C* was made following cut *B*, in which a 13-deg. front-rake- and a 16-deg. side-rake-angle tool having a 1/16 in. nose radius was used with 0.005 in. depth of cut. This surface is shown at *C* in Fig. 19. Surfaces *C* are directly comparable to surfaces *A*, Fig. 19, except for the front- and side-rake angles on the tool. The surfaces shown for cut *C* give, in general, a smoother appearance than those for *A*. Bar 424 is a notable exception, whereas bars 414, 415, 418, 419, and 423 show considerable improvement.

The chips from cut *C* were curled for bars 424 and 425, whereas all others were straight.

Fig. 20 shows the finished surfaces at 1.8 magnifications on the various specimens left by a shaper tool. At *D* is shown the surface left by the 13-deg. front-rake and 16-deg. side-rake tool having 1/16 in. nose radius with a feed of 0.013 in., a depth of cut of 0.015 in., while the shaper provided 40 3-in. length strokes per minute. The surfaces from cut *D* correspond quite favorably with those of cut *C* for the same tool, although cut D was taken with a depth three times greater than that for cut *C*. The surfaces of cut *D* show a marked improvement over the surfaces of cut *B*. This improvement is due to the front and side rake of the tool, as otherwise the cutting conditions were identical.

Cuts *E* and *F*, shown in Fig. 20, were taken under the same conditions, except that in cut *E* the shaper made 40 3-in. strokes per minute, while in cut *F* 82 3-in. strokes per minute were made. In both instances the depth of cut was 0.010 in. and the feed was 0.027 in. The 13-deg. front-rake- and 16-deg. side-rake-angle tool having a 1/16 in. nose radius was used in both instances. It seems apparent that the finish of cut *F* is better than that of cut *E* taken at the slower speed. The surfaces of cuts *D* and *F* were enlarged another five times, making nine times natural size. It was thought that this magnification would bring out features of the surface not evident at the lower magnification. The results, however, were disappointing in that detail was lost due to the poor focus on the surfaces at different elevations.

It also is realized that the method of securing finished surfaces on these various bars by taking cuts 2 in. in length in a shaper was not the most desirable. Continuous cutting by turning would have been used had the specimens been adequate in size and amount for this purpose. This was not the case, however, so that the shaper had to be used. Just why some of the chips were curled while most of them were straight is not known. It is presumed that the formation of a built-up nose on the cutting edge of the tool varies with the material. The nature of the surfaces in all cases indicates the presence of this so-called built-up nose.

Stainless Steel

In addition to the 13 cold-drawn steel bars reported on, various machinability tests also were run on two stainless-steel or stainless-iron bars. Table 4 shows the chemical and physical properties of these bars, which were numbered 412 and 413. Bar 412 is called a stainless chromium iron and is in an annealed condition. Bar 413 is a free-machining rustless steel. The carbon content of both bars is low.

Photomicrographs of these two bars are shown in Figs. 21 and 22 at 100 magnifications. Bar 412, stainless chromium iron, has a very fine grain size and shows the results of the annealing,

while bar 413, the free-machining rustless steel, has the appearance of being cold drawn and has considerable banded sulphides. These specimens were etched with a standard ferric-chloride and hydrochloric-acid etch.

The results of the machinability data on these stainless-steel bars are given in Table 5. The force on the planer tool for the chromium-iron bar 412 was found to be 1586 lb., which compares favorably with the force for bar 418, the $1^1/_2$ in. sq. SAE 1120 steel. The chips from the $^1/_2$ in. wide planer tool, in making this cut, resembled very closely the chip of a fully annealed low-carbon steel. There was a built-up nose on the cutting edge of the tool, causing the under surface of the chip to be rough with rather parallel continuous cross ridges (resembling the teeth of a file) approximately $^1/_{16}$ in. or more apart. The surface left on the bar was very rough and showed slip lines quite similar to those on the bottom of the chip; that is, rather parallel ridges running crosswise of the cut approximately $^1/_{16}$ in. apart. The force on the planer tool for the free-machining rustless steel, bar 413, is only 1245 lb., which is lower than those for any of the cold-drawn screw-stock steels reported on. These chips were spiral in shape with the whole length of cut in one piece. The chip itself was burnished smooth on the bottom where it rubbed over the face of the tool, but was built up of chip sections slightly less than $^1/_{16}$ in. in width, which in themselves were made up of numerous shear elements. The sections were jointed only at their ends into the whole chip which gave a ladder-like construction. The surface of the chip away from the tool face was rough and made up of definite sections considerably larger than those for the annealed chromium-iron bar 412. A close examination of the chip from the planer tool for bar 413 does not show clearly any built-up nose on the cutting-tool edge. The surface left by the tool on bar 413 was considerably smoother than that on bar 412, but showed burnished lines, quite parallel across the cut $^1/_8$ in. apart.

TABLE 4 CHEMICAL AND PHYSICAL PROPERTIES OF STAINLESS STEEL

Bar No.	Material	Chemical analysis C	Mn	Phos	S	Si	Cr	Mo	Physical properties Tensile	Yield point	Elongation	Reduction area
412	Stainless chromium iron annealed.	0.06	0.31	0.012	0.022	0.18	13.35	..	90,000	50,000	30.0	60.0
413	Free-machining rustless steel.....	0.09	0.30	0.016	0.46	0.14	15.15	0.25	90,000	57,000	28.0	50.0

TABLE 5 MACHINABILITY DATA ON STAINLESS STEEL

Bar No.	Material	Force on planer tool "F," total	"U," unit	Hp. per cu. in. per min.	Drilling Torque, ft-lb.	Thrust, lb.	Hp. per cu. in. per min.	Drill penetration, in. per 100 r.p.m.	Milling "E," ft-lb. per chip	Hp. per cu. in. per min.	Brinell hardness
412	Stainless chromium iron annealed.	1586	317.2	0.961	49	1500	1.755	0.0444	59.1	5.730	....
413	Free-machining rustless steel.....	1245	249.0	0.754	25	800	0.900	0.0837	34.16	3.304	....

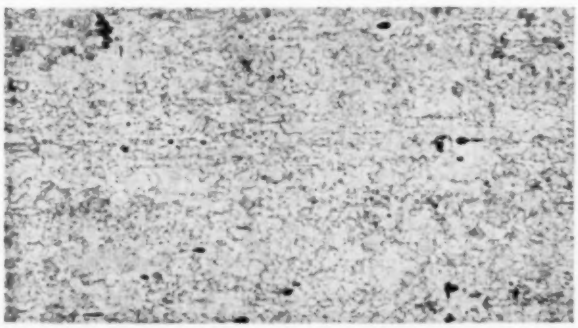


FIG. 21 BAR 412, STAINLESS CHROMIUM IRON × 100
(Very fine grain size; shows results of annealing.)

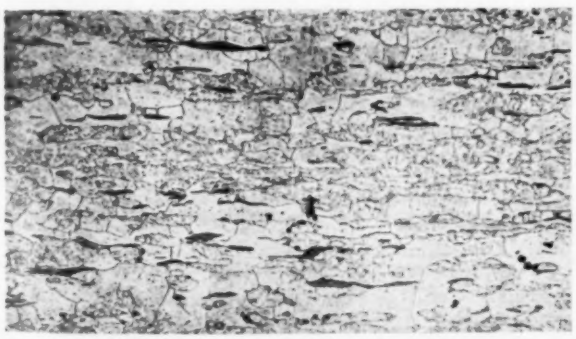


FIG. 22 BAR 413, FREE-MACHINING RUSTLESS STEEL × 100
(Has appearance of being cold-drawn; considerable banded inclusions.)

The torque for bars 412 and 413 is seen to be 49 and 25 foot-pounds, respectively. The value of 49 for bar 412 is considerably higher than the drilling torque for any of the cold-drawn steels reported on, as seen in Fig. 17, while 25 ft-lb. for bar 413 is rather an average value. The drill penetration of the stainless chromium-iron bar 412 is only 0.044 as compared with 0.081 for bar 423 which is the lowest in the group of steels. The drill penetration for bar 413, free-machining rustless steel, is 0.0837, which is in the range of bars 422, 423, and 424 of Fig. 17, the hardest of the group. The "milling energy up" for bars 412 and 413 is 59.1 and 34.16 ft-lb., respectively. The first value is much higher than any of the values for steel, while the second compares favorably with the values for the hardest steels.

CONCLUSIONS

The primary object of this paper has been to present certain machinability data obtained from a variety of cold-drawn steels, including two stainless steels. Because of the varied conditions under which these bars were manufactured and because of the differences in chemical analyses of supposedly similar bars, it is difficult to draw definite conclusions as to correlation between the machinability data, the structure, and the type of steel involved. Only by reference to the various tables and the texts can completely satisfactory conclusions be presented. Some conclusions, however, may be mentioned as follows:

(1) Data as to cutting force or energy required in planing, milling, and drilling, under similar standardized conditions, for 15 different kinds of cold-drawn steels are given. These data are directly comparable with similar data obtained from 18 ferrous metals and 21 non-ferrous metals, previously reported on in Trans. A.S.S.T., November, Vol. 16, pp. 659 to 710.

(2) Data as to horsepower per cubic inch of metal per minute are presented for each of the 15 cold-drawn steels.

(3) An attempt is made to determine the correlation between the machining properties of each cold-drawn steel and its hardness. A satisfactory correlation is obtained for some steels, but many show inconsistencies.

(*a*) The machining properties of each steel depend upon

its chemical composition, its heat treatment and method and degree of cold working, as shown finally by the resulting structure, and by its carbon content.

(*b*) The alloy steel bars 422, SAE 6140, and 423, SAE 2320, obviously do not belong in the family of plain carbon or manganese steels so far as their position on the machinability diagram is concerned.

(*c*) There appears to be little consistent correlation between the various machining properties of the steels as shown in Fig. 17*a*. One bar may show a low torque but high milling energy or a low penetration and low planing force, etc. The size of the bar, resulting from more intensive cold working for materials of similar chemical analysis, does not seem to cause wide variations in machining properties. Fig. 17*a* shows that bars 417 and 418, which are SAE 1120 steels $^3/_4$ in. sq. and $1^1/_2$ in. sq., respectively, have approximately the same machining and hardness values, in spite of their difference in size. Bar 417 has small grain size while bar 418 has medium grain size, which shows the effect of cold working. Also bars 416 and 419, which are $1^1/_4$ in. sq. and $^3/_4$ in. sq., respectively, SAE 1112 steel, show practically the same machining properties and hardness properties in spite of their difference in size. Bar 416 has medium grain size while bar 419 has small grain size, which shows the effect of cold working. The torque on bar 419, the smaller bar, is less than that for bar 416. The influence of size on tool life has not been investigated in this paper.

(*d*) The finish produced on the high-sulphur screw-stock steels appears to be better (than the finish on the alloy steels or manganese steels) for both the tools having 0-deg. front rake and 0-deg. side rake and 13-deg. front rake and 16-deg. side rake. Under the same cutting conditions, the high-sulphur and high-manganese steels have the best finish, the SAE 6140, bar 422, and SAE 2320, bar 423, being the poorest with the 0-deg. front-rake and 0-deg. side-rake tool, while bar 424, manganese-carbon steel, has the poorest finish, with the 13-deg. front-rake and 16-deg. side-rake tool.

(*e*) From Fig. 17*a* it is seen that, by omitting the first three steels at the right, which are SAE 6140, SAE 2320, and manganese-carbon steels, the Brinell curve slopes upward to the right, while the thrust curve slopes downward to the right; the penetration curve slopes downward to the right; the planing-force curve slopes downward to the right; the drilling-torque curve slopes downward to the right; and the "milling-energy-down" curve slopes downward to the right. This indicates that the higher the Brinell hardness the easier the bar machines.

(4) It is seen from Fig. 17*a* that each bar gives a certain energy required to remove a given chip by milling when the cut is "up" and a higher, equal, or lower value when the cut is "down." This indicates that the chemical analysis and structure influence the energy required to remove a chip, rather than any inherent difference in cutting process. The "milling-energy-up" curve in Fig. 17*a* seems to indicate no correlation with other machining or hardness values.

(5) The thrust curve seems to follow the Brinell curve almost exactly from point to point, as shown in Fig. 17*a*.

(6) The planing curve seems to follow the Brinell curve inversely from point to point, as shown in Fig. 17*a*, as does the penetration curve with a few exceptional points.

(7) The drilling-torque curve shows a reverse characteristic of the Brinell curve from point to point, as shown in Fig. 17*a*.

Acknowledgments

Grateful acknowledgment is made for the assistance of the Department of Engineering Research for meeting the expense of making the photomicrographs and various chemical analyses. The photomicrographs were made by Mr. Richard Schneidewind, of the Department of Engineering Research, and they were reviewed by Professor Claire Upthegrove, of the Department of Chemical Engineering. The machinability tests were made in most part by Mr. Charles E. Kraus and other former student assistants.

Most of the steel bars, together with the chemical analyses, were furnished by Mr. M. N. Landis, of Landis & Landis, chemists and metallurgists, Chicago, Ill., while the balance have been furnished by various users and manufacturers.

Discussion

M. N. Landis.[4] The writer has made a survey of the results obtained, taking into consideration a general knowledge of their

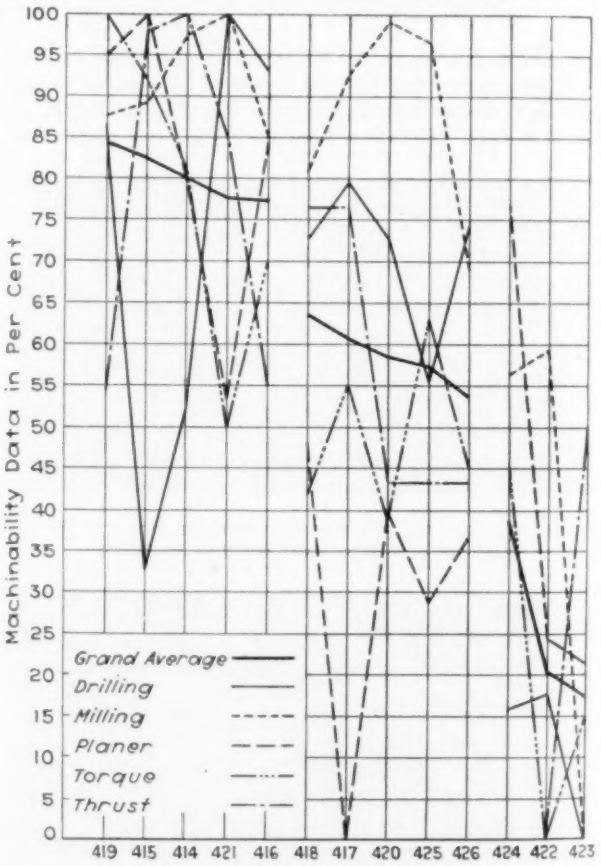


Fig. 23 Machining Properties

comparative machining properties from a production standpoint and has divided them into three groups:

First Group. Bessemer steels with low carbon and high sulphur:

414, $1^1/_{16}$-in. round super-free cutting.

415, 1 in. round (imported).

416, $1^1/_4$-in. square S.A.E. 1112 cold-drawn, $^1/_{16}$-in. draft.

419, $^3/_4$-in. square S.A.E. 1112 cold-drawn, $^1/_{16}$-in. draft.

Second Group. Open-hearth steels with high sulphur:

417, $^3/_4$-in. square S.A.E. 1120 cold-drawn, $^1/_{16}$-in. draft.

418, $1^1/_2$ in. square S.A.E. 1120 cold-drawn, $^1/_{16}$ in. draft.

420, $1^1/_8$-in. round manganese screw stock cold-drawn, $^1/_{16}$-in. draft.

[4] Landis & Landis, Chicago, Ill.

TABLE 6 MACHINABILITY DATA

		Milling, ft-lb. per chip, per cent	Drill penetration, in. per 100 r.p.m., per cent	Drilling: Torque, ft-lb., per cent	Drilling: Thrust, lb., per cent	Planer tool, force "F," total per cent	Average rating, per cent
419	3/4-in. square S.A.E. 1112	85.46	86.20	100.00	54.35	95.02	84.21
415	1 -in. round (imported)	89.10	32.90	92.00	97.83	100.00	82.37
414	1 1/16-in. round super-free cutting	97.20	51.80	81.00	100.00	80.00[a]	80.00
421	1 -in. round S.A.E. 1315	100.00	100.00	50.00	84.79	53.27	77.61
416	1 1/4-in. square S.A.E. 1112	85.05	93.10	70.00	54.35	84.24	77.35
418	1 1/2-in. square S.A.E. 1120	80.60	72.40	42.00	76.09	48.04	63.83
417	3/4-in. square S.A.E. 1120	92.34	79.30	55.00	76.09	00.00	60.55
420	1 1/8-in. round manganese screw stock	98.82	72.40	39.00	43.48	39.42	58.62
425	1 -in. round special manganese carbon alloy	96.39	55.20	63.00	43.48	28.90	57.40
426	1 1/2-in. round S.A.E. 1320	68.85	74.20	45.00	43.48	36.56	53.62
424	1 1/4-in. round manganese carbon alloy	56.30	15.70	45.00	00.00	76.58	38.72
422	1 3/16-in. round S.A.E. 6140	59.13	17.40	00.00	00.00	24.73	20.25
423	1 1/8-in. round S.A.E. 2320	00.00	00.00	15.00	50.00	21.25	17.25

[a] This result estimated on basis of other tests for this sample to complete results.

TABLE 7 VARIATION OF TEST RATINGS FROM GRAND AVERAGE

		Drilling torque, ft-lb., variation, per cent	Drill penetration, variation, per cent	Planer, total force "F," variation, per cent	Drilling, thrust variation, per cent	Milling, ft-lb., per chip, variation, per cent	Total variation from average, per cent
419	3/4-in. square S.A.E. 1112	15.79	1.99	10.81	29.86	1.25	59.70
415	1 -in. round (imported)	9.63	49.47	17.63	15.46	6.73	98.92
414	1 1/16-in. round super-free cutting	1.00	28.20	00.00	20.00	17.20	65.40
421	1 -in. round S.A.E. 1315	27.61	22.39	24.34	7.18	22.39	103.91
416	1 1/4-in. square S.A.E. 1112	7.35	15.75	6.89	23.00	7.70	60.69
418	1 1/2-in. square S.A.E. 1120	21.83	8.57	15.79	12.26	16.77	75.22
417	3/4-in. square S.A.E. 1120	5.55	18.75	60.55	15.54	31.79	132.18
420	1 1/8-in. round manganese screw stock	19.62	13.78	19.20	15.14	40.20	107.94
425	1 -in. round special manganese carbon alloy	5.60	2.20	28.50	13.92	38.99	89.21
426	1 1/2-in. round S.A.E. 1320	8.62	20.58	17.06	10.14	15.23	71.63
424	1 1/4-in. round manganese carbon alloy	6.28	23.02	37.86	38.72	17.58	123.46
422	1 3/16-in. round S.A.E. 6140	20.25	2.85	4.48	20.25	38.88	86.71
423	1 1/8-in. round S.A.E. 2320	2.25	17.25	4.00	32.75	17.25	68.50
Total variation from average		151.38	224.80	247.11	254.22	271.96	....

421, 1-in. round S.A.E. 1315.

425, 1-in. round special manganese carbon alloy cold-drawn, 1/16-in. draft.

Third Group. Alloy steels:

422, 1 3/16-in. round S.A.E. 6140 cold-drawn, 1/16-in. draft.

423, 1 1/8-in. round S.A.E. 2320 cold-drawn, 1/16 in. draft.

424, 1 1/4-in. round manganese carbon alloy cold-drawn, 1/16-in. draft.

Recapitulation of the machinability data obtained in the various tests has been made on the basis of the best machining specimen being rated 100 per cent, the poorest 0 per cent, and the others graded between. These results are arranged in Table 6 in the order of their comparative ratings.

Analysis of these results, with the machinability calculated in percentages, indicates that the averages of the various tests follow very closely the generally accepted production machinability rating. There is the exception of No. 421, which apparently has a high rating, but it has been the writer's experience that certain items of the manganese screw-stock steels machine exceptionally well. In any one of the types of steel tested considerable variation in machinability rating will be encountered, such as is displayed by the two samples of S.A.E. 1112.

Analysis of the tests conducted, calculated by determining their variation from the average machinability rating for each specimen, indicates that the torque results more nearly approach the grand average, and the milling results are farthest from the average. These results are arranged in Table 7.

The hardness tests have not been considered in analyzing the results, as it has been the writer's experience that they are not sufficiently indicative of the degree of machinability in testing screw stocks. The drilling tests, unfortunately, are dependent on the grinding, and, as stated by the author, it is "almost impossible to regrind the drill, even on a machine, to give results comparable with those of first grinding." If this variable could be overcome, the writer feels that the drilling tests should give a very desirable indication of machinability.

Alvan L. Davis.[5] It would be interesting, and the writer believes valuable, if the author would add a brief appendix showing the maximum deviation and also the average deviation from the mean for a series of, say, ten determinations, all on the same test material, for the planer, for drill torque, and for the miller. Such maximum and average deviations would allow an estimate of the probable error pertaining to the mean figures in the paper.

In the absence of such means of arriving at the probable error, the writer has assumed that the figures are not to be relied upon further than two significant figures, and in order to simplify discussion as much as possible, has used two significant places only in preparing the tabulation of horsepower per cubic inch of metal removed per minute. (See Table 8.) An arbitrary separation of the materials has been attempted into classes, according to the power required for machining. The idea of this simplified table is to attempt to bring out more clearly the extent to which the different kinds of machinability tests are in accord with each other, in classifying various steels as to machinability.

The writer attaches relatively small importance to the drill-penetration tests, because considerable experience with this type of test has led him to conclude that it is subject to rather wide variations due to the metal loading on the bottom of the drill and altering its rate of cutting. Nevertheless, the last column of the table does show the penetration, thus permitting comparison with the "horsepower required to remove 1 cubic inch per minute."

Table 8 shows that all four methods of test are in agreement; that class 1 (which is ordinary stainless chrome iron) requires the greatest power for machining, and that it yields the shallowest penetration under constant load. Likewise, all four tests agree that class 2 (comprising S.A.E. steels 2330 and 6140) stands next in point of resistance to machining operations; also, class 3 (made up of the high-manganese steels and the open-hearth screw stocks) requires still less power and yields deeper penetration under constant load. Passing by class 4 for a moment and

[5] Research Engineer, Scovill Manufacturing Company, Waterbury, Conn. Mem. A.S.M.E.

TABLE 8 HORSEPOWER PER CUBIC INCH OF METAL REMOVED PER MINUTE

Class	Type of steel	Bar No.	Planer	Drill	Miller	Penetration of drill under constant load
1	Ordinary stainless chrome iron	412	0.96	1.76	5.7	0.044
2	S.A.E. 2320	423	0.94	1.08	3.4	0.081
	S.A.E 6140	422	0.93	1.14	3.0	0.091
3	S.A.E. 1320	426	0.81	0.98	2.9	0.124
	S.A.E. 1310	420	0.81	1.00	2.7	0.139
	S.A.E. 1315	421	0.79	0.96	2.8	0.139
	S.A.E. 1340	424	0.76	0.98	3.0	0.090
	S.A.E. 1120	418	0.80	0.98	2.8	0.123
	S.A.E. 1120	425	0.78	0.92	2.7	0.113
			0.76-0.81	0.92-1.00	2.7-3.0	0.090-0.139
4	Free-cutting stainless chrome iron	413	0.75	0.90	3.3	0.084
5	S.A.E. 1112	416	0.75	0.89	2.8	0.135
	S.A.E. 1112	419	0.73	0.78	2.8	0.131
	Super-free cutting, imported	415	0.72	0.81	2.7	0.100
	Super-free cutting, domestic	414	0.00	0.84	2.7	0.111
			0.70-0.75	0.78-0.89	2.7-2.8	0.100-0.135

considering class 5 (which comprises Bessemer screw stock and also more particularly the super-free-cutting steels, both imported and domestic), the table shows planer and drill in agreement in grading these steels as easier to cut than any of the foregoing classifications. The miller shows only very slight improvement, however, over class 3 steels, and the penetration test here fails wholly to denote the relative machinability.

Class 4, the free-cutting stainless chrome iron, appears to be intermediate between classes 3 and 5 as measured by planer and drill. The miller, however, is not in accord in this respect, and would place this material along with S.A.E. steels 2320 and 6140 in class 2. Why this should be so the writer is not in a position to state, but it presumably arises from the nature of the milling test itself, which apparently is influenced by other properties than those which control the results in planing and drilling. In commercial machining it has been observed (in the case of certain alloy steels and also tool steels) that the degree of annealing giving optimum results for drilling and planing will not always give the best results for milling. Also, thread tapping sometimes requires a still different degree of annealing to give the best results.

Thus there is found quite general agreement between the three tests based on cubic inches of metal cut. However, it is disappointing that there should be so little difference between the super-free-cutting steels (bars 414 and 415) and the ordinary open-hearth screw stock (bars 418 and 425), especially in view of the fact that actual shop tests have proved these two super-free-cutting steels to give about 100 per cent greater production on automatic screw machines than is possible with ordinary open-hearth screw stock.

No stress in this résumé is placed upon hardness as indicated by the various tests recorded. This is partly because there is not complete uniformity in the different hardness tests reported for some of the specimens, but also because in the case of cold-drawn bars the hardness is built up out of two factors: first, the hardness of the bar before being subjected to cold drawing, and, second, the amount of hardness conferred by cold drawing. While the hardness resulting from cold drawing seems to promote free cutting, the initial hardness acts in the reverse direction. Therefore, no consistent results can be looked for, unless it is known how much of the final hardness is due to cold work and how much to initial hardness of the steel.

The writer's experience in testing machinability shows it to be an elusive factor, and he is not surprised to find lack of complete accord between different methods of measuring it.

ALFRED V. DE FOREST.[6] This paper, as well as the author's previous one on machining properties, well deserves the full attention of all those interested in problems of machinability. Speaking from the standpoint of the steel maker, particularly the cold-drawn and free-cutting steel industry, the question as to whether an intrinsic property of the steel loosely covered by the term machinability can be reduced to some system of measurement is of great consequence. The measurements made by the author undoubtedly have a bearing on the problem, but the factors which cause a steel to be successful or the reverse in actual practice are usually more complicated than questions of energy absorption. To get the best results out of a given steel, the shape and material of the cutting tools, the lubricant, and the type of machine used all play a very vital part, and usually in actual practice the machinability of a steel is judged by whether it works well under the cutting conditions which have been established by custom for some other materials. The supplier of steel must then endeavor to change the operation of the machine tool so as to get the best out of the material he supplies. In some cases this can be done. Frequently it cannot even be attempted, and the steel is blamed for any failure.

The writer recently came across a good example of a case in which machinability was important. A vital part was being manufactured from screw stock on screw machines. Complaints as to quality caused the substitution of a low-carbon alloy steel without free-machining properties. Production immediately dropped to almost nothing, and the steel was pronounced unmachinable. However, with assistance and encouragement, it developed that if the steel were given an extra draft of cold work and the tooling and lubrication were changed, almost the same production could be obtained as with the free-machining steel previously used. In this case it is not doubted that a similar amount of attention bestowed on the free-machining steel would have raised the original rate of production far above that obtainable by the alloy steel, but in this case a little attention and perseverance saved a job which from the protests of the manufacturer appeared to be impossible.

This case also brings out a point not very well covered in the paper, and that is that under certain conditions the additional hardness due to cold work is of advantage in machining operations. Probably conditions of this kind have discredited hardness tests as having any bearing on the machinability problem.

If one separates hardness due to chemical constitution from hardness due to cold work and heat treatment, it is possible that a better correlation can be obtained. Hardness, however, is rather remote from the strength, hardness, and ductility of the metal just beyond the tool point at the temperature which exists during the cutting operation, and the properties at this point are undoubtedly those which control machinability.

It is very much to be hoped that the machine-tool industry, the tool-steel manufacturer, and those interested in supplying steels for machining join together and clear up the lack of knowledge surrounding the whole subject.

AUTHOR'S CLOSURE

The author is grateful indeed to those who have discussed this paper, as each discusser added a new viewpoint, together with

[6] Consultant, Wyckoff Drawn Steel Company, Pittsburgh, Pa.

his different method of analysis of the experimental data. Each discusser is well qualified to make comments in view of his years of experience with these problems.

The author fully appreciates that the experiments involved in this study of machinability do not include all desirable methods of tests. This work was done by the author and a few students with practically no financial assistance.

The analysis by Mr. Landis is extremely interesting and valuable. It would seem to the author in considering equipment, condition of tools, etc., that the planer tests would represent more nearly the machining properties, as far as power is concerned, of the various metals. The drilling tests, also, would seem to be quite reliable, inasmuch as a drill of a size of $^3/_4$ in. diameter can be reground on our grinding machine so that tests can be duplicated with a great deal of satisfaction.

The author believes the quotation of Mr. Landis regarding the impossibility of regrinding a drill to give the same results with different grinds was made in the paper to apply particularly to the $^1/_4$-in.-diameter drill as used in the penetration test. The $^3/_4$-in. drill can be reground repeatedly to give practically the same experimental results. The author has conducted tests as to the influence of dulling on a drill in use and has found that, when cutting dry, after the first few inches drilled, there is little difference in the torque and thrust values in drilling an additional 60 or 70 in. of metal. This dulling effect seems to be more perceptible, however, when a coolant is used. The cutting actions of the $^3/_4$-in.-diameter drill and the planer tool seem to be quite similar and the results of the test in quite satisfactory agreement. The drill penetration test, however, is obviously a different principle, as is that of milling. The author has considerable information dealing with the results of milling different metals when the cut is with the feed and against the feed of the work. For some metals the results are practically the same; on other metals there is quite a divergence. Some metals require as much as 25 per cent less power when the milling cut is taken with the feed rather than against the feed, while other metals require as much as 25 per cent more. This is brought out in Figs. 17 and 17-*a* in which the milling-energy values for each material are shown when cutting up and when cutting down. By cutting down is meant cutting with the feed.

These differences are not as outstanding in Figs. 17 and 17-*a* as though there were a greater divergence in types of materials, although the divergence is due, in the author's opinion, to the influence of the material characteristic rather than through inaccuracy of the test methods.

Mr. Davis raises the question as to the maximum deviation from the average of the results obtained by the different machinability tests. In determining torque and thrust with the $^3/_4$-in.-diameter machinability drill, torque values from individual tests may vary 2 or 3 per cent in the same material. An average of three or four tests, however, will agree with the average of two or three other similar tests run at another time within 1 per cent. In obtaining the torque values for bars 417 and 419, each of which was $^3/_4$ in. square, two pieces of the same bar were clamped together and then drilled with the point of the drill to one side of the separating plane. The area of the cut, however, was approximately two-thirds in one piece and one-third in the second piece. This would lead to a condition approximating intermittent cutting with the bumping effect eliminated. While the author has made no tests to compare the results obtained by this method of drilling with those obtained when drilling into one solid piece, yet it is believed that the error is not appreciable.

The planer-force values tabulated here are an average of the readings on 10 successive cuts. The tool was fed 0.010 in. downward for each cut, using a dial gage attached to the planer head. While each cut was supposed to be 0.010 in. in depth, it may have been slightly less in one cut and slightly more in the following cut, so that maximum deviations between individual readings may be 4 or 5 per cent. In taking 10 cuts, however, the accumulated depth of cut would be 0.100 in., as close as could be read on the dial indicator. The average of one set of 10 readings, however, will agree with the average of another set of 10 readings within $^1/_2$ to 1 per cent. The drill penetrator tests will vary possibly 12 or 14 per cent between individual tests, as a small error in time of starting and stopping the watch during the 0.15 to 0.20 minute required to penetrate the 0.100 in. amounts to a large percentage. But again an average of 6 or 12 tests will agree with the average of another set of 6 or 12 readings within 2 or 3 per cent.

In obtaining the milling data, individual tests on these cold-drawn steels varied as much as 6 per cent, but by taking 10 consecutive readings an average would be obtained which would agree within $^1/_2$ to 2 per cent of similar average readings obtained from another set of tests on another bar. It seems, therefore, that where these different methods fail to confirm machining characteristics of a metal, the difference is due to the reaction of that metal to the particular machining process rather than excessive errors in the test data. It is for just this reason that these four different methods of tests were all used in obtaining the information reported on in this paper.

The point brought out by Mr. Davis dealing with optimum heat treatment for best machining purposes is one on which further accurate information is needed. The influence of structure and grain size appears to be a point on which little agreement exists. Just why the methods of tests employed in this paper do not agree with the results of tool life cannot be explained here, inasmuch as the author has no definite information on tool life. Not enough material was available to obtain data of this nature.

Mr. Davis and Mr. De Forest both refer to the hardness of a piece of cold-worked steel as consisting of two factors: the initial hardness of the steel and the final hardness after being cold-worked. The author has no information to throw light on this complicated subject, but appreciates the value of more detailed information dealing with this subject. Mr. De Forest has raised the question of optimum tool size and shape and condition of the material being cut, as well as the best lubricant or coolant to be used. Data are now being accumulated dealing with the performance of various types of cutting fluids, which may help to explain some of these factors. Optimum conditions for cutting by any process do not seem to be available.

MSP-53-7

Mechanical Design of Electric Motors as Regards Standardization Interchangeability

By J. L. BROWN,[1] EAST PITTSBURGH, PA.

The electric motor has become a machine simply to do a job, and yet because it is vital and because the failure of a unit means its immediate replacement with a new one, there has been a growing need for uniformity in physical dimensions of units. The rapidity with which uses for electric motors are multiplying calls for the lowest possible cost of manufacture of what may be called the standard type, but also of the many variations necessary to meet various conditions. This dictates uniformity also of the many accessory parts.



TIMES have changed for the electric motor. The halo which it wore as it assumed form out of the mists of the nineteenth century has faded. The mantle of mystery has been transformed into the tunic of service. The child of the scientists has become the patient ox of industry. No longer is the electric motor given a place of prominence and consideration, having subordinated to it the structure and characteristics of the driven machinery. It has become simply a machine to do a job—any job; a tool for use—and abuse. In the vast variety of elements making up the modern world, it is but a cog in the wheels of industry—a thread in the fabric, a link in the chain, a bolt in the structure. Perhaps the keystone of the arch, but still only one stone among many.

Yet because it is vital, and the failure of one unit grown old in service must be remedied instantly by replacing it with a new, there has been a growing need for a greater measure of uniformity in physical dimensions of units, which are otherwise sufficiently similar to do the same job, irrespective of the manufacturing plant in which they originate. Not only so, but the rapidity with which uses for electric motors are multiplying calls for the lowest possible cost of manufacture of not only what might be regarded as the standard type, but the many variations from this standard that are necessary to meet the various conditions of service. This dictates every possible measure of uniformity among the various standard and semi-standard accessory parts composing the variations of any given motor unit, in the interest of both the tooling equipment used in manufacture and the ease and rapidity of assembling from a given set of parts any type of unit which may be required for any particular class of service.

Standardization of a simple line of hardware such as bolts and nuts, though attended with many difficulties, is by no means the complex problem encountered in the standardization of motor dimensions. In addition to the necessity of effecting an agreement among motor manufacturers, it is necessary also to effect a compromise among types of motors which in many of their detail parts are essentially dissimilar. Consideration of details regarding the possible extent of standardization of external mounting dimensions of electric motors is beyond the scope of this paper. Nevertheless, it may be interesting and profitable to consider briefly the development of one of the newer lines of motors which demonstrates to what extent standardization as among dissimilar detail motor parts may be effected, thus resulting in great flexibility in machining and assembly methods when producing units which, although similar in external dimensions, are essentially different in type.

The simplest form of electric motor, and at the same time the one most widely used and manufactured in the largest quantities, is the so-called squirrel-cage induction type. The rotating element of this motor has been brought to the irreducible minimum in simplicity and economy of manufacture. Fig. 1 shows such a rotating element consisting of a shaft *a*, a stack of sheet-steel laminae *b*, and a grid of aluminum alloy *c* passing through slotted openings in the laminae and forming, with the continuous end-ring portions, the whole system of electrical conductors in the rotating element, as well as the necessary ventilating means, in the form of axially extending blades of the same die-cast structure. It will thus be seen that there are three principal parts only —no commutator, no collector rings, no wire windings, no insulation, no band wires or slot wedges to retain otherwise unsupported parts of the structure, no supporting plates for the sheet-steel laminae; even core rivets to hold them in assembly are a superfluity.

Not all applications for electric motors, however, are most satisfactorily served by a constant-speed, polyphase, alternating-current motor, and in meeting the requirements of certain applications, types of motors must be used in which the simplicity of the squirrel-cage induction type is to some extent lost. Variable-speed induction motors, for example, require the insertion of resistance into the rotor winding circuit necessitating that these windings be insulated from each other and the core and connected together in such a way that terminals may be brought out and connected to the necessary resistance through collector rings and brushes. This change in the type of winding involves the addition of insulation, band wire, and slot wedges to hold the coils in place, separate clamping means for the core of sheet-steel laminae, and separate blowing means to secure ventilation, while the addition of the collector rings makes the rotor unsymmetrical about the center line.

For a number of purposes the characteristics of direct-current motors outweigh in importance the simplicity and low cost of the induction motor. In this type the rotor windings are arranged much as are those of the variable-speed induction type, but are attached to a commutator instead of to collector rings. Fig. 2 gives a comparison of the three types of rotors. It should be noted that in spite of their dissimilar electrical features, the

---

[1] Mechanical Engineer, Industrial Motor Engineering Department, Westinghouse Elec. & Mfg. Co. Mr. Brown was born near Clinton, Ontario, Canada, in July, 1891. He received his early elementary education at country school, and was graduated from Clinton Collegiate Institute in 1908, after completing academic, commercial, and elementary scientific courses of study. He acquired electrical and mechanical engineering and business training by private study. Mr. Brown has specialized in mechanical design engineering as related to electrical apparatus with the Crocker-Wheeler Co., Ampere, N. J., and the Westinghouse Company since 1912.

Presented at the Fourth National Machine-Shop Practice Meeting, Chicago, Ill., September 22 to 24, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

three are similar in diameter and length from the center line toward the right; that the two last are similar to each other from the center line toward the left; and that that part of the first from the center line toward the left is similar to that part of all three from the center line toward the right.

Turning to a consideration of the stationary structures, Fig. 3 shows that whereas a frame or stator portion of exactly similar description is suitable for both constant- and variable-speed induction motors, the direct-current motor-frame assembly is greatly dissimilar. The magnetic circuit in the stationary part of induction motors is confined to a stack of sheet-steel laminae carried in a frame structure which furnishes physical strength and means of attachment to the driven apparatus or foundation. The laminated core structure is usually much smaller in diameter than the maximum diameter of the supporting frame, due to the necessity of providing passages for ventilating air. On the other

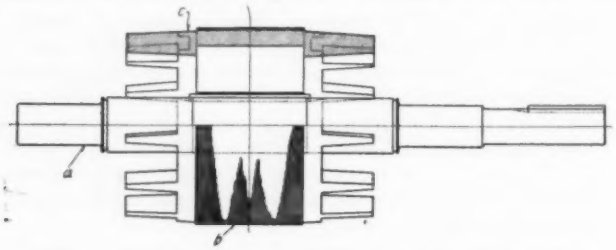


FIG. 1 ROTOR FOR SQUIRREL-CAGE INDUCTION MOTOR

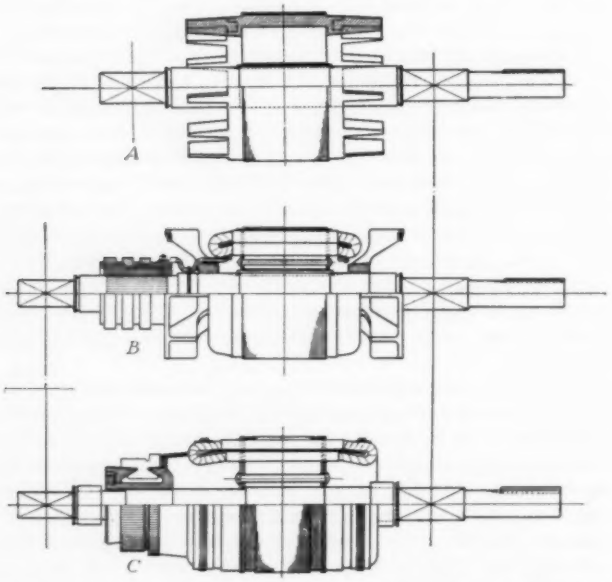


FIG. 2 COMPARISON OF CONSTRUCTIONS OF ROTATING ELEMENTS OF THREE TYPES OF ELECTRIC MOTORS

(*A*, squirrel-cage rotor; *B*, alternating-current wound rotor; *C*, direct-current armature.)

hand, the direct-current motor requires no lamination of the material forming the stationary part of its magnetic circuit due to the fact that in this part the magnetic lines of force are unidirectional rather than alternating, but the diameter of the part carrying the magnetic flux must be relatively larger than in the case of the induction motor due to the salient pole construction required for most efficient design. Instead, then, of a separate laminated core carried in a frame structure as in the case of the induction motor, the frame structure of a direct-current motor must perform the additional functions of composing a large portion of the stationary part of the magnetic circuit and of supporting the pole pieces and the associated stationary windings which are employed in exciting the magnetic circuit of the machine.

The problem is that of suiting these essentially different sets of active elements with accessory mechanical parts necessary to form a motor unit which can be applied to a job. Fig. 4 shows the stationary and rotating parts of the three types being considered in assembled relation. At the right the journal portions of the

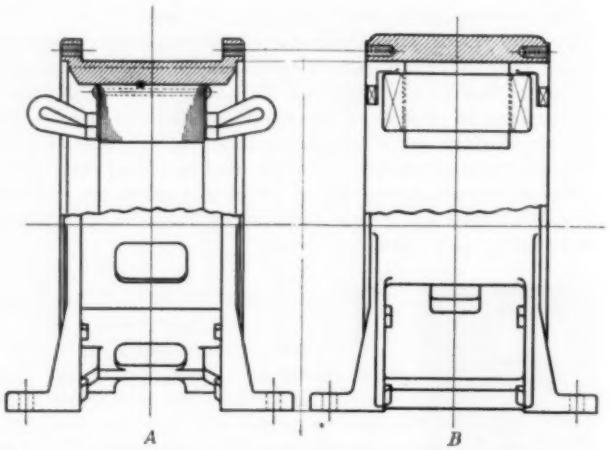


FIG. 3 COMPARISON OF ALTERNATING-CURRENT AND DIRECT-CURRENT FRAME STRUCTURES

(*A*, frame for squirrel-cage and wound-rotor alternating-current motors; *B*, frame for direct-current motors.)

three shafts are in identically the same position from the center line of the machines. Also, the machined ends of both types of frames are identical in position and mounting detail. To support the rotating members from the several frames at this end, it is possible to design an interchangeable bracket structure, fitting and bolted to the frame ends and supporting at its outer end a bearing adapted to fit the journal portion of the shafts.

In a similar way a bracket structure may be designed for the opposite end to serve both the wound-rotor induction motor and the direct-current motor, except that allowance must be made for the different types of brush rigging required for the collector and the commutator. The squirrel-cage induction motor, being symmetrical about the center line, will use the same bracket structure at both ends. Thus, for all motors of a given size, only two distinct bearing brackets are required to serve every electrical type, resulting in maximum activity and minimum cost for this important accessory part.

Electric motors differ not only in electrical type but in mechanical type. Differences in mechanical type are occasioned by requirements of the service. The horizontal, open, ventilated type is the most widely used, but there are many others, differing in the relation of the shaft to the horizontal plane, the measure and method of enclosure, and the method of attachment to the supporting structure. Hence there are both horizontal and vertical motors, open, totally closed, enclosed-ventilated, force-ventilated, self-ventilated, explosion-proof, watertight, floor-mounted, end-mounted, built-in, overhung, etc. In addition, there are variations due to type of bearing and to facilities for mounting accessories such as brakes.

Fig. 5 shows a longitudinal sectional assembly of a horizontal open, sealed-sleeve bearing, squirrel-cage, induction motor. By removing the end brackets (which are shown also in dotted outline in Fig. 6) and replacing them and the shaft with similar parts adapted to ball bearings, as indicated in full lines in Fig. 6, the motor becomes of the ball-bearing type without change in mounting dimensions or overall length.

Referring next to Fig. 7, it will be clear that replacement of the brackets and shaft just shown and the addition of other accessories results in an enclosed-ventilated motor in which the windings are completely protected from destructive elements

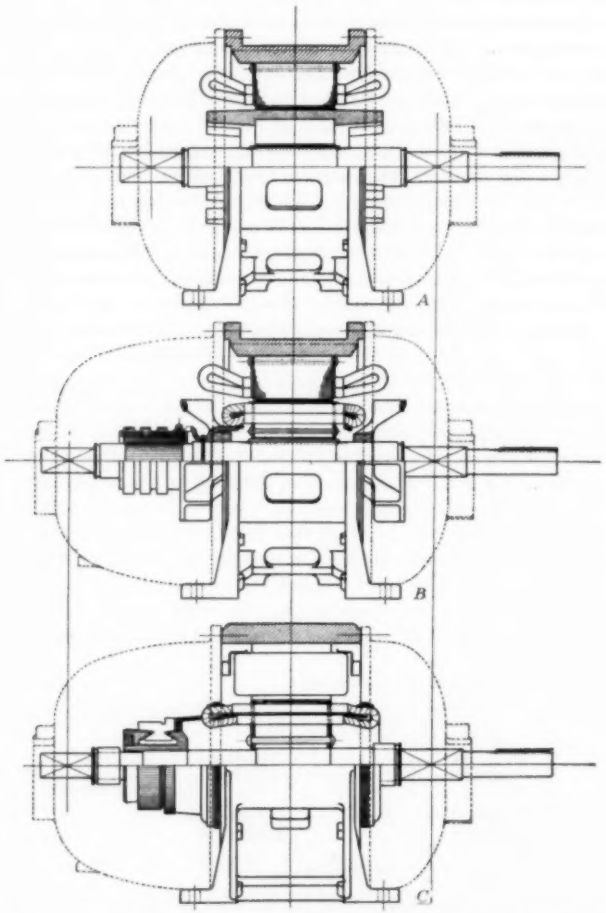


FIG. 4 COMPARISON OF ROTOR AND STATOR ASSEMBLIES OF DIFFERENT TYPES OF MOTORS

(*A*, squirrel-cage induction motor; *B*, wound-rotor induction motor; *C*, direct-current motor.)

permeating the surrounding atmosphere, yet ventilation is effected by blowing air on the exposed portions of the enclosure. Again, the transformation is accomplished without change in mounting or overall dimensions and by using the same frame and electrical parts as in the original, open, horizontal, sleeve-bearing motor.

In a similar way other types can be produced, so that it is possible from a limited number of parts to build any one of many different types or to convert any type into any other type at minimum cost.

Many of the variations in type are accomplished by closure or partial closure of the handhole openings in the end brackets of the open motor, either with or without gaskets. Attention is directed to the end-bracket design in which plane surfaces are invariably employed for engaging the covers and their gaskets. Fig. 8 shows a drip-proof motor. In this, a louvered frame cover is used, while flat steel plates and sheet-cork gaskets effectively close the upper handholes of the end brackets.

In the mechanical variations which must be used, some of which have been described in the foregoing, it is desirable to use identically the same parts as far as possible. If, however, it is not possible to use exactly the same parts, it is still a great advantage in manufacture if parts which must be different from others in some respects can be similar in form and size so that the same tools can be used for casting and machining with as little change in adjustment as possible. These advantages have been secured to a very large extent in the design of the parts forming the combinations described in the preceding paragraphs.

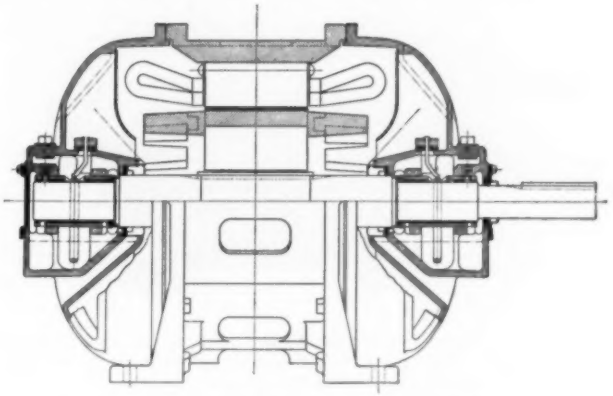


FIG. 5 LONGITUDINAL SECTIONAL ASSEMBLY OF OPEN SEALED-SLEEVE SQUIRREL-CAGE INDUCTION MOTOR COMPLETE WITH END BRACKETS

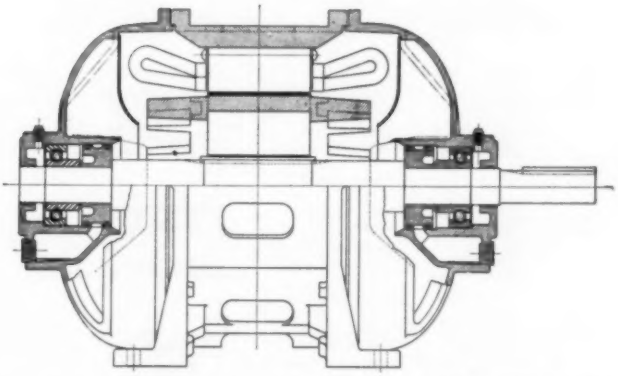


FIG. 6 LONGITUDINAL SECTIONAL ASSEMBLY OF SQUIRREL-CAGE INDUCTION MOTOR SHOWING CONVERSION OF SLEEVE-BEARING MOTOR TO THE BALL-BEARING TYPE

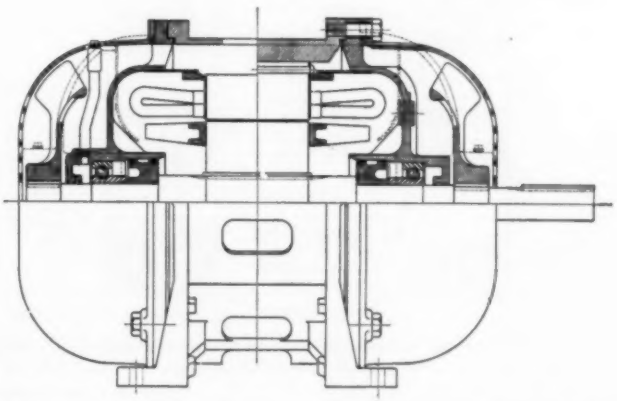


FIG. 7 LONGITUDINAL SECTIONAL ASSEMBLY OF SQUIRREL-CAGE INDUCTION MOTOR SHOWING CONVERSION OF OPEN-BALL-BEARING MOTOR TO THE ENCLOSED VENTILATED TYPE

Of recent years ball and roller bearings have gained favor in

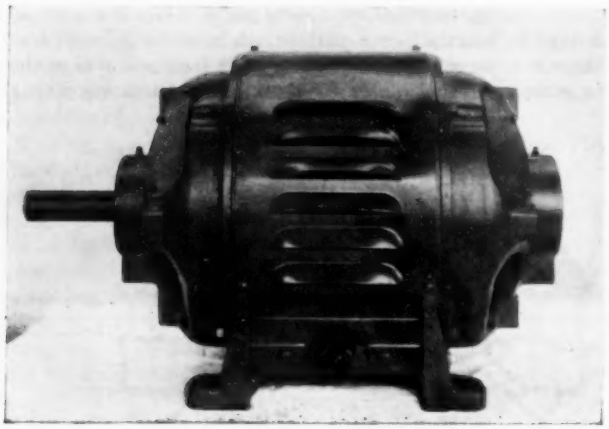


FIG. 8 DRIP-PROOF MOTOR SHOWING SIMPLICITY OF END-BRACKET COVER DESIGN RESULTING FROM PLANE-SURFACE BORDER OF HAND-HOLE OPENINGS

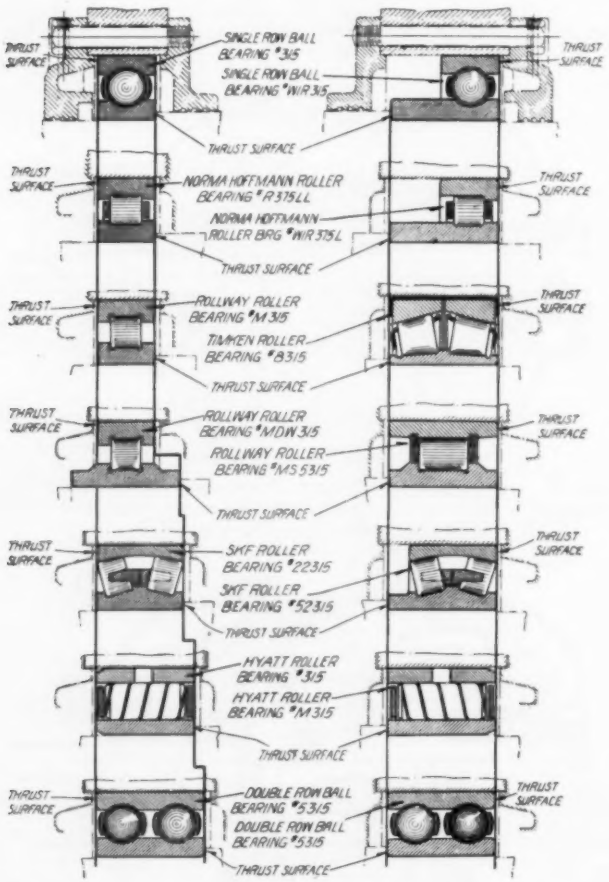


FIG. 9 NON-INTERCHANGEABLE AND INTERCHANGEABLE BALL- AND ROLLER-BEARING DESIGNS

some quarters for motor application. The great variety of types and makes of these bearings, each with its following of proponents among users, has introduced many a problem in their application to motors, and has tended toward diversity of mounting, involving unnecessary expense and delay in delivery of the apparatus to the user. In the design of the line of motors under discussion, the problem of standardization of ball- and roller-bearing housings has been given careful attention.

Ball bearings are made in single and double width by almost all manufacturers. One manufacturer has an intermediate standard. Roller bearings are made to all three standards by various manufacturers, and some differ only a small amount from the standard. Others are made in inch dimensions, whereas most ball and roller bearings have their outside diameters and bores in millimeters. This great diversity in dimensions is one of the factors in making an interchangeable mounting difficult of achievement. A number of bearings of dissimilar width are illustrated in the left-hand column of Fig. 9. The right-hand column shows the various makes of bearings reduced to interchangeability. In this arrangement, the width of the double-row ball bearing is adopted as standard. Single-row bearings are provided with an inner race extended at one side to give a total width equal to that of the double-row bearing. All makes of bearing can be obtained in the double-row width, thus making it relatively easy to serve customers with their preferred brand.

The greatest bearing load occurs at the shaft extension end of the motor, the load at the opposite end being relatively light. To suit this disparity in loading, it has been customary to use a large bearing at the shaft extension end and a small one at the opposite end. This, however, results in two different end brackets for a squirrel-cage induction motor which is otherwise symmetrical. In the new design, this difficulty is overcome and at the same time the bearing capacity is suited to the load by employing the arrangement illustrated in the upper half of Fig. 10. Here a large capacity double-row bearing is used at the pinion end and a relatively lower capacity interchangeable single-row bearing with wide inner race is used where the load is relatively light. The relative simplicity of the new design as compared with

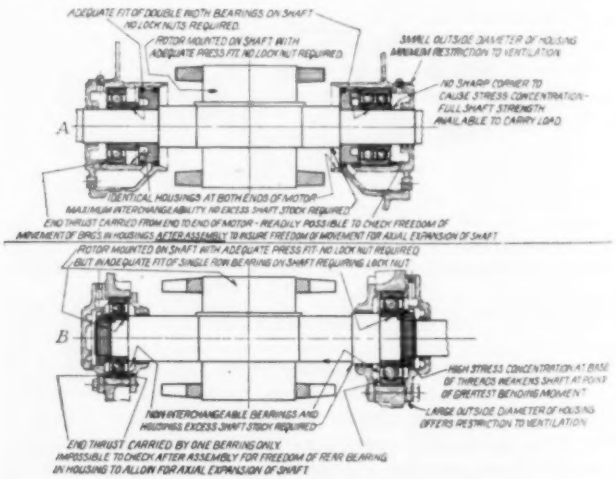


FIG. 10 COMPARISON OF NEW AND OLD DESIGN OF BALL-BEARING MOUNTING IN GENERAL-PURPOSE MOTORS
(*A*, new design, maximum simplicity; *B*, old design, relative complexity.)

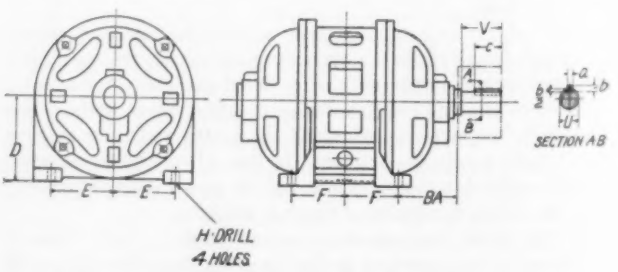


FIG. 11 MOUNTING DIMENSIONS FOR HORIZONTAL MOTORS

the old will be evident by comparison of upper and lower views of Fig. 10.

As ball bearings have increased in capacity due to progress in the art, the contact with the shaft of the inner race of single-row bearings has become insufficient in some of the severe service conditions encountered. Increasing the width of shaft contact by extending the inner race is a means of keeping pace with the advances made in internal design and manufacture of the bearings, without the questionable assistance of locking devices such as appear in the lower view of Fig. 10. It will therefore be seen that the same arrangement which results in the highest degree of uniformity and interchangeability among detail motor parts is ideal in other respects as well.

Important as is the internal design of electric motors, it is no less important that the attachment of the motor to the foundation or driven machine receive careful consideration. Interchangeability from this standpoint has suffered a measure of neglect in the past, and is a feature that recently has been the subject of much profitable discussion.

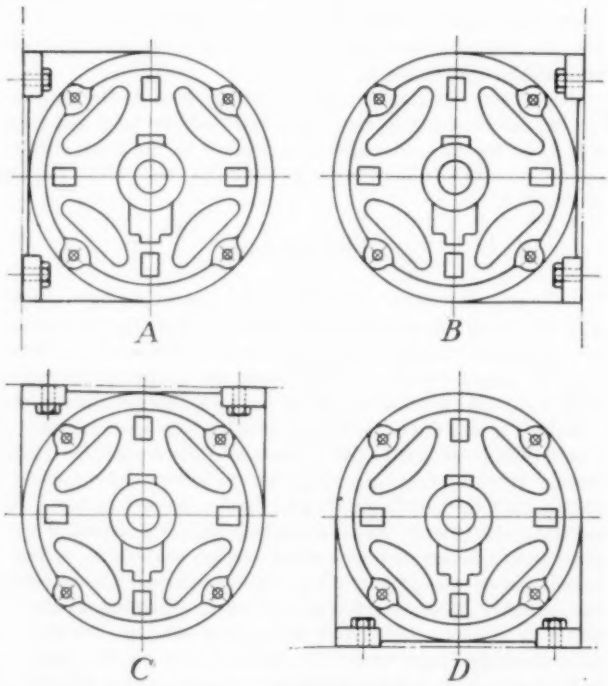


FIG. 12 ILLUSTRATING FLOOR, WALL, AND CEILING MOUNTING FOR ELECTRIC MOTORS

(*A*, left-hand wall mounting; *B*, right-hand wall mounting; *C*, ceiling mounting; *D*, floor mounting.)

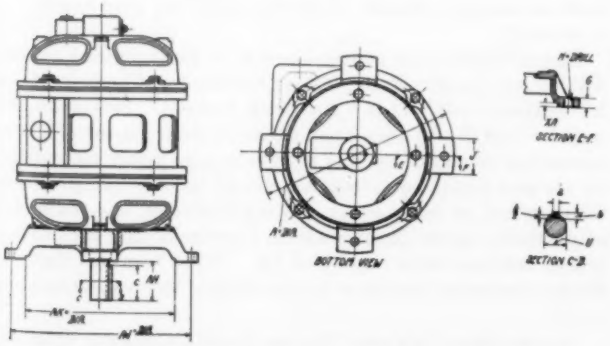


FIG. 13 MOUNTING DIMENSIONS FOR VERTICAL MOTORS

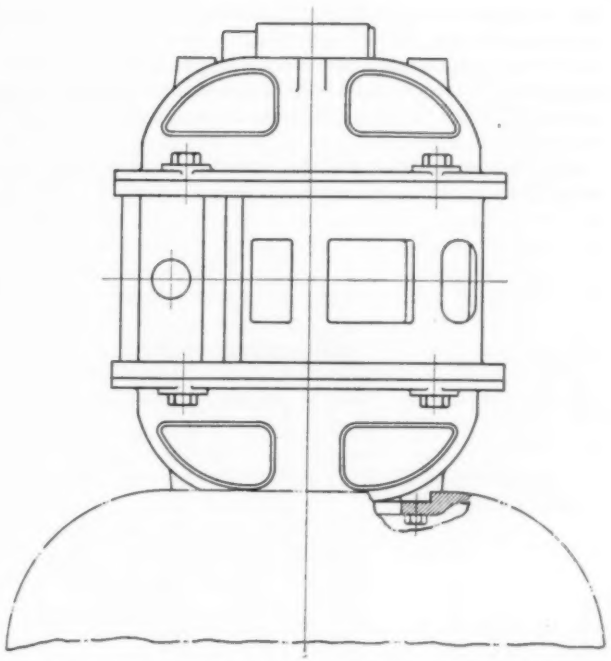


FIG. 14 INTIMATE END MOUNTING FOR HORIZONTAL OR VERTICAL MOTORS

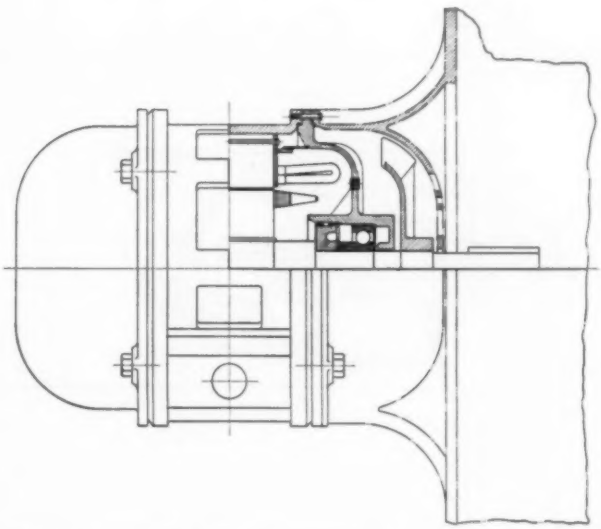


FIG. 15 INTIMATE END MOUNTING OF ENCLOSED VENTILATED MOTOR WHERE DRIVEN MACHINE FORMS REAR BLOWER CASING

The important dimensions affecting interchangeability of standard horizontal motors are the shaft and keyway dimensions, the distance from the face line of the pinion or pulley to the first pair of holding-down bolts, the distance between the first and second pairs of holding-down bolts, the distance between holding-down bolts at right angles to the shaft, the size of holding-down bolts, and the height of the shaft center line above the bottom of the motor feet. These dimensions are illustrated in Fig. 11.

Motors having this mounting arrangement can be attached to a horizontal support below the motor or to the wall or ceiling with the shaft in a horizontal position, and can often be arranged for mounting in intermediate positions as indicated in Fig. 12. Oil-lubricated motors may be tilted so that the shaft makes an angle of 10 or 15 deg. with the horizontal. For greater inclina-

tion up to the vertical position, grease-lubricated motors are the common solution. However, for strictly vertical operation with inclination from the vertical not exceeding 15 deg., it is possible to obtain motors having specially designed oil-lubricated bearing assemblies. Fig. 13 shows the important mounting dimensions for vertical motors.

It often happens that it is desirable to mount motors either horizontally or vertically in more intimate association with the

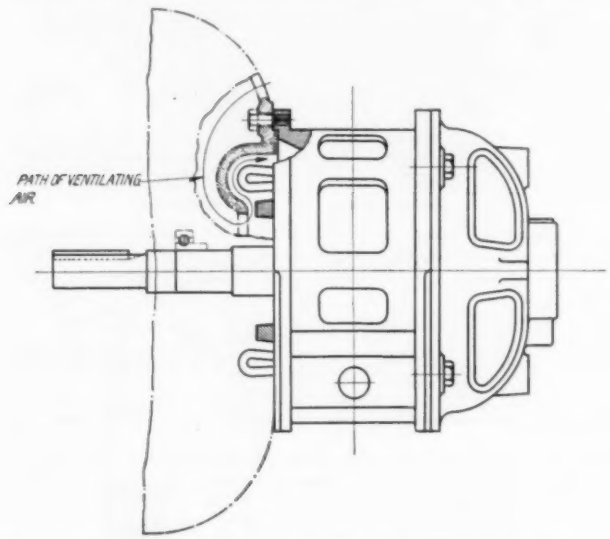


FIG. 16 INTIMATE END MOUNTING OF ELECTRIC MOTOR WHERE MOTOR-DRIVEN MACHINE PROVIDES REAR BEARING AND END GUIDES

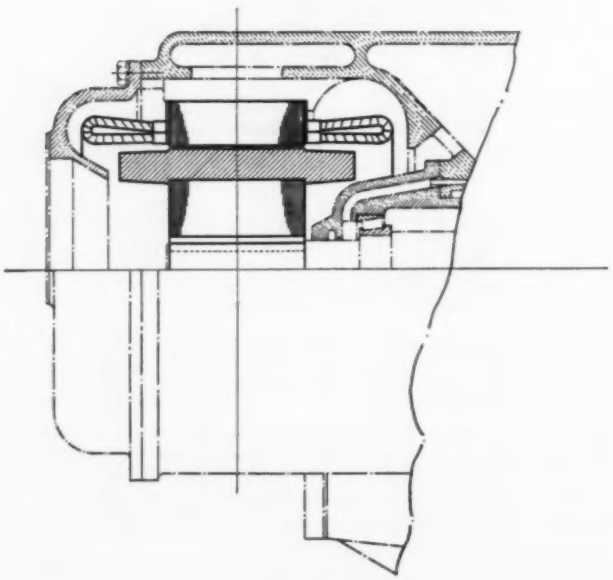


FIG. 17

driven machine than is possible using the more widely used standard mountings. Fig. 14 shows how the driven machine may be substituted for the vertical base so that bosses on the end brackets are employed directly as the means of attachment. It will be readily seen that this provision incorporated in the brackets introduces no interference with machining operations arranged for brackets where this is not required. Indeed, the presence of these bosses in standard foot-mounted horizontal motors is in no way detrimental and may be useful in mounting brakes, gear guards, or other accessories.

Fig. 15 shows an intimate mounting of an enclosed-ventilated motor in which a portion of the driven machine becomes the substitute for the fan or blower casing. The motor is in this case supported through the bracket to the frame ring, and a very neat and compact arrangement is obtained.

In the case of open motors, the driven machine is sometimes formed to take the place of the end bracket, winding clearances and air guides being provided as required. In such cases a so-called "three-quarters" motor is provided, as illustrated in Fig. 16.

In some cases it is expedient, where the shaft of the driven machine is relatively large and rigid, to overhang the motor rotor on an extension of such a shaft without an outboard bearing. The wound stator core may be supported in an extension of the frame of the driven machine, with the net result of great compactness and simplicity. The motor parts furnished in such a case are illustrated in Fig. 17. The wound stator without frame (which is shown in dotted outline), being a unit by itself, can readily be pressed into any frame structure provided, as long as it has the requisite machined fits, clearances for windings, and passages for ventilating air.

Of necessity, in a short paper, a subject such as this must be treated rather sketchily, but it is hoped that this brief discussion will serve to show the efficiency possible in designing electric motors to fulfil in widely varying conditions their simple mission of "service."

Discussion

GEORGE A. BOUVIER.[2] Evidently Mr. Brown has some quarrel with the term "flange-mounted motor." Possibly he is reluctant to identify the types of motors generally designated as flange type for sound reasons; nevertheless, the writer is inclined to believe that the general use of the term justifies its continuance.

Industry, as a whole, is wont to look upon an electric motor as a unit machine in itself, distinct and separate from other equipment. It is true that this is so when the motor is mounted on a ceiling, wall, or floor for driving machinery, lineshafts, etc. The electric motor in such cases is a machine unit used instead of some other power device such as a steam engine, gas engine, or water wheel because of its greater utility or convenience. The machine-tool industry 15 years ago was justified in regarding an electric motor in this way. It is now the general practice, however, in approximately 75 per cent of the cases to drive machine tools by individual motors, but unfortunately the practice of incorporating motors as component parts of the structure is not carried out in as many cases as is practicable. This is also true as to fuse panels, switches, starting equipment, wiring, etc., with the result that a machine, otherwise an engineering achievement to be proud of, looks as though a number of afterthoughts had been hastily incorporated.

Unquestionably the standardization of foot-mounted motors will aid considerably in the further application of built-in drives for machine tools and is a long step forward. Personally the writer would like to see a more intensive development of a corresponding standard covering the flange-type motor mountings. As has been pointed out, these are already used by foreign manufacturers and on many of our own machine tools, and the writer emphatically agrees that it results in a neater looking job and impresses one as a better engineered job. It has been said that the flange-type motor costs more because there is only a small demand

[2] Development Engineer, Western Electric Company, Chicago, Ill.

for them. On the other hand it seems that the demand is influenced by the cost. However, the increased cost, relative to the cost of the general class of machines on which the flange motor is installed, is practically negligible and should not unduly influence any but price buyers. It is the matter of delivery time that is particularly objectionable, and the writer has often been forced, because of the time involved, to use a foot-mounted motor when he would have preferred, for engineering reasons, to use a flange-type motor.

It is remembered, when electrical starters and generators were first used on automobiles, that most of them were equipped with feet, which at that time were most convenient to use because of the tacked-on nature of the first installations. The quickness and skill with which the automobile industry incorporated the starter and generating units as built-in parts of the motor can very well be followed profitably by the machine-tool industry. The adoption of the flange, or what amounts to the equivalent of a flange for these installations, is good evidence of the value of this mounting, particularly for types of machines such as machine tools where the drive should be an integral part of the machine.

The writer would like to see more built-in units—built-in not as attachments or accessories to a machine, but designed to be just as much an integral part of the machine as the clutches or gear boxes. He personally would not have much objection to purchasing such a machine from a competent and reliable builder of machine tools. It would be enough to know that the supplier was as much responsible for this element of the machine as for any other. The writer is of course unfamiliar with the facts on which Mr. Brown based his statement that they had to be prepared to furnish motors with many different types of ball or roller bearings to suit the whims of the customer. It is thought that the majority of users could be satisfied if the maker of the motor would give assurance that he is using a ball or roller bearing as satisfactory for the purpose as any other part of his motor.

The automobile industry is one that many older industries can emulate and are emulating to their advantage. No other industry has done so much in the way of standardization, yet this has not restricted it, either, to hide-bound practices of tradition. So by all means standardize motor dimensions, but do it with the thought that machine-tool engineers will regard this unit as requiring no more special consideration or pampered treatment than any other element of their machine.

MSP-53-8

Flange- Versus Foot-Mounted Motors

By HERBERT CHASE,[1] NEW YORK, N. Y.

Advantages in favor of flange mounting given by the author include: improved appearance, compact arrangement, more rigid mounting, saving in weight, less cost, better alignment, more easy attachment, and better protection against oil, chips, and dirt. Full advantage cannot be taken of the benefits of flange mounting until some effort at standardization is made. Motor manufacturers can easily bring about such a standard to their own benefit and to the benefit of machine manufacturers generally, and the latter, it is believed, will readily accept such a standard.



ENGINEERS who give the subject reasonable consideration hardly can fail to arrive at the conclusion that flange-mounted motors present a number of advantages as compared with the ordinary foot-mounted type. It is far from being the author's contention that the foot-mounted motor has no place in industry, but careful analysis of the conditions involving motor applications to machine tools and to other forms of industrial machines brings out at least eight advantages in favor of the flange-mounted type, whereas the disadvantages, in cases where they are applicable at all, are relatively few. Chief among these is a lack of availability from stock, a drawback which would be overcome quickly, of course, once the flange-mounted type is standardized and when it has found more extensive use.

It is not the contention that the base-mounted or foot-mounted motor should be displaced where it possesses any real advantage over the flange-mounted type, but rather that the subject of flange mounting should be given more consideration than has been accorded to it in the past.

The advantages in favor of flange mounting, as they already have been presented elsewhere[2] by the author, include the following:

1 Improved appearance of the finished machine. The motor looks as if designed as an integral part of the machine, not as if added as an afterthought.

2 Compact arrangement often reduces the floor space required.

3 A more rigid mounting. Overhanging brackets are avoided.

4 A considerable saving in combined weight of machine and motor.

5 Less expense, since a separate bracket is not required.

6 Correct alignment is readily secured, especially if the flange of the motor or the machine flange is recessed to pilot the mating member.

7 Attachment is more easily made, without shimming or other adjustment.

8 Often better protection is afforded against oil, chips, and dirt.

As against these advantages, not all of which, of course, are likely to be realized in any one design, there is the disadvantage of relative unavailability. To date, the mounting flange has not been standardized, and ordinarily flanged motors are not carried in stock. Consequently, replacements, in the event of motor trouble, are somewhat less readily made than they would be with a base-mounted or foot-mounted motor.

In this connection, however, it must be admitted that the foot-mounted motor is not completely standardized. Certain proposed standards for certain types of foot-mounted motors have been adopted, and motors made to these standards are shortly to become available commercially, but this standardization extends so far to include only squirrel-cage induction types.

It is doubtful whether any two makers supply motors for flange mounting with the flange itself and its position in respect to the shaft and other parts of the motor the same. This means, of course, that a different connecting flange must be provided by the machine manufacturer if the machine purchaser is given an option as to make of motors, but a similar diversity exists also in respect to foot-mounted motors, as a variety of brackets is necessary to adapt these motors to the same make of machine. In many cases, also, the same make of motor is not interchangeable as between a.-c. and d.-c. types, but at least one company (Westinghouse) has standardized, so far as its own practice is concerned, a considerable group of sizes of flange-mounted motors in which both a.-c. and d.-c. types are interchangeable within certain limits. It has been found easier, in fact, to effect interchangeability with the flange-mounted type than with the foot-mounted type.

Generally speaking, a smaller and less expensive pattern is necessary to adapt a flange-mounted motor to a machine than is required to adapt a bracket for accommodating a foot-mounted motor.

Today, motors for flange mounting usually cost somewhat more than the foot-mounted type, not because they are inherently more expensive to make, but because they are regarded as special and are made in smaller quantity. Many makers who build machines of various kinds in large quantities have adopted their own standard on flange-mounted types of motors in order to gain one or more of the eight advantages named as applying to this type.

To establish further the case in favor of the flange-mounted motor, the advantages already enumerated will bear some elaboration.

That improved appearance is realized will hardly be questioned by those who compare the accompanying illustrations of flange-mounted motors with motors mounted in other ways, especially on the ordinary pedestal or base. Compactness also is evident from an examination of the designs here illustrated. It is easy to set the flange-mounted motor into the base of many machines and in some cases to house it entirely within the base. While this is true also with other types of mountings, in certain instances, the same degree of compactness is rarely obtained, except perhaps with the barrel-mounted types widely used in woodworking machinery.

Reduction in floor space occupied is obtained in some cases by using a vertically mounted motor placed above the machine. In the opinion of some engineers, a flange is the logical form of con-

---

[1] Associate Editor, *Product Engineering* and *American Machinist*.

[2] "Flange- or Foot-Mounted Motors?" *Product Engineering*, Feb., 1930, p. 84.

Presented at the Fourth National Machine-Shop Practice Meeting, Chicago, Ill., September 22 to 24, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

nection when a vertical motor is to be employed, but to the author it seems as illogical to employ a vertical bracket to mount a foot-type motor as to use a horizontal bracket when the same motor is horizontally mounted. In other words, it is equally logical to use the flange whether the motor is placed vertically or horizontally.

In many installations of foot-mounted motors, overhanging brackets are required, or seem to present the best mounting arrangement for that type. Substitution of a flange-mounted motor under similar circumstances usually will provide a more rigid mounting without the same degree of overhanging. Examination of accompanying illustrations will make this point clear. For similar reasons there is a marked saving in weight with the flange mounting because the heavy casting for the bracket is eliminated.

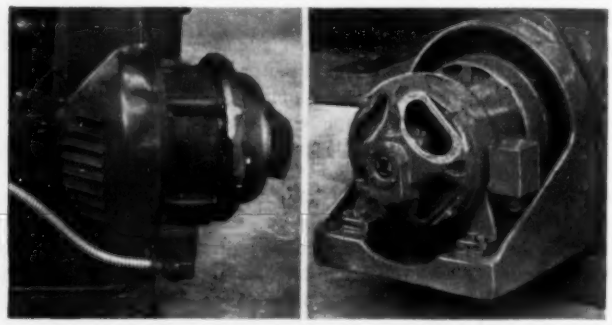


FIG. 1 MOTORS ON GOULD & EBERHARDT SHAPER.
(At the left, the flange-mounted motor; and at the right, a foot-type motor with special mounting bracket. Obviously the former is the neater and less expensive arrangement.)

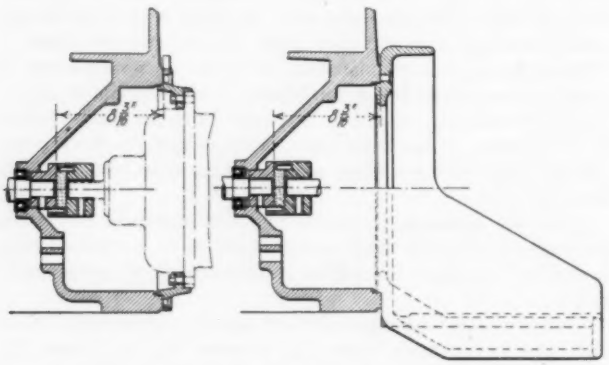


FIG. 2 SECTIONAL VIEWS OF THE TWO MOUNTINGS ILLUSTRATED IN FIG. 1

On the same score, the cost of combined motor and mounting is less in the case of the flange-mounted type, assuming that an extra charge is not added for the flange motor merely because it is classified as a non-standard type. By countersinking the flange into the machine, or by providing on the flange or on the machine a centering boss which is recessed into the mating member, perfect alignment can be assured without any shim, a decided advantage compared to the average arrangement with foot-mounted motors. No shims are required to adjust the shaft height, and the end of the shaft remains in fixed relation to the flange itself.

Better protection against grease, dirt, and chips is likely to be afforded with the flange-type motor, especially when it is placed at the top of the machine, as is done in many installations. It does not follow, of course, that a similar advantage cannot be secured with the foot-mounted type, but in general it is less likely, partly because the foot-mounted motor frequently is mounted on a base extension closer to the floor, where more dirt, oil, and chips fall upon it.

Among machine-tool manufacturers there are already a large number of users of flange-type motors, despite the lack of standardization. Many more certainly would be used if standardization had been effected and the price of the motor was not higher than for the foot-mounted type. Even so, the savings in a flange coupling as compared to a bracket sometimes more than offsets the difference in cost of the motor itself, and brings about a net saving in weight of the combined units.

Ray Millholland, of the Millholland Sales and Engineering Company, is among the designers who have employed large numbers of flange-mounted motors. He mentions the following points as being highly desirable in designs using such motors: Proper protection against dust and chips and provisions for proper cooling. Accurate balance of rotating parts to prevent the transmission of vibration to the frame and tools of the machine. Minimum overhang of the motor. In such machines as the Millholland Company builds it is necessary, of course, to have a motor of ample power, even though the tool used may be dull. Much the same observations might be made concerning foot-mounted motors. It is, for example, just as undesirable to have a bracket with excessive overhang as to have a motor which overhangs more than necessary.

The illustrations (Figs. 1 and 2) of motors mounted on a Gould & Eberhardt shaper afford an excellent opportunity for direct comparison between flange and foot-mounting, as this machine is supplied in either way. W. F. Zimmerman, chief engineer of the company, points out that the flange type of mounting is preferable from the standpoint of attaching, but says that, unfortunately, all motors cannot be mounted in this way because there is no part of the motor housing that can be machined so as to centralize the motor and at the same time secure it to the attaching plate. Presumably this refers to motors with standard frames but without feet. It certainly does not apply to a motor designed for flange mounting.

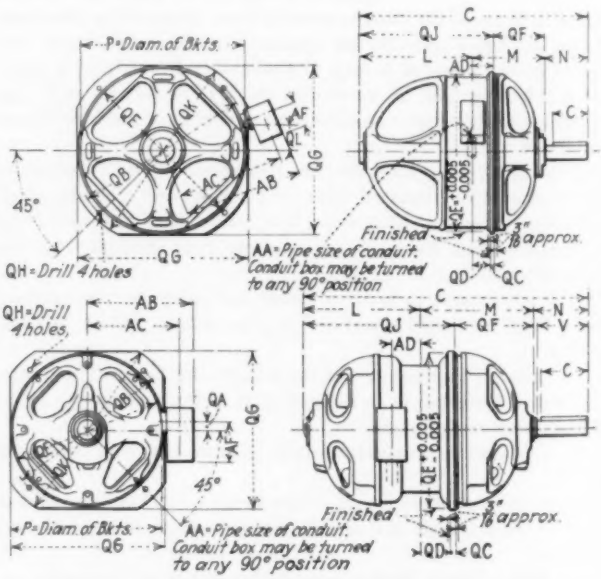


FIG. 3 OUTLINE DRAWINGS OF MOUNTING AND OTHER DIMENSIONS
(These are the interchangeable d.-c. and a.-c. flanged motors standardized by Westinghouse.)

"The flange type is the neater mounting," continues Mr. Zimmerman, "and by centering some portion of the motor frame

there is very little difficulty in attaching as compared with the foot-type mounting."

A flexible coupling is employed to connect the motor shaft and the driven shaft of the shaper. In the same machine, when it is fitted for pulley drive, the driven shaft is continued across the housing to a separate flange where a Timken bearing is employed. This arrangement appears to meet an objection raised by another engineer to the effect that a machine designed for use with a flanged motor is not readily adapted to a pulley drive. This may be true in some cases, but the substitution of a flange containing a bearing in place of the flange of the motor seems to afford a ready solution of this particular problem under certain conditions.

Contrary to some assumptions, the flange-type motor frequently lends itself readily to a belt or chain drive or to the use of gear reduction. It is not a difficult matter to secure almost any reduction desired with either flange type or foot-mounted type of motors. Neither is it difficult to provide for belt or chain adjustment, if this is required, either by turning the flange-mounted motor about one of the flange bolts as a center or by sliding it bodily with its flange mounting in suitable ways. In the case of the gear burnisher made by the Cincinnati Shaper Company, a compact arrangement has been effected by removing one of the standard motor end brackets and substituting a flange with a recess adapted to the particular special bearing required. This works out nicely, but naturally requires a different flange casting for different makes of motors.

According to C. E. L. Dapprich, general engineer of the Westinghouse Electric and Manufacturing Company, interchangeability as between a.-c. and d.-c. types of motors is readily accomplished by the use or flange mounting. This is borne out by the practice of the Westinghouse company, which has standardized a ball-bearing and sleeve-bearing series of flange-mounted motors arranged in four groups covering a wide range of power. Each frame in each a.-c. group of this make is interchangeable in mounting dimensions with every other frame in the same group, and also with the frames of motors in the corresponding d.-c. group. The various dimensions involved in this standardization are shown in an accompanying illustration (Fig. 3).

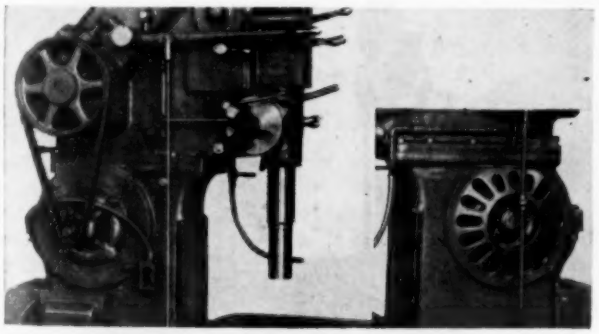


FIG. 4 TWO VIEWS OF A BARNES DRILL WITH FLANGED MOTOR MOUNTED INSIDE THE COLUMN
(Openings for ventilation are shown.)

That it is an easy matter to provide complete enclosure of a flange-mounted motor, as well as proper ventilation, is indicated by the illustration (Fig. 4) of the Barnes drill. In this instance, if the machine is ordered for belt drive instead of for motor drive, a plain column is substituted for that shown. Another illustration of the use of a flange-mounted motor that is built into a machine is that of the Pratt & Whitney jig borer. In this case a vertical motor is employed, the lower end frame being a part of the housing of the head of this machine.

In some respects, European machine-tool manufacturers have gone farther in the use of flange-mounted motors than have American manufacturers. This applies in particular to their use on radial drills, in which considerable overhung weight is added to the arm by placing the motor on the arm and close to the point at which the power is to be applied. An alternative construction used in the Cincinnati-Bickford radial drill is shown in Fig. 5. In this case a flange-mounted motor is employed, but it is placed on the opposite side of the column from the arm, and thus tends to counterbalance rather than to add to the weight of this arm. This same radial drill is furnished, when desired, with a foot-mounted motor, but in that case a heavy and rather expensive bracket is required to hold the motor.

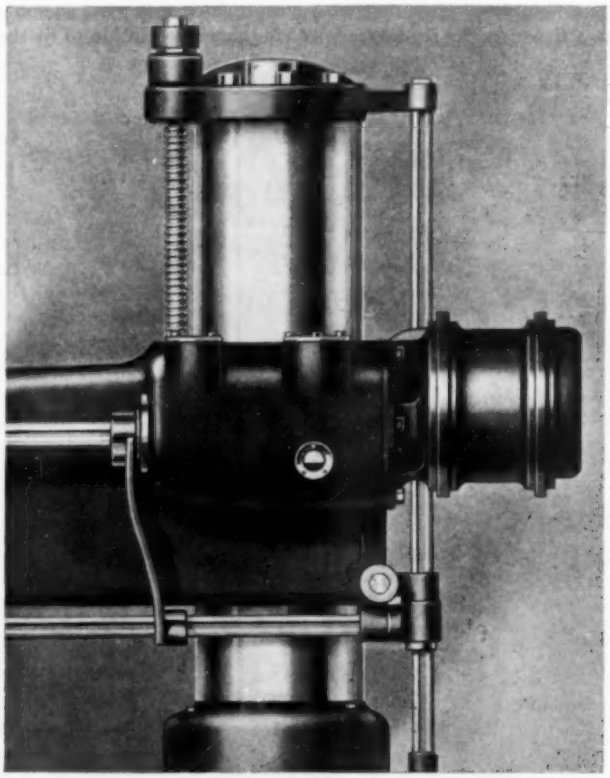


FIG. 5 FLANGED MOTOR ON CINCINNATI-BICKFORD RADIAL DRILL
(Considerable weight is saved and the arm is partly counterbalanced by this location.)

Several manufacturers of machine tools incorporating drilling, milling, and tapping units today are employing flange-mounted motors driving direct to the spindle. In some cases the unit is arranged to be swung through a considerable angle. Among the advantages of the flange-type motor for this type of application is that it gives a compact and self-contained unit. In some such drills built by the Kingsbury Machine Tool Corporation three or more flange-type motors are employed (Fig. 6), and in another arrangement the same motor is used with a belt drive, in which an idler is employed to facilitate speed changes. A small pulley at the lower end of the same motor drives the coolant pump.

Various engineers have pointed out that considerable care is necessary when flange motors are employed to avoid unbalance which produces vibration, not only in the motor itself, but in the machine to which it is attached. Undoubtedly this is true, but it is equally true that many manufacturers require considerable care in balancing motors of any type that are mounted on the

machine, for otherwise the motors are likely to be noisy and to transmit vibration.

A rather unusual form of flange-mounted motor is that employed by the Porter Cable Company in its horizontal disk grinder. In this case the motor is built in as part of the column of the machine, being placed between the grinding wheel and the column.

Another and somewhat similar vertical arrangement of the motor is used in the Gorton milling machine illustrated in Fig. 7. In this case, however, the motor is placed on top of the machine and drives the head through a four-step grooved pulley and belt. The motor itself is not flanged, but is provided with what amounts to a flange in the form of a split ring machined inside to fit the

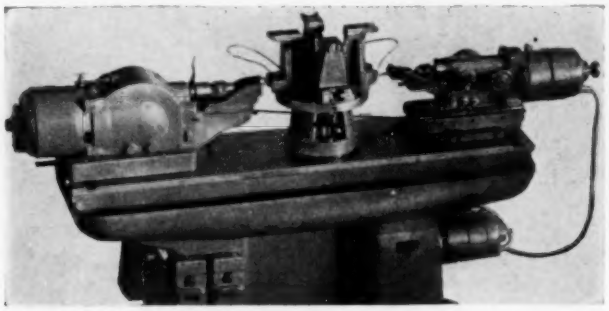


FIG. 6 KINGSBURY DRILL USING THREE FLANGED MOTORS

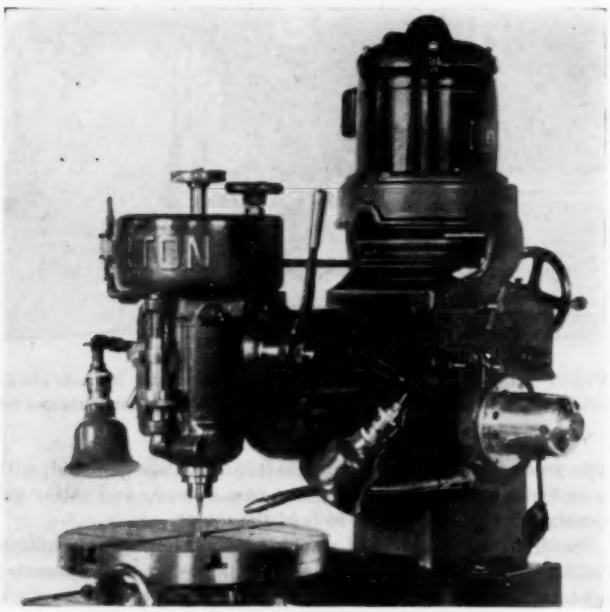


FIG. 7 VERTICAL MOTOR MOUNTING ON GORTON MILLER
(This incorporates two flange-mounted motors.)

motor housing and outside to fit into a recess on the supporting bracket. This bracket forms part of a slide which can be moved back and forth to tighten the belt. The same machine employs a small flanged blower motor driving an impeller which provides a current of air for blowing chips off the work.

A number of manufacturers of tools in England and Germany employ motors with flange mountings for much the same reason that the same type of motor is being used in this country. Some of the illustrations given (Figs. 8 and 9) show slight differences in practice, but basically the designs are similar to those in American tools.

Although most of the illustrations apply to machine tools, it need hardly be said that similar applications of flange motors are easily made to machines of other types. Many are now used extensively on small machines built in large production, such as vacuum cleaners, pumps, and gear reducers. Naturally these

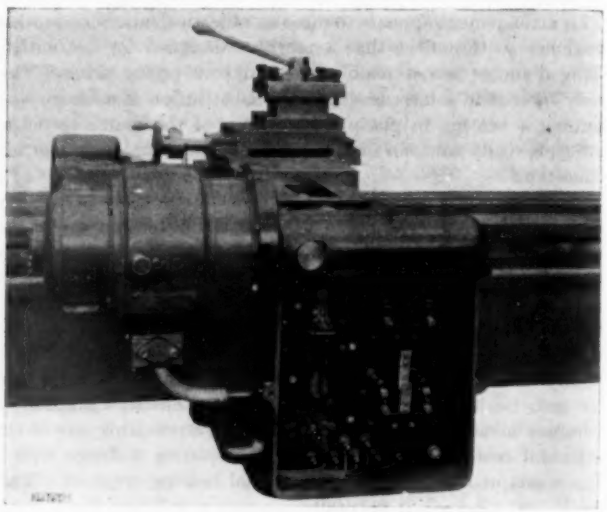


FIG. 8 FLANGED MOTOR ON DEAN, SMITH & GRACE LATHE
(This is used to traverse the carriage on a 25-ft. bed.)

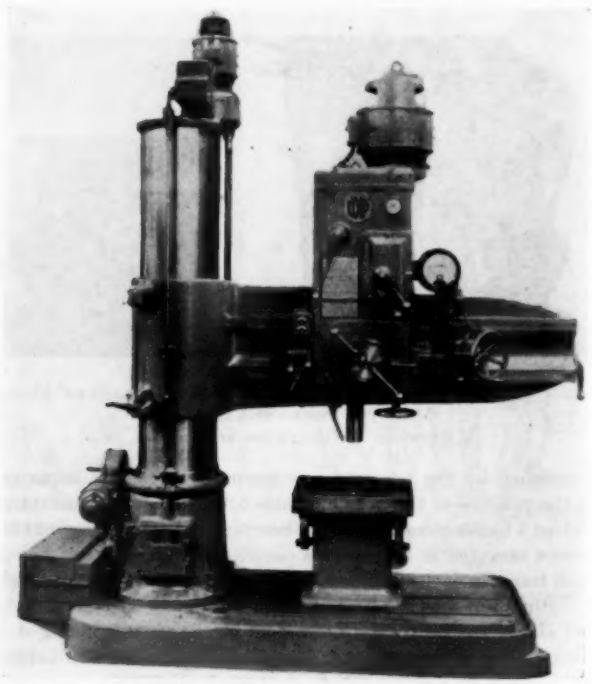


FIG. 9 GERMAN KOLB RADIAL DRILL
(This has three motors of the flanged type.)

applications differ in detail from those described, but they involve similar conditions and advantages.

In cases where the machine manufacturer requires very large numbers of identical motors he can purchase these with flange mounting when desired. However, each purchaser designs his own "standard" flange, except in the case of the Westinghouse

company, already cited, and in this case the standard refers primarily to motors of quite large size.

Naturally, full advantage cannot be taken of the benefits of flange mounting until some effort at standardization is made. Just so long as machine-tool and other machine manufacturers are willing to put up with the unnecessary expense of special mountings, so long will somebody pay the necessary extra expense of special brackets or flanges. If machine manufacturers would get together, however, and cooperate with the motor manufacturers in bringing about motor standardization, including, let it be hoped, a flange-mounted type, both groups would reap considerable advantages, as also would the purchasers of the machines. It is time that this subject of standardized mountings and mounting dimensions for motors be given the attention it evidently demands.

Several motor manufacturers already have their own standards for footless motors. To make such a motor into a flange-mounted type it is necessary only to add a flange to the casting for the drive-end bracket. The dimensions and drilling of such a flange could be standardized easily in such a way as to make it adaptable to any of these motors, without interfering in any way with the design of other portions of the motor than the drive-end bracket. It would remain then only to standardize the drive end of the shaft and the position of the standard flange with reference to the end of this shaft to make all such motors completely interchangeable, at least for all overhung applications.

Motor manufacturers could easily bring about such a standard to their own benefit and to the benefit of machine manufacturers generally. The latter group, it is believed, would readily accept such a standard, thereby relieving themselves and the motor manufacturers of needless delays and expense incident to the production of special flanges.

Discussion

[NOTE: A discussion of this paper jointly with the paper of J. L. Brown by George A. Bouvier will be found with the discussion of the latter paper.]

J. W. HARPER.[3] The question of standardizing a flange-mounted motor, as suggested by the author, already has been given considerable thought by electric-motor manufacturers. At the present time there is before N.E.M.A. a proposal outlining certain dimensions for a flange-mounted motor to be used in connection with the dimensions already standardized for foot-mounted motors.

Five principal dimensions are being considered in this proposed standardization:

(1) Distance from face of flange to shoulder on the shaft (distance from shoulder to end of shaft has already been standardized)
(2) Outside diameter of flange
(3) Diameter of bolt circle in flange
(4) Thickness of flange
(5) Size and number of bolts in flange.

The type of construction proposed would be similar to that shown in Fig. 4 of the paper. By machining both sides of the flange, the motor can be mounted either inside the machine, as shown by Fig. 4, or with the motor outside, and only the pulley end shield inside. Machining the outer periphery of the flange permits it to be used as a rabbet for locating if this should be necessary.

While such a mounting has numerous advantages, there are also several disadvantages. In the first place, if the motor is mounted with the pulley end shield inside the machine, the ventilation on that end is likely to be very much restricted or cut off altogether, unless the machine designer makes provision in his base castings for louvers or the like, as was done in Fig. 1 (left). Another disadvantage is that different flanges are usually required for each different diameter. In order to obtain proper bolt clearances, etc., the bolt circle must be based on the largest diameter frame, so that if the same flange is used for smaller diameters, it becomes abnormally large.

Both these disadvantages can be overcome by still another type of flange where the face of the flange, instead of being located near the end frame of the motor, is located near the nose of the bearing housing. By having the flange thus spaced away from the motor proper, provision can be made by the motor manufacturer for adequate air supply. Furthermore, since the motor diameter does not figure in so prominently, the same flange mounting can be used for more than one diameter of motor.

A disadvantage of type 2 is that the overhang is increased materially over that of type 1. In the latter it is desirable to keep the flange as far from the bearing nose as possible to minimize the overhang. In the former the flange should be spaced away from the bearing nose a distance approximately equal to the thickness of the machine frame. This is to allow mechanical clearance for a chain or belt drive inside the machine if such is used. Type 2 mounting is the one principally used by European manufacturers, as evidenced by the two examples given by the author.

Still another mounting which might be considered of the flange type is where the motor, usually without feet, is furnished less pulley end shield. Under such conditions the end shield is incorporated into the machine design itself, and of course must be furnished by the machine builder to meet his own particular requirement. Usually some special form or feature peculiar to that design is required. An example is where the end shield might be combined with a chain guard or support, or where the end shield would be part of a gear case. Such a construction naturally results in the minimum overhang and most compact arrangement, and for these reasons has found considerable use in this country.

Unquestionably, flange mounting in any of the forms outlined is coming more and more to the front, and its more general use will be hastened by standardization of mountings on the part of the electrical manufacturers.

J. L. BROWN.[4] In commenting on this excellent paper, the writer wishes to offer a few comments which have to do with some of the problems and difficulties encountered in the attempt to make this type of monting practical. These are as follows:

1 Question as to the field for the flange-mounted type of motor.

(*a*) Westinghouse has a line of flange motors available, yet there is no great demand for them, and at the same time there is some demand for other types of end mounting, where an external flange on the motor is not suitable.
(*b*) Machine-tool speeds are usually either higher or lower than available motor speeds requiring some sort of speed reduction drive, rather than direct coupling and rendering end mounting of any kind less advantageous than it otherwise would be.
(*c*) Foot-mounting standardization is in process of development. It may be suggested that after its advantages are fully available and in profitable use, the field for end mounting will be more clearly defined.

2 Type of end mounting to be standardized.

(*a*) If and when end mounting is standardized, the type

[3] General Electric Company, Schenectady, N. Y.

[4] Industrial Motor Engineering Department, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

should be such as to be applicable to the standard motor parts with minimum addition of material and machining operations to avoid penalizing this type of mounting with large additional motor cost

(*b*) The design should, as far as possible, dovetail into already existing designs to avoid adding to the number of types of motor, as this tends to reduce the activity of each type, and this, in turn, increases costs

(*c*) It may be suggested that when a real flange mounting makes the most efficient design, a steel ring may be provided and bolted to the motor end bracket and the driven machine, respectively.

R. E. W. Harrison.[5] The author brings up an interesting subject at an opportune moment, having regard to the efforts

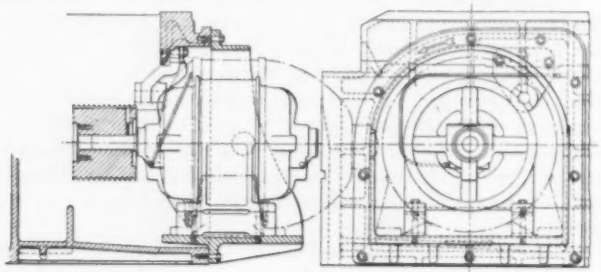


Fig. 10

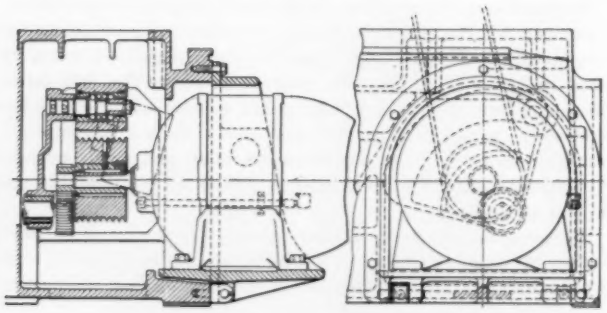


Fig. 11

now being made by the motor manufacturers to accomplish standardization of all those important dimensions of electric motors which affect the type and method of mounting.

In his enthusiasm for the subject, the author, however, has drawn a number of conclusions with which issue can be taken, and based on an experience with close on to one-thousand applications of end- or flange-mounted motors, the following data are offered, taking the list of eight advantages which the author attributes to the flange mounted motor in the order in which they are presented:

1 Improved appearance of the finished machine.

(*a*) It is readily agreed that the flange-mounted motor presents a better appearance than any other type of mounting. However, the disadvantages of lack of flexibility in securing alignment between the driving pulley of the motor and the driven member frequently lead to a condemnation of the flange method of mounting the motor

(*b*) With a flange mounting the inner bearing of a motor is often in a difficult place to reach for inspection, oiling, and eventual replacement

(*c*) The average maintenance foreman and mechanic today frequently expresses the desire for a motor which can be slid into its operating position and then bolted down when the niceties of alignment have been obtained.

2 Compact arrangement often reduces floor space required. While this is true, there is available an alternative method of mounting the motor which gives precisely the same advantages as regards consumption of floor space without many of the inconveniences inseparable from the flange-mounted motor.

3 A more rigid mounting. It is readily granted that a properly designed flange mounting constitutes one of the most rigid forms of mounting which can be devised. However, when this degree of rigidity is secured at the expense of convenience of operation, its merits are likely to take on a diminished importance, having regard to the improvements affected in motor manufacture during the last few years. Such improvements include improved dynamic balance, closer fitted bearings, and improvement in constancy of torque.

4 A considerable saving in combined weight of machine and motor. This saving in weight is apparent only in those cases where a relatively clumsy shovel-type bracket is the alternative method of mounting under consideration.

5 Less expense. Practical experience in both the engineering department and the works and a survey of the final costs indicate that with the facilities for end mounting or flange mounting now available, it is very definitely much more expensive to mount a motor by this method than the alternative foot-mount-method. Numerous patterns have to be kept readily available to accommodate the varying sizes and types of motors of different makes, and these patterns have to be duplicated over and over again to accommodate the peculiarities of the individual motor maker's designs.

6 Correct alignment is readily secured. Agreed, providing that the flange or mounting ring is designed to suit each individual make and size of motor likely to be mounted on any one of a variety of machines.

7 Attachment is more easily made. The consensus of opinion of a number of maintenance departments of the leading manufacturing concerns is to the effect that the job of attaching a barrel-type flange-mounted motor is more costly in time and effort than is the case when the motor is equipped with feet.

8 Often better protection for the motor is obtained. While this in a measure is true, it is possible to obtain all the advantages of adequate protection with a properly designed shelf-type flange mounting of the general proportions shown in Fig. 10. Further development of this type of mounting is shown in Fig. 11, whereon it will be noted that the shelf mounting is provided with a hinge pin so that the motor may be tilted in such a way that the driving belts can be readily removed from the pulley.

The chief difficulty up to date which has been experienced with motors of the flange type is their relative unavailability at short notice, and this has necessitated manufacturers who use them carrying comparatively large stocks of expensive equipment, which is often slow moving on account of the great variety of electrical circuits in general use.

It is therefore the writer's recommendation that, in the present state of the art, extreme caution be used in making a general recommendation in favor of flange-type motors.

There are and there always will be drives on machine tools and other mechanical equipment where the flange-type motor is the only logical way of securing a mechanical drive which is soundly designed in accordance with the best engineering practice. These instances, however, are relatively few, and until the leading motor manufacturers can get together and standardize on flanges with a guaranteed interchangeability, it is suggested that use be continued of the foot-mounted motor with all its shortcomings.

[5] Chief Engineer, Cincinnati Grinders, Inc., Cincinnati, Ohio. Mem. A.S.M.E.

A typical example of where it would be difficult to improve upon the foot-mounted motor is provided by the modern milling machine where a standard foot-type motor is mounted in the base of the column, from which position it can be readily drawn out and serviced whenever the necessity arises. Furthermore, the motor in this position is thoroughly protected against chips, oil, and dirt.

The writer submits the following recommendations as regards procedure:

Suggest that the motor manufacturers carry through and intensify their efforts toward the standardization of foot-mounted motors, but that, concurrent with this, a study be made by all the leading electrical-equipment manufacturers of the possibilities of interchangeable standards of flange mounting, so that no consideration of mounting design will deter the designing engineer from using either the flange-mounted or foot-mounted motor whenever he is convinced that the overall efficiency of his product will be improved by the use of either one of these two types of mountings.

J. D. Riggs.[6] In an intricate machine of course it is the sum total of the various elements that determines its merits, but in an analysis of the same machine we necessarily consider and discuss one element at a time together with some others closely related. In the present case it seems the author might have extended his discussion to include some consideration of the means for connecting the motor shaft with the driven machine.

The direct connection through a flexible coupling certainly has its merits, but does not always give the required speed. Certain forms of roller drive seem to have been used in Sweden on centrifugal machines running at high speed; in England, the Garrard toothless gear has been applied in various ways with apparent success; and in France, it is learned the Société Anonyme has at least got into the patent records with a roller drive of some considerable merit.

If some form of roller transmission can be brought out to serve as a speed increaser in some places and as a reducer in others, and also function as a flexible connection, then it seems the flange mounting for motors might become the common practice. But as one purchaser can see only the merits of toothed gearing, another the reliability of the leather belt, a third wants steel belt, a fourth insists on the short-center silent chain, a fifth wants tex-rope drive, and a sixth intimates that the machine-tool builder should have already established some standard drive, so long as such a variety of demands has to be met it is not easy to standardize on a motor mounting with any hope of having the standard generally accepted.

C. E. L. Dapprich.[7] In order to place flanges on two or more circular or approximately circular motor frames and have the complete motors interchangeable in regard to mounting it is necessary to control only three or four major dimensions. This compares with five major dimensions which must be controlled on foot-mounted motors. The four major dimensions on flange-mounted motors are:

(1) The diameter of the bolt circle for the mounting bolts
(2) Size of mounting bolts
(3) The axial distance from the face of the flange to the nose of the rear bracket on the motor; and also, if accurate centering is required, as is usually the case
(4) A pilot fit on the flange which is to register with a corresponding fit on the mounting surface.

[6] Designer, Indianapolis, Ind. Jun. A.S.M.E.
[7] General Engineer, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.

If complete interchangeability is desired instead of only interchangeable mounting, then the shaft extension and its keyway dimensions also must be specified.

When a flange is to be added to the frame or bracket of a motor, it seems quite natural that a circular one, or, more accurately expressed, an annual disk flange, should be used. This is quite in line with old-established practice on flanged pipe. But the conditions surrounding the mounting of a motor on a machine are very different from those which govern the coupling of flanged pipe. The fundamental difference is that the pipe coupling is to make a joint tight against heavy internal fluid pressure, whereas on the motor it only has to resist the motor torque and keep the pilot fits in engagement. This very moderate requirement combined with the fact that in many applications it is desirable or even necessary to place motors close together or immediately adjacent to some obstruction, shows that although the annular disk flange is satisfactory in many cases, the "squared" disk is better. As long as four symmetrically spaced mounting bolts are used, it is advantageous to "square" the disk. The "squared" disk flange furnished at least as good a support as the circular disk, and it allows several motors to be mounted together with the minimum possible center distances. From this viewpoint the extra material in the annular disk flange in every case is not only wasted, but on short-center jobs where this extra material must be machined off it causes an even greater waste in labor.

The German standards on flange-mounted motors are available and have been issued in leaflet form (DIN-VDE 2941), which shows that they use the annular disk flange and on the larger sizes use eight mounting bolts instead of four. Placing the flange on the bracket has the advantage that it can be kept within the frame diameter.

As far as strength is concerned, the writer believes that on small and medium sizes of motor frames which are to be standardized four mounting bolts are quite sufficient and that even where severe vibration is encountered four bolts will still be sufficient if the pilot fit is snug.

A full and detailed discussion of a subject as important as standardized dimensions on flange-mounted motors is very desirable, and following that a careful approach to standardization is also very desirable, since all realize that a standard once set is a rigid and almost immovable thing.

Robert C. Deale.[8] While flange-type motors have undoubted advantages for many applications, particularly for special machinery, it is believed that the foot mounting lends itself better to general use, such as a standard machine tool, which must be adaptable to varying service conditions. It is known that the first machine-tool manufacturer to make extensive use of the flange mounting for motors became very much discouraged because of the difficulties encountered, and the chief engineer expressed the opinion to the writer that the company was just about ready to abandon that type of mounting, after a trial that covered several years.

Most machines, particularly machine tools, must be adaptable for motors of several different horsepower to suit the varying conditions under which the completed machine is to be used. In addition, because of the different frequency conditions found in practice, driving motors of different speeds must be used, while it is usually necessary to hold spindle speeds constant. As a result, changes must be made, either in a pair of gears, or in a belt or chain drive, between motor and spindle, in different machines. For manufacturing reasons this change is usually made in a pair of gears outside the machine proper, or in the motor pulley or sprocket. It is believed that these changes are easier to make

[8] The Norton Company, Worcester, Mass.

with a conventional foot-type motor than with a flange-type, because of the greater accessibility of the motor spindle. Some years ago the writer was interested in a machine that had a motor bracket which covered the spindle end of the motor, and consequently gave an effect similar to that of a flange mounting, although standard foot-type motors were used. It was found that there was considerable difficulty in lining up the first pair of driving gears, of which the pinion was mounted on the spindle, so that they would run smoothly. Unless all gear reductions are built inside the machine and a flexible coupling is used between motor and machine, it is believed that there will be more or less difficulty in lining up the motor and the machine. In the past it has been found that the standards of accuracy to which the average motor builder works are not such as to allow a motor of any type to be mounted without a considerable amount of time being spent in aligning it. It is believed that the foot-type motor lends itself to such alignment much better than the flange-type.

There is some question whether a flange-type motor mounted horizontally is stiff enough for the ordinary machine. When a vertical motor is so mounted, the line of action of the weight falls within the base and helps to hold the motor in place. On the contrary, the weight of a horizontally mounted flange-type motor introduces a bending action that it is believed would be undesirable and would make the machine particularly susceptible to the setting up of vibrations. In the case of most machines this would be very serious particularly for the driving motors of metal-cutting machines.

There would seem to be no inherent difference as far as protection of a motor from oil, chips, etc., between the two types. It is believed that most of the differences found are caused by differences of design. With an equal attention to this detail, it is believed that one type would be as satisfactory as the other.

Where all shafts of a machine are necessarily vertical, as in the jig borer mentioned in the paper, or in a drill, the vertical motor eliminates the use of bevel gears, that are usually less satisfactory than the spur or spiral gears generally used.

In conclusion, it is believed that it is unwise to attempt the general use of flange-type motors, particularly for general purpose machinery, unless such use is clearly called for, as such use will be very likely to cause difficulty.

DR. ING. MAX KRONENBERG.[9] The writer notes that also in the United States the flange motor is receiving more attention than before.

Among the advantages justly attributed by the author to the flange motor, there still is missing a very important viewpoint. Through use of the flange motor not only are extending parts, floor space, etc., saved, but also a great amount of machine parts. In the horizontal flange motor-driven boring machine, in use here, the short driving distance is evident. The flange motor allows for installation close to the working place, and this the writer considers its best characteristic. No more complicated transmission of the developed power is necessary; the source of power is mounted directly on the spindle box. In preceding designs much superfluous intermediate gear was necessary to transmit the power to this point. Hence, the advantages are:

(*a*) Short-distance drive in the machine and consequently improved efficiency, thus increased utilization of the initial power.

(*b*) Omission of shafts, flat circular and conical gears, bearings, etc., and also simplified construction, making for less expensive manufacture and better accessibility. Furthermore, the easy interchangeability of flange-motor and single-pulley drive is of advantage. As a further advantage of the flange-motor standardization, existing already in Germany, one will recognize:

(*c*) The easy interchangeability of the various methods of driving. The drive may be installed after sale according to the desire of the purchaser.

Allow the writer to add a few words to another point in the paper. On the fourth page the author states that the motors for radial boring machines in the United States are installed as a counterweight on the arm, whereas in Europe they are installed on the drill head. The author considers the American method to be more advantageous. However, the writer wishes to state that the motor "as counterweight" would be correct only in case the machine is not working. As soon as the radial boring machine works, an upward-directed force is applied on the axis of the drill which tends to bend the arm upward. Hence, if the motor is installed as a "counterweight," then it assists in this upward bending. However, if the motor is installed on the drilling head, then this bending is counteracted. Hence the additional weight of the motor is no disadvantage, but an advantage.

[9] Consulting Engineer, Berlin, Germany.

MSP-53-9

Fundamentals of Machine Polishing

By ROBERT T. KENT,[1] UTICA, N. Y.

The success of machine polishing depends on the proper coordination of the variables present in the work itself, in the polishing wheel, in the abrasive, in the machine, and in the glue used in setting up the polishing wheel. These variables are discussed in detail, and their relation to each other are set forth. Typical polishing machines are described, as well as examples of successful machine polishing.



IN NO INDUSTRY is there so much misunderstanding, due to improper use of terms, as exists in the polishing and metal-finishing industry. In order to avoid confusion in the present paper, therefore, it will be well to establish a few fundamental definitions.

Polishing is the art of producing a smooth uniform surface on metal by means of an abrasive wheel, of greater or less flexibility, the abrasive being glued to the face of the wheel. Polishing begins where grinding leaves off, and should more properly be termed flexible grinding.

Buffing is the art of putting a luster or "color" on a polished surface. It is done by means of an abrasive composition bonded with a wax or grease, smeared on the face of a buffing wheel, made up of a series of cloth disks. The terms *polishing* and *buffing* are frequently interchanged by misinformed persons, and this interchange is often the cause of confusion and expense.

A polishing wheel is a flexible wheel with abrasive grain glued to its cutting surface. It may be of many different forms and materials, as described in detail later in this paper.

A polishing or buffing machine is a machine for presenting the work to be polished or buffed to the polishing or buffing wheel. The machine may be a separate piece of apparatus from that driving the wheel or the wheel may be integral with the machine. Polishing machines are frequently called automatic machines. This is incorrect, as the present machines are not automatic in their action, but require manual control and loading. Usually they also require manual unloading, although sometimes automatic unloading is possible. The full-automatic machine is possible but is still in the future.

The success of machine polishing depends on the consideration and coordination of all the variables that may affect the condition of the final surface. These variables may be enumerated as follows:

1 *Variables of the Work*
- *a* Material, as iron, steel, brass, copper, aluminum, etc.
- *b* Condition of original surface, as hot rolled, cold rolled, ground, machine finished, i.e., showing tool marks, pitted, scaled, etc.
- *c* Condition of final surface, i.e., degree of luster desired, whether edges must be held sharp, whether or not holes must be free from dragging or rounding over, whether or not the finished surface must be held within close tolerances
- *d* Contour of work.

2 *Variables of the Polishing Wheel*
- *a* Type of wheel, as disk, clothflex, compress, etc.
- *b* Material of the wheel, as canvas, muslin, leather, walrus felt, sheepskin, paper, etc.
- *c* Density, as soft, medium, hard, superhard, etc.
- *d* Cushion of the wheel, i.e., the depth of the flexible portion of the wheel.

3 *Variables of the Abrasive*
- *a* The kind of abrasive, as natural abrasive, such as Turkish emery, or manufactured abrasive, as Alundum, Crystolon, Carborundum, Aloxite, etc.
- *b* The grain size of abrasive.

4 *Variables of the Machine*
- *a* The peripheral speed of the polishing wheels
- *b* The speed at which the work is fed past the wheels
- *c* The pressure of the wheels on the work
- *d* The provision, or lack of provision of oscillation, in either the work or the polishing wheels, in order to avoid straight-line polishing marks in the surface
- *e* The cushion under the work as it passes beneath the polishing wheels
- *f* The tendency of the machine to set up vibration in either the work or the polishing wheels
- *g* The type of fixture used for holding work in the machine.

To these variables, there may be added another set, namely, the variables affecting the glue with which the abrasive is bonded to the wheel. These variables are as important as, if not more important than, any of the others. They are, however, completely covered in B. H. Divine's paper, "Prerequisites of Successful Polishing,"[2] and further reference will not be made here to them.

These variables are all more or less interrelated. For instance, the character of the material will determine the character of abrasive. A sharp, hard, abrasive that fractures along definite planes as it wears, such as Alundum, is the proper abrasive to use for steel. Such an abrasive is unsuitable for cast iron that requires a mirror finish. This material calls for an abrasive such as Crystolon that will crush rather than fracture. Then, too, the condition of the original surface influences the size of the abrasive grain that is to be employed. The initial abrasive should be large enough to cut down to the bottom of the deepest pit, and to reduce the metal surrounding that pit to the same level. This is followed by abrasives that are successively finer, until the final one produces the desired surface. On the other hand, it will be a mistake to use an abrasive coarser than necessary on the initial wheels, as its effect will be to put scratches in the work that the

[1] Director of Engineering, Divine Brothers Company. Mem. A.S.M.E. Mr. Kent was graduated from Stevens Institute of Technology in 1902 and for a time was associated with the Link Belt Co. of Philadelphia. For four years he was engineering editor of *Iron Trade Review*, and for many years was engaged in consulting work in the shop-practice and management fields. During the war he served as chief engineer of Meyer, Morrison & Co. and in this capacity supervised the organization of several large metal-working plants. Robert T. Kent is the son of William Kent, the author of Kent's "Mechanical Engineers' Handbook," and is himself the editor-in-chief of the 10th edition.

Presented at the Fourth National Machine Shop Practice Meeting, Chicago, Ill., Sept. 22 to 24, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] A.S.M.E. Trans., 1927–1928, vol. 49–50, MSP-50-6.

following wheels and finer abrasives must remove. Much money may be wasted in initial investment and in operating costs, by unwise selection of the sequence of abrasives.

The contour of the work will influence the selection of the type of wheel. Work with a plane surface calls for a hard, somewhat rigid wheel. Work that has a curved contour calls for a softer, flexible wheel that will fit itself to the surface to be polished. The degree of finish to be obtained also influences the speed of the work past the polishing wheel, as the slower the speed of the work past the wheel the better will be the finish.

One factor in machine polishing that is frequently overlooked is the fact that in a machine, once the work has passed the wheel, that wheel is finished with it. In hand polishing, if the surface formed is not satisfactory, the work can be applied again and again to the wheel as often as may be necessary. Hence, machine polishing usually calls for a larger number of wheels than hand polishing. For example, if a hand polisher obtains a satisfactory finish by passing work four times over a wheel set up with No. 90 abrasive, and finishes it with two passes over a wheel set up with No. 120 abrasive, the indications are that a six-wheel machine will be required to simulate the hand operation. This, however, may be modified by the differences inherent between machine and hand work. It is at this point that experience fails to be a guide, and the laboratory should take charge.

With the interrelation of the several variables in mind, and also the limitations of machine polishing as outlined, a detailed discussion of the variables affecting the application of the machine to polishing may be undertaken.

Variables of the Work

a Material. The material of which the work is composed determines the character of abrasive that must be used. Alundum or any manufactured abrasive of similar characteristics is suitable for steel and brass. These materials require sharp cutting edges to be continuously presented to the work. The abrasive, therefore, should fracture in such a manner, and along such fracture planes, as to form cutting edges that are always of the same character. An abrasive that fractures in such a manner as to cause splinters and slivers will prove unsuitable, as these splinters will put scratches in the work that will later have to be removed.

Soft cast iron presents an entirely different problem, due to the difference in metallurgy between it and steel. The sharp-cutting grains tend to dig the iron particles out of the softer carbon matrix and to roll them along the work under the wheel. This produces deep scratches in the finished surface, and attempts to remove them with following wheels set up with the same abrasive only tend to aggravate the difficulty. A solution is found in the use of an abrasive that will crush instead of fracturing. An example of this type is Crystolon. The importance of the selection of the proper abrasive for the material is indicated by the fact that much time and money were spent in attempting to machine polish the cast-iron parts of a certain article that is in wide household use. The attempts were a consistent failure until the laboratory discovered the unsuitability of the sharp cutting abrasives for this material and substituted Crystolon.

Space will not permit a detailed discussion of the selection of abrasives suitable, not only for the different metals, but also for the alloys and grades of those metals. The following is an abbreviated list of abrasives that have proved suitable for the metals indicated, and under the conditions under which the work was done:

Steel.....	Alundum	Cast iron....	Crystolon
Aluminum..	Turkish emery	Duralumin..	Turkish emery
Brass.....	Alundum	Copper	Turkish emery

Another factor that depends on the character of the material is the horsepower required. Brass requires several times the horsepower that steel does. Other materials require still different amounts of power. Research has not yet progressed to the point where it can be stated definitely what power is required over a certain range of conditions to polish the different materials. One fact, however, can be stated with certainty: *The size of motor in a hand lathe that will produce a desired finish is absolutely no criterion of the size of motor that should be installed in a polishing machine.* There are a number of reasons for this fact, but they may all be summed up in the statement that there are inherent differences in the action of the wheel on the work in the two methods.

The practice of the company with which the author is connected is to simulate in the laboratory the actual conditions under which the work will be done in the machine, and then to take wattmeter readings of the power required. It is believed that this is the only safe method to use in powering polishing machines.

b Condition of the Original Surface. The condition of the surface of the work as it comes to the polishing machine falls under several heads. The work may come directly from the forge shop or rolling mill, as in the case of automobile bumpers or steel shapes. It then will be coated with mill scale, and perhaps be deeply pitted. The mill scale will be hard and difficult for the polishing wheel to break. Furthermore, it will clog the wheel and render it ineffective in a short time. Deep pits require that the polishing wheels remove all the surface metal down to the bottom of the deepest pit and may impose a wholly unnecessary burden on the wheels. Tumbling or pickling may improve both of these adverse conditions, but if not carefully done may enhance the difficulty.

The surface may be ground or it may be machine finished. In either event, the polishing wheels must first remove the metal down to the bottom of the deepest grinding scratch or deepest tool mark. The surface may be uniformly coated with a hard scale or oxide through which the polishing wheel must break before it can actually begin its true work of polishing. This scale may dull or clog the wheel, greatly reducing its efficiency, or even putting it out of business in a very short time.

The condition of the original surface is an item of major importance in machine polishing. It is axiomatic that the polishing wheel will work best on a clean, uniform surface. If that surface is not provided for it in the beginning, the polishing wheel must make it. It follows then that the better the original surface, the lower will be the investment in machinery, in polishing wheels, and the lower will be the labor cost of the polishing operation.

A case in point is the experience of a large company making automobile bumpers. A machine was installed for polishing these, the number of wheels being determined by tests on production quantities of the regular run of steel used by the bumper manufacturer. After the machine had been in operation for some time, complaint was made that it was not cleaning up the bars, and that they required a second and third pass through the machine. An examination of the steel as it came to the machine showed that it was much inferior, so far as surface was concerned, to the steel in the reference sample from the test run. Inquiry revealed that the bumper manufacturer had changed his source of supply. An insistence on adherence to steel specifications, rigid inspection, and liberal rejections of poor steel, shortly thereafter caused the polishing operation to resume its original efficiency.

c Condition of Final Surface. The condition of the final surface of the work dictates both the number and character of polishing wheels that must be used in a machine. Several considerations enter into the final surface, one being its appearance or "color," and the other the dimensional characteristics of the work itself. By the latter is meant the closeness to which the work

must be held to dimensions, and also whether corners and edges must be held sharp and square. Quite frequently work that is susceptible to machine polishing in all other respects presents considerable difficulty, for the reason that its contour is such that the polishing wheel will round edges as it passes over them, or drag the edges of drilled holes until the holes are elliptical. When such difficulties are encountered, they may be overcome in several ways. A proper selection of the wheel may obviate some of them. A change in the design of fixture may eliminate others. A change in the direction or rotation of a wheel quite frequently offers a solution. A soft polishing wheel striking against a sharp edge or corner will invariably round it over. If each wheel be made to trail over the edges with which it comes in contact, it is possible to hold the edges square and sharp.

Frequently polished work must be held within tolerances sometimes as fine as plus or minus 0.001 in. Such close tolerances call for micrometer adjustments in the machine and a careful selection of the wheel. They also call for extreme care in setting up the wheel and making it truly cylindrical at operating speeds. The wheel should be of hard density, as a soft wheel will flex on the work, and thereby cause lack of uniformity in the contour of the surface polished.

Too much stress cannot be laid upon making the wheel truly cylindrical at operating speeds, particularly where the surface to be polished is relatively wide. It is a comparatively easy matter to make a wheel truly cylindrical when truing it up in the lathe preparatory to applying the abrasive head. However, it should be borne in mind that in the ordinary wheel-dressing machine or engine lathe, which is sometimes used for the purpose of truing wheels, the speed of rotation is relatively low. When in operation, the polishing wheel rotates at from 1800 to 2500 r.p.m., depending upon the diameter. The wheel will become distorted at these high speeds, and the surface that comes in contact with the work is no longer cylindrical. After considerable difficulty, due to the failure to produce a uniform finish upon wide areas, it was discovered that the cause of the trouble was the distortion of the polishing wheel at operating speeds. To correct this condition, a machine was developed to true the wheels at the same speed at which they were to be used. The wheel, when at rest, would not be truly cylindrical, but would take this form when brought up to working speed. Thereafter there was no difficulty obtaining truly plane surfaces over wide areas.

If the work is to be held within close tolerances the material entering into the construction of the wheels must be considered. As indicated, a soft wheel will flex on the work and produce uneven surfaces. This immediately rules out a large number of materials, and limits the choice of the wheel to practically two types, the compress-leather or the compress-canvas wheels. Since both of these wheels can be made to extremely hard densities, it is a comparatively simple matter to maintain tolerances with them.

Variables of the Polishing Wheel

a Type of Wheel. Probably the oldest type of polishing wheel is a wooden disk made in one piece, or built up of laminated sections with a strap of leather glued and pegged to the face. Such a wheel produces a high-class finish upon almost every type of metal, but it is rapidly passing out of use for a number of reasons which need not be enumerated here. In general, it may be stated that this wheel is poorly or not at all adapted to use in a polishing machine.

The type of polishing wheel that has been in most general use up to a few years ago is what is technically known as the clothflex wheel, consisting of sections of ordinary cotton buffs glued or sewed together to form a wheel of the requisite width of face. These wheels were frequently home-made, in that the polisher bought the buff sections, and glued them together to fit his needs. Wheels made in this manner possess many defects, as they usually are out of balance, lack uniform flexibility and cushion, and possess other defects that render them unsuitable for use in the polishing machine. As polishing has gradually gained recognition in the industry as a precision operation with considerable engineering behind it, the fact has become recognized that good work could not be obtained with crude tools. The clothflex wheel, as today manufactured for precision work, as in the polishing machine, is made from carefully selected buff sections properly balanced, and with the sewing of the buff sections themselves, and the sewing and gluing of one section to another, so designed as to give a wheel exactly suited for the work that it is to do.

Many manufacturers even today have the idea that a buff is a buff, and a polishing wheel is simply a collection of buffs glued together. To illustrate the fallacy of this idea, it is only necessary to state that eight different grades of cloth are used in the manufacture of buffs which enter into polishing wheels. Every one of these grades is selected because it is particularly adapted to one class or another class of work. Furthermore, the buffs made from these different grades of cloth may be sewed in several different ways. They may be spirally sewed over the entire surface, at wide or narrow intervals, or they may have two or many rows of concentric sewing from the center hole out to the periphery. Each class of sewing is dictated by the character of work that the wheel is to do, and a wheel that is eminently fitted for one job will prove a hopeless failure on another. Furthermore, the various buff sections of which the clothflex wheel is composed may be glued together from the center hole clear out to the circumference of the wheel, or the gluing may stop at 3 in., 2 in., or any other distance from the circumference, depending upon the hardness and the cushion desired in the wheel. If a very soft wheel is required, the gluing may be omitted altogether, and the various buff sections hand sewed together with two or more rows of concentric sewing.

The foregoing are but a few of the variables entering into the clothflex wheel. The fact that there are so many variables, each one of which has its influence upon the character of work done by the wheel and on the efficiency of the wheel, is sufficient indication that the selection of the proper wheel for the work is a most important item in machine polishing.

Another type of wheel which is largely used in machine polishing is the compress wheel. This wheel is constructed of blocks of material set on edge, clamped between two steel flanges, and compressed radially, circumferentially, and longitudinally. It is essentially different in its action from the clothflex wheel, in that the cutting edges of the fabric in the clothflex and other disk wheels run parallel to the movement of the work across the wheel. In the compress wheel, the cutting edges are transverse to the movement of the work, and the effect of this change in direction is most important. One advantage of the compress wheel that is possessed by no other type of wheel is that the cushion and density of it can be exactly controlled. By cushion is understood the depth of flexible material projecting out from the rigid part of the wheel, in this case from the steel flanges.

By density is meant the degree of hardness of the wheel. Various jobs require different depths of cushion and different densities, in order to produce satisfactory work. The compress wheel can be made in all depths of cushion up to 3 in., and in certain cases, up to 4 in. Within certain limits of diameter and cushion, the density can be varied from super-soft to super-hard. A super-soft wheel is so soft that a person can push his thumb into the surface of the wheel, while a super-hard wheel is so dense that it is difficult to drive a nail into it with a hammer. The compress wheel can be made of many different materials. The most common are heavy canvas and bullneck leather, although other ma-

terials sometimes used are walrus, sheepskin, paper, and felt. The characteristics of each of these materials vary from the characteristics of each other in a given density and cushion. The proper selection of the material, of the cushion, and of the density in a compress wheel will determine whether or not a machine-polished job will be a success or a failure.

Another type of wheel that is commonly used is the disk type, built up of disks of canvas or bullneck leather, wool felt, walrus hide, and other materials that are homogeneous in structure and can be cut into disks and glued or clamped together. Each of these wheels has its own special field on the polishing machine. For instance, for roughing out work, such as automobile bumper bars, the disk canvas wheel, headed with the coarser abrasives, is superior to any other type of wheel. It is highly unsuited, however, for finishing, and for this purpose the compress-canvas wheel of medium density and 2-in. cushion has shown itself superior to any other.

It is impossible in the space available to give a complete list of the uses to which the several types, grades, and densities of wheels available are adapted. The following notes are merely indicative of some of the fields in which these wheels may be successfully used:

The disk canvas wheel is especially adapted to roughing work, with the coarser grains, say up to No. 60. It is flexible and can be used on work with curved contours such as agricultural machinery parts, plowshares, oval or straight-faced automobile bumpers, etc.

The clothflex wheel is used for contour work, similar in character to the disk canvas wheel, but with the finer grains. It may be made highly flexible and will adapt itself to curved surfaces such as shovels, automobile bumpers, and more or less irregular shapes.

The compress wheel is a precision polishing tool of the highest order. It is an all-around wheel that can be used for practically every purpose that any other wheel can be used. Its face can be formed to fit contour surfaces, and by the possibility of regulating its cushion and density it can be made exactly to fit the job it is to do. Due to its high first cost, it is in some cases not the most economical wheel for roughing with the coarser grains, but for all other work up to the finest finishing it is believed to have no equal. Its high first cost is counterbalanced by its extremely long life. Wheels that have remained in service for upward of 20 years are not uncommon, and one instance is known of a wheel that was taken off the polishing lathe after 31 years of continuous use.

The compress-canvas wheel is extensively used in the polishing of cutlery, flatirons, shears, skates, golf-club heads, bicycle and automobile parts, machine parts, and similar work.

The compress-leather wheel has its field in the initial operation on flatirons, the polishing of carpenter's chisels, plane blades, axes, etc. It is used where edges have to be held sharp, and holes kept free from dragging.

The compress-felt wheel is employed on work where a high color is required, such as surgical instruments.

The compress-sheepskin wheel is comparatively little used, except in those cases where a flexible wheel that will give a very fine super finish is needed.

The compress-paper wheel is adapted to rough work. It has little flexibility, and approximates in its action the solid grinding wheel.

The solid felt wheel is used for polishing glass, and for greasing and coloring operations where a high color is required. It has a distinct field in the polishing of formed surfaces of stainless steel.

Variables of the Abrasive

The variables of the abrasive as they affect the operation of the polishing machine have been discussed in general in the preceding paragraphs. They will not be further discussed here.

Variables of the Machine

a The Peripheral Speed of the Polishing Wheel. The polishing machine should be designed, wherever possible, to maintain a constant peripheral speed of the polishing wheel as close as possible to 7500 lin. ft. per min. In general, the higher the peripheral speed, the more efficient is the action of the abrasive. The figure 7500 ft. is chosen because this speed is about the maximum at which the glue holding the abrasive to the surface of the wheel will maintain its strength under the action of the heat generated by the operation of the wheel upon the work. An increase in the speed of the wheel would increase the efficiency of the cutting momentarily. It would, however, generate sufficient heat to soften the glue, and thereby permit the abrasive to be torn from the wheel before it had rendered its full service as a cutting medium. The overall efficiency, therefore, would be less than it would be were the wheel operated at a lower speed.

If the peripheral speed of the wheel is decreased below 7500 ft. per min. the cutting action of the abrasive is less efficient and a longer time is required for the removal of a given amount of metal. The general rule, therefore, for wheel operation is to drive the wheel at the maximum speed that is possible without softening the glue. With the best grades of glue, this has been demonstrated to be in the neighborhood of 7500 lin. ft. per min.

Another factor, however, enters into the question of speed in its relation to heat generated. This is the work itself. Speeds that may be entirely safe from the standpoint of the wheel may be ruinous to the work, for the reason that the heat generated will have adverse effects upon the metal of which the work is composed. For instance, in die castings or other alloys of zinc, if the wheel speed is so high as to generate sufficient heat to distort or melt the metal, the speed must be decreased even though it means lowering the speed below the ideal speed for maximum efficiency of the wheel. Other metals may suffer changes in temper, hardness, etc., as a result of the heat generated by excessive wheel speed. All of these things must be taken into consideration in fixing the speed at which the wheel spindles of a polishing machine shall operate. Here again the laboratory is the safest guide in determining the proper wheels and wheel speeds, and also the speed at which the work passes under the wheel.

The polishing wheel, when correctly built and properly maintained, should not decrease appreciably in diameter with use. The wheel should never be operated after the abrasive head has worn through to the fabric of which the wheel is composed. Otherwise, it will burn and require trimming, which means a reduction in diameter, with a consequent reduction in peripheral speed. In setting up the wheel, a protective head of glue should be placed on it before the glue that holds the abrasive is applied. As soon as the wheel wears down to this protective head, it should be removed from service, the old abrasive removed, and a new head applied. Given this treatment, the diameter of the wheel should be maintained fairly constant for a period of years, and no provision, therefore, need be made for a change in spindle speed due to wheel wear. On the other hand, if the polishing machine is to be used for a variety of work requiring different sizes of wheels, and therefore, different spindle speeds, provision should be made for changing the spindle speed with change in wheel diameters. This has proved a more or less difficult matter until recently, but the advent of flexible endless-belt drives has made it a comparatively easy matter to change spindle speeds by a change in the pulley diameters in the drive. In the design or selection of a polishing machine, attention should be given to the ability to change the spindle speed if the character of work is

such that there will be a change in the wheel diameter from time to time.

b Speed of the Work. The speed of the work past the wheel is an important consideration in machine polishing. The speed depends upon the material of which the work is composed and the degree of finish that is required. It varies between wide limits. For instance, strip zinc can be fed past the wheels at the rate of 200 lin. ft. per min., whereas 15 ft. per min. is a high rate of speed for stainless steel. Another item entering into the speed at which work can be done in machine polishing is the speed at which separate pieces can be loaded on the machine. Usually this is a limiting factor in the capacity of polishing machines. It frequently happens that a polishing machine has a capacity of from 200 to 300 pieces of a given size and shape per minute, but it is manifestly beyond the ability of an operator to feed work at this rate, particularly where the work must be more or less accurately placed in work-holding fixtures. It sometimes is possible to arrange a machine so that two or three operators can load it, the unloading being taken care of by an equal number of operators, or automatically. Sometimes the shape of the work permits the provision of mechanical feed, but such cases are the exception, rather than the rule.

Another factor governing work speed is the degree of finish required. A polishing wheel requires a definite interval of time in which to do its work. The work must remain under the wheel long enough to permit the wheel to get a bite upon the metal and to remove it. While it is entirely possible to design machines that will feed the work past the polishing wheel at rates of 200 ft. per min., or even higher, it is hardly feasible to operate at this speed. The polishing wheels would merely ride over the work without removing metal. Then, too, the slower the work speed the higher the luster that will be obtained with any given size of abrasive grain. The overall finishing time for a given piece of work may actually be reduced by the adoption of a lower work speed, in that this lower work speed will give the desired finish which otherwise might require an additional operation to obtain. Again, the choice of work speed is, as a rule, best determined in the laboratory, rather than in the shop.

c Pressure of the Wheels on the Work. Ability to regulate the pressure with which the polishing wheels bear on the work is highly important from two standpoints: first, the effect on the wheels, and second, the effect on the work. Excessive pressure causes rapid destruction of the abrasive head, excessive heat, and eventual burning of the wheels. As regards the work, the heat engendered by the high pressure in some classes of work may change the character of the metal and render it useless for its purpose. In any event, excessive pressure greatly increases the operating cost, as it wastes grain, glue, and labor for the setting up of wheels, and it also requires more operations on the work due to the inefficiency of highly heated wheels. In the design or selection of polishing machines, attention should be given to the possibility of adjusting the pressure, either manually or automatically. A machine that has no provision for this regulation should be rejected forthwith, since it will be nothing but a constant source of trouble and expense.

d Provision for Oscillation. In hand polishing the workman handles his work in such a manner as to "cross the scratches." That is, he presents the work to the wheels at varying angles, so that wheel marks intersect, and thus tend to obliterate each other. In most cases of machine polishing, and particularly in straight-line work, it is desirable to approximate this effect by moving either the wheel or the work transversely to the line of travel of the work past the wheel. This is done by means of an oscillator on the wheel spindle or a mechanism that oscillates the table. The resulting wheel marks then take the form of a sine curve, which is a much more desirable surface than one with straight-line marks, parallel to the direction of motion of the work. The reason for this is that, without oscillators, each wheel tends to follow in the wheel marks left by the preceding wheel and to deepen them instead of cutting them out. If the final surface of the work requires the appearance of parallel lines, oscillation may be omitted on the last one or two wheels. As a general rule, however, oscillation is a desirable feature in nearly all classes of machines.

e The Cushion Under the Work. In some classes of work it may be desirable to divide the cushion required for flexible grinding between the wheel and the work. The wheel is made of the proper density and cushion to give the desired finish, and a cushion surface is provided under the work to enable it to give sufficiently to take care of slight variations in contour that would cause the wheel to cut deeper were the work not able to recede somewhat from the wheel. The cushion can be provided by means of a resilient material, such as a canvas or leather belt, or springs. The greater the cushion the less accurate will be the surface so far as dimensional properties are concerned. Hence it follows that on highly accurate work, where the dimensions must be held within close limits, or corners and edges held up sharply, little or no cushion should be provided under the work. On the other hand, where the work does not have to be held to close limits, and where the shape may be slightly irregular, production is facilitated by some cushion under the work. For instance, in polishing cylindrical work, which may be slightly out of round, cushion under the work will permit it to recede slightly when the longer diameter comes under the wheel. Were this not possible, a flat spot might be ground on the work, the work might be burned, or the abrasive stripped from the wheel. Cushion under the work is not so important a factor as some of the others enumerated, but, nevertheless, lack of proper cushion may be the cause of failure in adapting a certain class of work to machine polishing.

f Vibration in Machine or Work. Vibration in a polishing machine produces the same evil effects that it does in any other kind of machine, except that these effects are magnified due to the very nature of the work that the machine is called upon to do. The object of polishing is to provide a smooth uniform surface on the work. Vibration causes the polishing wheel to put chatter marks in the work and thus destroys the very object aimed at. Due to the high speed of the polishing wheel, vibration is more easily set up than in machinery operated at lower speeds, and consequently must be guarded against more carefully.

The more common causes of vibration are: Inadequate machine foundations, improperly or lightly constructed buildings, light floors, machines improperly located on upper floors, insufficient weight in machine bases, shafts too light, or with bearings too far apart, giving them a tendency to whip at high speeds; insufficient bearing surfaces; improperly braced machine members, or members whose vibration period corresponds to some recurrent motion in some moving part of the machine; and unbalanced polishing wheels. One of the mistakes made in the design of earlier polishing machines was the provision of light frames. Some were made of structural sections riveted or bolted together. Most of these machines have failed in one respect or another. The present-day polishing machine is a rugged structure, with heavy box-type frames, ball or roller bearings in the moving parts, massive spindles, and automatic lubrication.

Vibration can also be caused by the work being improperly seated in the fixtures, although such vibration usually is local, and confined to the work itself. A good machine should be sufficiently heavy to absorb such local vibration, although its effect on the polishing operation may be just as serious as vibration in the machine itself.

Too much cushion also may tend to set up vibration in the polishing wheel. An extreme example of this is the polishing of automobile bumpers. The springiness of the bumpers themselves furnishes a particularly resilient cushion, and at times the vibration period of the bumpers is such as to amplify the movement and actually to cause the wheels to bounce off the bumpers as they pass over them. In this case, the cushion is lessened by providing supports at two or three points under the bumper, and attaching dash pots to each wheel spindle to damp the vibration.

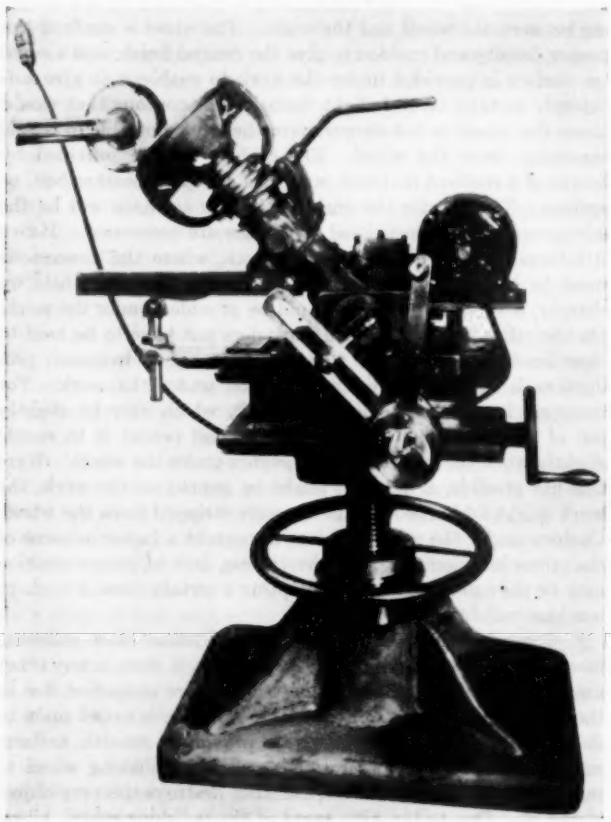


FIG. 1 MACHINE FOR FINISHING THE INTERIOR SURFACE OF HEADLIGHT REFLECTORS

g Types of Fixtures. Properly designed fixtures are a prime essential in machine polishing. Their function is to hold the work firmly, to present it in the correct position to the polishing wheels, and, in such cases as may be necessary, to provide a certain amount of resiliency and automatic adjustment of the work to the wheel. Fixtures must have certain characteristics embodied in their design and construction. First, and most important, they must hold the work so that it cannot be dislodged when it comes in contact with the wheels. It is just about as safe to stand in the path of a rifle bullet as to stand in the path of a piece of work thrown out of a fixture by a polishing wheel. Many jobs that are otherwise ideal for the polishing machine are impossible because of the impossibility of safely holding them in any practical type of fixture.

Another important point is ease of loading and unloading the fixture. A polishing machine that is capable of passing 60 pieces of work per minute under the wheels is badly handicapped if the fixture is designed so as to require two or three seconds for loading. In many cases the fixture is attached to the machine, and must be loaded and unloaded "on the run" as it were. It therefore must be easy to load and unload, and free from complicated locking devices. In other cases, the fixture is detachable, is loaded at the bench, and transferred loaded to the machine. Such fixtures should be light, yet rigid. All fixtures should have provision for compensation for wear, and in those cases where the fixture masks a portion of the work for one reason or another, the masking part must be capable of easy replacement. The fixtures should be made so that they can be placed on the machine in one position only. Otherwise the work may be ruined or thrown from the machine, or the fixture itself may be thrown. Several serious accidents have occurred through neglect of this precaution.

Where the shape of the work precludes the use of mechanical fixtures, substitutes may sometimes be employed. For very flat work such as ornamental plates or spherical or similarly shaped work that is to be polished on the inside, and which is impossible of gripping in any type of mechanical chuck, a vacuum chuck frequently will prove to be feasible. For articles made of steel or iron the magnetic chuck is sometimes employed, although for polishing operations it possesses certain disadvantages. For flat strips or sheets, pressure rolls on either side of the polishing wheel frequently may be substituted for fixtures.

The success or failure of many machine-polishing jobs more often than not depends on the ability to design a fixture that will properly hold the work. There is a wide field for ingenuity and real engineering in this one phase of polishing alone.

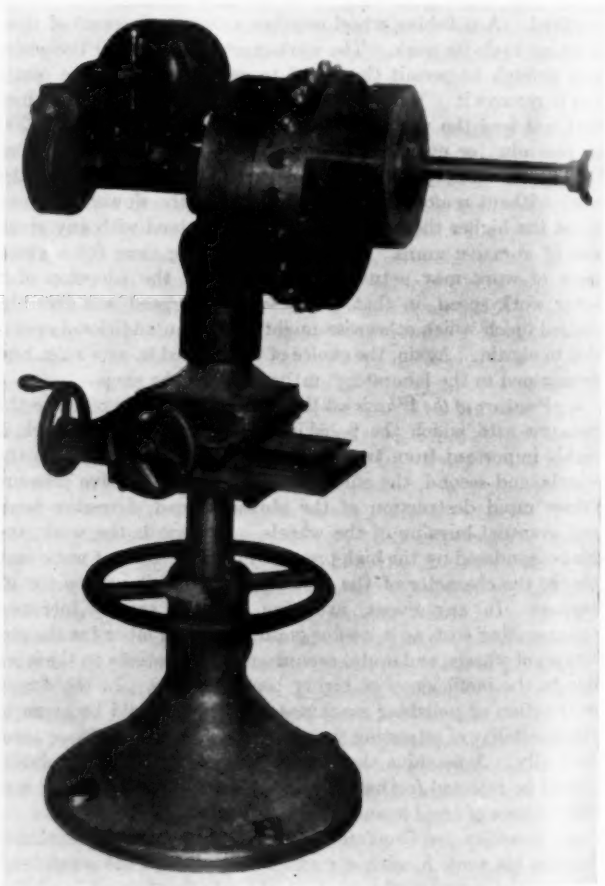


FIG. 2 POLISHING MACHINE FOR FINISHING HEADS OF BOLTS, ETC.

TYPICAL POLISHING MACHINES

Polishing machines may be divided roughly into two classes:

those which rotate the work in contact with the polishing wheel, and those which pass it in a straight line past the wheels. Examples of the former class are shown in Figs. 1 to 5 inclusive.

Fig. 1 is a machine for finishing the inside of headlight reflectors and similarly shaped articles. The reflector is placed in the chuck, which is set at an angle from the vertical. The chuck is mounted on a slide which is moved forward to bring the work in contact with the wheel, being actuated by the handle in the foreground. When in contact with the wheel, the work is rotated by means of the small motor shown above the slide, working through a bevel-gear train.

Fig. 2 shows a machine for finishing the heads of bolts and other objects which have a long stem. It consists essentially of a drum containing a number of chucks which rotate about their own axes while the drum itself is rotated about an axis parallel with the axis of the polishing wheel. The operator of the machine drops the work to be finished into the holes in the chuck, which lock on the work as it approaches the polishing wheel. After the work



FIG. 3 FOUR-SPINDLE MACHINE FOR POLISHING CIRCULAR WORK

has passed the polishing wheel, the chuck opens and the work drops out when it comes to the lowest position on the drum.

Fig. 3 is a four-spindle polishing machine for finishing small circular objects, such as hub caps, etc. The work is carried on chucks mounted on spindles projecting from the face plate. Each spindle is rotated about its axis as it comes in contact with the polishing wheel. The machine is so arranged that the face place indexes at definite predetermined intervals, thus allowing the work to remain in contact with the polishing wheel for a sufficient length of time to obtain the finish desired. Machines of this type are frequently provided with chucks which automatically unload the work when the spindle is in the lowest position.

Fig. 4 illustrates a machine for polishing long circular work. The work is mounted upon a chuck, supported at the outer end by the overhanging arm, and rotated against the wheel. At the same time, the spindle is reciprocated back and forth so that the polishing wheels cover the entire cylindrical surface.

Many variations of the foregoing machines are made to adapt them to different classes of work. The illustrations are merely typical of the varieties of work that are possible in this type of machine.

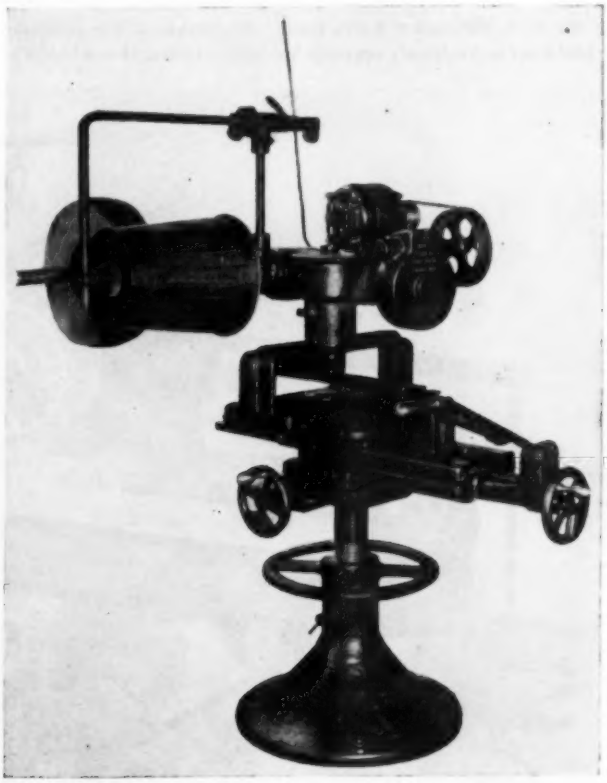


FIG. 4 RECIPROCATING TYPE OF MACHINE FOR CIRCULAR WORK



FIG. 5 ROD AND TUBE POLISHING MACHINE

Fig. 5 shows a machine for finishing tubes or other long cylindrical objects. The work is carried upon the two longitudinal

rollers in the foreground, and is moved past the wheels by the belts shown immediately behind the rollers, these belts being set at an angle to correspond with the speed of longitudinal movement desired. The belts are driven from the motor shown at the rear of the slide, through a worm gear. In operation, the polishing wheel is set immediately opposite the belt, which acts as a brake to prevent the work rotating at the same speed as the polishing wheel, and also causes it to move past the wheel. In the illustration, the relative position of the polishing wheel and the belt is shown on the second machine. The polishing wheel opposite the first belt has been omitted in order to show the details of the machine.

Fig. 6 shows one of the earlier types of continuous straight-line polishing machines. Originally designed for polishing long flat bars or strips, it has been found adaptable to the finishing of a great variety of work of many shapes and sizes, some of which are illustrated elsewhere in this paper. The principal feature of this machine is the carrying of the polishing wheels upon a swinging arm, which has a movement about the spindle A, permitting the wheel to rise and fall with change in contour of the work. Each wheel is driven by a motor directly connected to the wheel spindle. Wheels and motors are balanced by means of counterbalance weights shown at the left of the illustration. The work to be polished is placed upon the conveyor belt, and

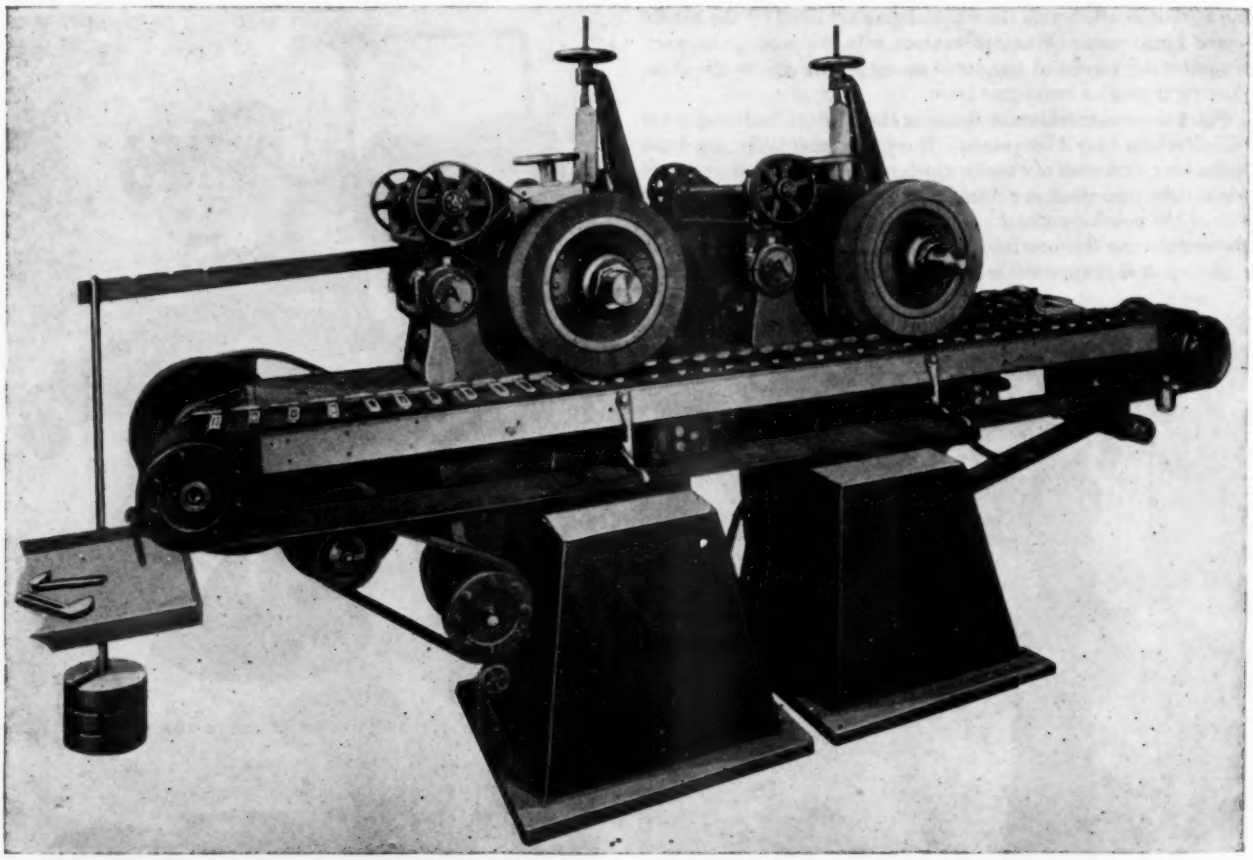


FIG. 6 EARLIER TYPE OF CONTINUOUS STRAIGHT-LINE POLISHING MACHINE

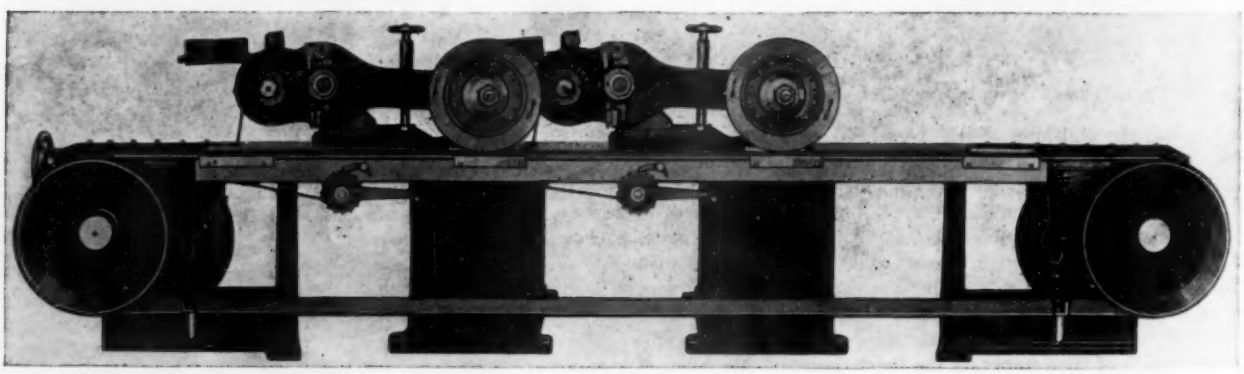


FIG. 7 PRESENT FORM OF CONTINUOUS STRAIGHT-LINE POLISHING MACHINE

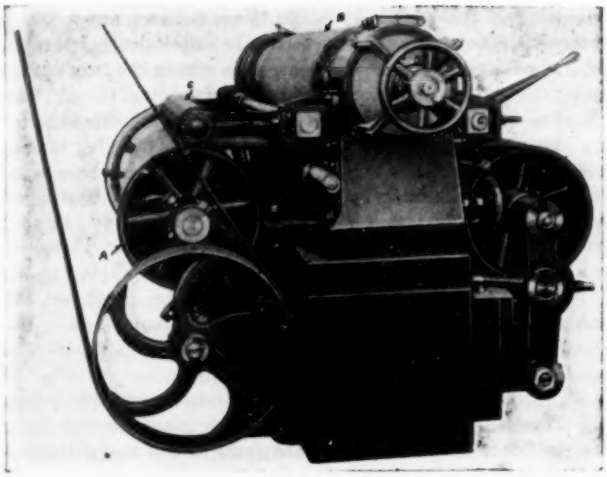


FIG. 8 SHEET-POLISHING MACHINE

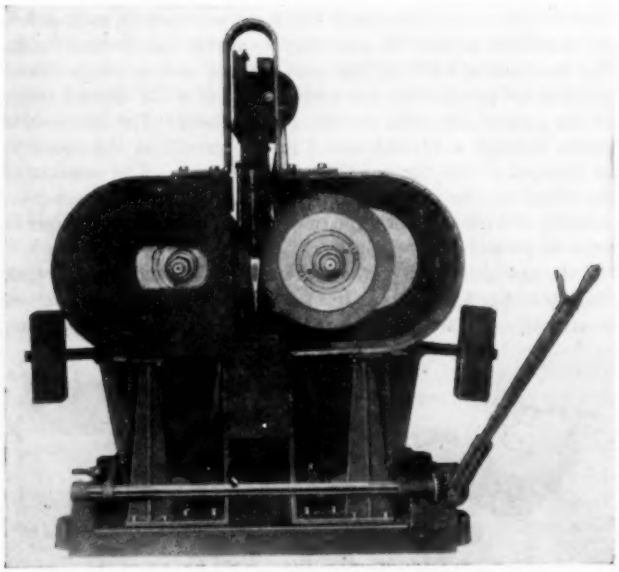


FIG. 10 AXE-GRINDING MACHINE

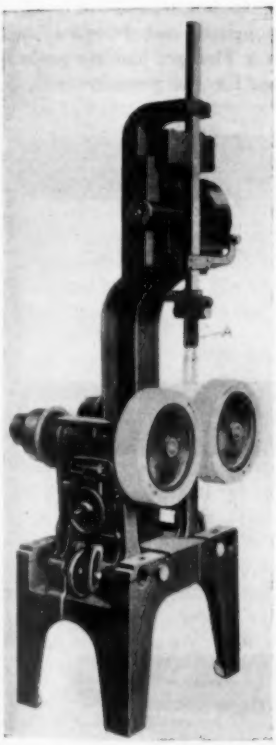


FIG. 9 VERTICAL DOUBLE-HEADER POLISHING MACHINE



FIG. 14 BICYCLE RIM POLISHED IN CONTINUOUS STRAIGHT-LINE MACHINE

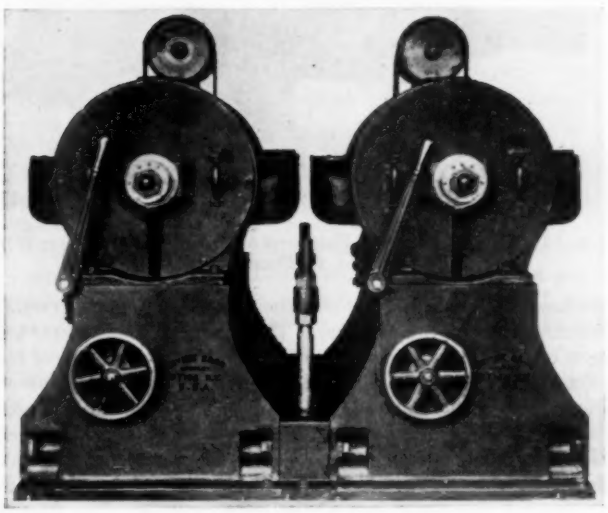


FIG. 11 STRIP-POLISHING MACHINE



FIG. 13 AUTOMOBILE FENDER GUARD POLISHED IN CONTINUOUS STRAIGHT-LINE MACHINE

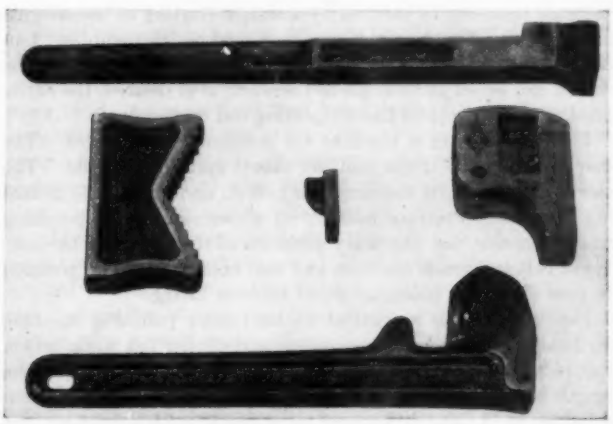


FIG. 12 STILLSON-WRENCH PARTS POLISHED IN CONTINUOUS STRAIGHT-LINE MACHINE

carried underneath the wheels which are set up with such different abrasives as may be necessary to secure the desired finish. The machine is built on the unit system, and as many wheel spindles are provided as are necessary to give the desired finish on one pass of the work through the machine. The feed belt is driven through a variable-speed drive, permitting the speed to be changed to suit the conditions of the work. The pressure of the wheel on the work can be varied by shifting the counterweights, and micrometer adjustment is provided for the wheels in order to permit of precision work.

This machine is in extensive use for the polishing of work ranging in size and contour from as large as automobile bumpers to as small as the smallest link in use in the construction of adding



FIG. 15 IRONING MACHINE, POLISHED WITH FORMED-FACE WHEELS

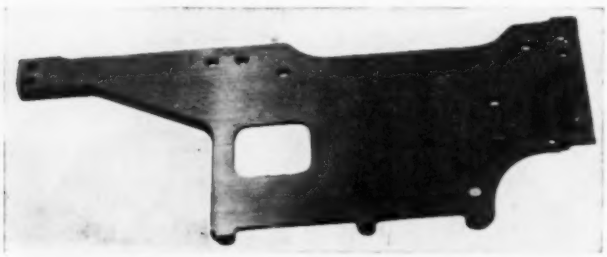


FIG. 16 TYPEWRITER SIDE FRAME—THE DESIGN OF FIXTURE WAS THE MAJOR FACTOR IN POLISHING THIS PIECE

machines. Fig. 7 shows the evolution of the continuous straight-line machine to its present form. The direct-connected motor has been replaced by a motor mounted upon the opposite side of the swinging arm from the polishing wheel, the wheel being driven from the motor by means of a Texrope drive. The base of each unit of the machine forms an oil tank, in which is a large float almost completely filling the tank. A heavy rod on the float, projecting through the base terminates in a ball-and-socket joint underneath the center line of the wheel spindle. The float is adjusted so that it bears the entire weight of the wheel and swinging arm, thus bringing zero pressure upon the work. Any desired pressure then can be secured by a simple shifting of the weights on the arm. The float also acts as a dashpot to damp any tendency to vibration on the part of the polishing wheels. The wheels can be set at an angle to the work, if so desired, the entire head swiveling around the supporting rod on the float.

Fig. 8 illustrates a machine for polishing sheet metal. The machine in question was built for sheets up to 50 in. wide. The sheets are carried by the conveyor belt *A*, underneath the polishing wheel *B*. Pressure rolls *C*, on either side of the polishing wheel, provide the necessary pressure of the work on the conveyor belt to furnish traction, and also hold the work in position to pass under the polishing wheel without lifting.

Fig. 9 illustrates a vertical double-header polishing machine for finishing putty knives, spatulas, and similar flat work, where the polishing marks should run in a direction parallel to the length of the work. The work to be finished is suspended from a holder at *A*, and is lowered by a crank motion between the polishing wheels. As the lower edge of the work approaches the line joining the centers of the wheels, a cam motion closes the two wheels upon the work. It passes down between them and is then withdrawn, being in contact with the wheels during the entire time of passage. As the work leaves the wheels the cam motion moves them apart ready for the entrance of the next piece. Usually these machines are operated in pairs, the first machine carrying the roughing wheels, and the second the finishing wheels. The machine can be arranged to stop at the end of the upward stroke, or it can be set to run continuously. With a skilled operator, continuous operation is the rule, as he can detach finished work from, and insert new work in, the holder without stopping the machine.

Fig. 10 is a machine for the finishing of axe heads, which operates somewhat on the same principle as the double-header machine just described. It is manual in its operation, instead of automatic as in the case of the double header.

Fig. 11 shows a machine for finishing strip brass. The polishing wheels are mounted on the spindles whose ends project through the large drum-shaped structure at the top of the machine. Each wheel is driven by a 50-hp. motor through a Texrope drive, and oscillation is provided on each wheel spindle. The oscillator is driven from the Texrope drive shown projecting above the top of the machine. The brass strip is fed from a coil, passes over pressure rolls underneath each polishing wheel, and is coiled by means of a friction winder at the opposite end of the machine. These coiling devices are not shown. The two handles projecting upward operate a quick release for the pressure rolls, to



FIG. 17 TIRE-VALVE PARTS POLISHED IN ROTARY TYPE OF MACHINE

permit dropping the work out of contact with the polishing wheels, if for any reason it is necessary to do so without stopping the machine. The two hand wheels at the front of the machine are for the purpose of raising or lowering the pressure rolls, to provide variations in pressure of the wheel on the work, and also to compensate for wear of the wheels.

TYPICAL MACHINE-POLISHING JOBS

A few typical machine-polishing jobs are illustrated in Figs. 12 to 17 inclusive. Fig. 12 illustrates parts of Stillson wrenches which have been finished in the continuous straight-line machine. These are typical examples of small jobs that can be handled in large quantities by the machine, and form a splendid example of

work for which the capacity of the machine exceeds the ability of the operator to load. In the case of these wrench parts, this difficulty was overcome by providing an extension end table, and putting two operators to work loading the conveyor belt. The parts were unloaded automatically as the conveyor passed over the tail pulley.

Fig. 13 is an automobile fender guard also polished in the continuous straight-line machine. The polishing wheel followed the curve of the fender guard from end to end, with the exception of a short space near the eye. The same machine also polishes automobile-bumper rails, from 16 to 18 units being employed for this purpose.

Fig. 14 represents an extreme case of straight-line machine polishing. This is a bicycle rim which was required to be polished on the inner surface. This feat was accomplished by fitting the conveyor belt of the continuous straight-line machine with a track, and mounting the rim in a fixture which held it in the proper relation to the polishing wheel while resting on the track. The movement of the conveyor belt and its track caused the rim to rotate in the fixture while in contact with the polishing wheel, and thus completely polished it around the entire circumference. Two operations were necessary, formed face wheels being used.

Figs. 15 and 16 are other examples of work that is adaptable to a straight-line machine. Fig. 15 shows the shoe for an ironing machine which required polishing on the inner concave face. This was accomplished by means of formed wheels. Fig. 16 is the side frame of a typewriter, which was polished with hard compress-leather wheels. This piece is a first-class example of a job wherein the proper design of the fixture was essential to the final solution of the problem. The piece is a difficult one to hold under the polishing wheel, and at the same time obtain the precision that is required in the finishing of this particular part.

Fig. 17 shows a job that was done in the four-spindle machine illustrated in Fig. 3. The upper piece is an automobile-tire-gage casing, while the two lower ones are automobile-tire valve caps.

Economics of Machine Polishing

Little has been said heretofore about the economics of machine polishing. It is difficult to lay down any general rule covering this phase for the reason that many different factors enter into it. Where the machine is adapted to the work and can eliminate hand operations, and where there is sufficient production to keep the machine busy a fair proportion of the working day, machine polishing usually will prove economical. Where, however, the time required to load and unload a piece for machine polishing is equal to the time in which a polisher could finish the work by hand, machine polishing would prove uneconomical. Where conditions are right, however, the substitution of machine polishing for hand work represents a tremendous saving in the cost of manufacture. Each case should be investigated on its merits, and the possible economies worked out before the decision is made either for or against machine polishing.

As an example of the saving possible by machine polishing, in one installation a single polishing machine displaced 30 hand polishers. The machine required four men to operate it, three of these men being used for loading it. In the case of the ironer shoe, Fig. 15, with hand polishing the number produced per man was three shoes per day of eight hours. The polishing machine, operated by two men, produces a completely finished shoe every three minutes.

Up to within a few years ago, the polishing department in nearly every industry was regarded, as has been very aptly expressed, as the "Cinderella of the works." Within the past few years, however, it has been recognized that there is no department of the manufacturing plant in which greater savings can be effected per dollar of investment than in the polishing department. There is a vast amount of engineering and research work to be done before polishing is on the same technical plane as other departments of the metal-working industry. It is, however, a field which merits the study and best efforts of the engineering profession.

Discussion

Floyd T. Taylor.[3] The paper makes a notable contribution to the literature on polishing, at a time when such contributions ought to be very welcome. A number of points naturally arise in the minds of those who have had contact with hand or machine polishing.

Polishing, at the present time at least, can be described as an art guided by a science. The line of demarcation between the art of polishing and the scientific guide now being applied to it is somewhat indefinite.

From the mechanical engineer's standpoint, polishing may be considered as the transfer of a portion of the energy in a revolving polishing wheel into luster on the work, and it should be noted that this transfer takes place across a coating of adhesive and abrasive. Heat losses, inherent with a transfer of energy, must be dissipated from the face of the polishing wheel and from the work. The polishing wheel itself is a poor conductor of heat, and consequently there can be little relief through the conduction of heat away from the face by the polishing wheel. Means of cooling the face of the wheel are therefore of paramount importance, and yet up to the moment, except as on such types of polishing machines as employ a "coolant" on all the wheels, little attention has been paid to this point.

It is also obvious that an improvement in the efficiency of polishing, through the use of abrasives which cut and do not drag, will reduce the heat losses at the face of the polishing wheel.

There are two distinct actions in polishing—one a cutting action which tears off metal and the other a burnishing action which flows the metal. The cutting action is a high-speed operation at light pressure. The burnishing action is a slow-speed operation at considerably heavier pressure.

The author has used the word "color" without defining color as applied to polishing. Color may be defined as a condition of the surface where the average depth of the defects has been reduced to a point which causes almost all of the light striking the surface to be totally reflected. Through the use of a photoelectric cell, definite measurements of color can now be made, and therefore the degree of color need no longer be a matter of individual opinion, based on eyesight alone.

The author states that the power required to drive a hand-operated polishing lathe is no criterion for the size of motor which should be installed on a polishing machine. This statement leads to the observation that machine polishing is very much more of a continuous load than hand polishing can possibly be. In addition to a difference in power requirements, there is a very considerable difference in the use and behavior of wheels and abrasives. A further point in connection with power supply is that the polishing wheel, whatever its linear speed may be, should be run at a constant linear speed, and therefore polishing machines will be much improved when synchronous motors are applied to them. Where it is necessary to use induction motors, the low-resistance end-ring type should be used, since these motors give a considerable closer speed regulation than the so-called high-torque types.

The author has aptly stated that the condition of the material when it comes to the polishing room has a vital and important

[3] The Bullard Company, Bullard-Dunn Process Division, Bridgeport, Conn.

bearing on the polishing operation. When iron or steel is covered with black magnetic heat scale, the first polishing operations are not polishing operations at all. They are very costly and inefficient grinding operations done with polishing wheels. Polishing wheels should never be required to do grinding. Moreover, grinding, sand-blasting, or pickling, plain or electrified, will not remove this black heat scale thoroughly and successfully. Electrolytic treatments have been devised, however, which remove this black scale completely and cheaply.

The author cites the case of polishing bumper bars, and in this connection it should be pointed out that in one of the large bumper-bar plants, where originally an effort was made to polish bumper bars before the black heat scale had been completely removed, wheels coated with these abrasive grain sizes were used, in succession, Nos. 30, 60, 90, 120, 150, 180, and grease. When, however, the bumper bars were completely free from scale, they were satisfactorily polished, beginning with the No. 90 wheel.

The elimination of polishing wheels set up with the coarser grain sizes makes another saving, namely, that of frequent wheel changing, because coarse grains are much more readily ripped from the face of the wheel than the fine grains.

In discussing the condition of the finished surface, the author states that the finish desired determines the number of wheels required. This statement should be qualified, because actually the number of grain sizes required is determined by a double consideration; namely, the condition of the surface at the beginning of the polishing operation and the final finish desired.

In speaking of the density of polishing wheels it should be pointed out that no measure of density has been standardized; therefore, what one might think is a hard wheel another might think is a soft wheel. A standard of wheel density should be sought and established. Furthermore, the density may vary considerably with the reduction of diameter as the wheel wears down and also with the aging of the adhesive used to hold laminated wheels together.

Under "Variables of the Abrasive," the author states that these variables are discussed generally in the fore part of the paper, but variables of the abrasive are so important that it would seem well to have a discussion of them and their effect on polishing in a section of the paper set aside for this purpose alone. The first question that arises is, Are the present standard grain sizings correct for polishing and are the tolerances within these sizings close enough?

Since it would be a great advantage if wheels could be run under water instead of oil as a coolant, the availability of the so-called water-dispersed rubber cements for facing polishing wheels should be checked. A final point in connection with the variables of abrasives on machines is this: Not nearly enough care has been taken on automatic machines thus far produced to see to it that the coarse grains detached in operation are completely removed from the work before it passes under a wheel of finer grain set-up.

Under "Peripheral Speed of the Polishing Wheel," the author states that, where it is possible, polishing wheels should be run at a surface speed of 7500 ft. per min. He gives as a reason that this speed is about the maximum which can be used on account of the holding power of glue. He further states than an increase of speed would increase the efficiency of cutting momentarily, but that this cutting efficiency would promptly become lower due to overheating. No mention is made at this point of the fact that pressure is also a factor. Furthermore, it seems worth while to point out that the polishing wheels generally used are 12, 14, 16, or 18 in. in diameter. At 7500 ft. of surface speed these diameters correspond to the following revolutions per minute:

12 in.	2385
14 in.	2045
16 in.	1790
18 in.	1590

The interesting point about these rotating speeds is that a great percentage of the modern polishing machines are run from 60-cycle circuits, and therefore the 16-in. wheel is the only one that can be run direct-connected if this linear speed of 7500 ft. per min. is to be maintained.

Under the heading of "Pressure of the Wheels on the Work," a statement should be made that the pressure should remain constant, and furthermore that it should be light to give the abrasive a real opportunity to cut. When a wheel is used for burnishing, however, the pressure must be heavy to flow the metal.

In the section on "Provision for Oscillation" the statement is made that unless the wheels oscillate they are likely to track, and when they track, the tendency is to deepen the scratches instead of cutting them out. It is doubtful if the scratches will be deepened on account of the reduction in grain size of the wheel facings as the work passes from one to another.

Under "Vibration of Machine or Work," it should be pointed out that soft wheels are freer from vibration than hard wheels, and this is particularly true of the cloth and canvas types.

Under "Machine Design," it should be noted that up to the present time little or no attention has been paid to designs which make the changing of wheels a quick and easy job. Too little attention has been paid to the protection of the machinery from the damage caused by its own abrasive, and too little attention has been paid to the cooling of the surface of the wheels.

Author's Closure

The author finds himself in agreement with practically all of the statements made by Mr. Taylor. All of the points raised are fruitful suggestions for further experiment and development. It is well recognized that means of dissipating heat from the polishing wheel would greatly prolong its life and increase its cutting efficiency. The engineering difficulties of accomplishing this, however, have put it some distance in the future as a practical addition to polishing machines.

So far as the use of the photoelectric cell for the determination of finish is concerned, this device has not proved as useful as may have been anticipated. It has failed to detect the difference between surfaces of widely different finish. After numerous experiments, this method of measuring finish or "color" reluctantly has been abandoned. The method developed by R. E. W. Harrison and described at the Annual Meeting of the Society in December, 1930, gives more promise of success, and experiments are still being carried out along these lines.

The author is in complete agreement with Mr. Taylor's statements concerning the removal of scale, the standardization of grain sizes and wheel densities, protection of machines and work from abrasive dust, and the regulation of pressure of wheels on the work. These and other matters are all highly desirable developments for the future. It should be remembered, however, that commercial considerations also must be taken into account. The polishing industry has not, as a whole, been educated to the point where it realizes that refinements are a profitable investment, and where it is willing to pay for these refinements.

MSP-53-10

Transmission of Torque by Means of Press and Shrink Fits

By J. W. BAUGHER, JR.,[1] PITTSBURGH, PA.

The most important formulas for calculating the stresses existing in press and shrink fits are presented, together with charts for readily obtaining numerical results when steel shafts and cast-iron or steel outer members are used. Equations and charts also are given for determining the pressure existing between fitted parts. From the records of a number of fits observed by the author and others, values of coefficients of friction are suggested for use with the pressure in computing the probable tonnage required in making a fit and the torque capacity of a fit when no key is used. Equations are given for the tonnage and torque capacity.



WITH the increasing use of press and shrink fits without keys for the transmission of power and the more general realization that high localized stresses are produced at the corners of keyways and keyseats, such fits have become of considerable importance. In some cases keys are necessary to locate parts in particular positions on their shafts, but there are many instances in which keys may be dispensed with, and press or shrink fits, perhaps slightly heavier than would be used with keys, will be adequate to transmit the torque. The elimination of keys, keyways, and keyseats reduces manufacturing costs, removes the danger of failure from concentration of stress, and permits the use of smaller shafts than would be possible if they were keyseated.

For street-railway gears, which are subjected to reversing loads, it has been found that keys become pounded over on the corners and eventually work loose, and the majority of these gears are pressed on the axles without keys.

It is certain that keys quite often serve no useful purpose and are a constant source of trouble.

A great deal has been written about press and shrink fits; yet unfortunately the material presented is not in too good agreement and is not always in such a form that it can be readily used. In this paper the more important formulas and other data will be given in a manner that the author has found convenient to use.

---

[1] Engineering Department, Westinghouse Elec. & Mfg. Co. Mr. Baugher was graduated in 1924 from the Virginia Polytechnic Institute with the degree of B.S. in mechanical engineering and a year later received the degree of M.E. He then entered the graduate student course of the Westinghouse Electric and Manufacturing Company and attended the mechanical design school. Following this training he was connected with the mechanical section of the company's Railway Equipment Engineering Department at East Pittsburgh. Since 1929 he has been a member of the Engineering Department of the Nuttall Works of the Westinghouse Elec. & Mfg. Co.

Contributed by the Machine Shop Practice Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Stresses

A number of formulas based on the theory of thick cylinders subjected to internal pressure are in use for the determination of the stresses in the outer ring of a press or shrink fit. Among these formulas are those of Lamé, Clavarino, and Birnie. According to several authorities Lamé's theory takes no account of lateral contraction as expressed by Poisson's ratio. The equations of Clavarino are derived for a cylinder with closed ends, and while they have been used considerably in the design of hooped guns, they are not applicable to the usual case of a machine fit. Birnie's formulas are for cylinders with open ends, and also are employed for the design of large guns. Another formula used to some extent is Barlow's. This expression, however, applies only to thin cylinders. The subsequent equations likewise are attributed to Lamé and are probably the best for the determination of these stresses.

When a hub or ring of appreciable thickness relative to its diameter is forced or shrunk on a solid shaft of larger diameter than the bore of the ring, two principal stresses are produced in the ring: a tangential tensile stress S_t and a radial

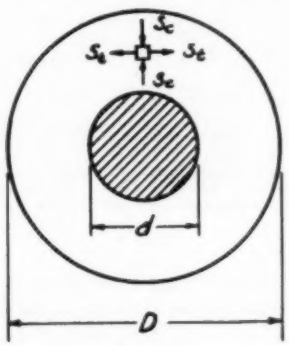


Fig. 1

compressive stress S_c. (See Fig. 1.) If the shaft and ring are of the same material, the maximum value of the tangential or hoop stress, which occurs at the bore, is

$$S_t = \frac{Ee}{2}\left[1 + \left(\frac{d}{D}\right)^2\right] \quad \text{[1]}$$

where E is the modulus of elasticity of the material, e is the fit allowance per inch of shaft diameter, d is the shaft diameter, and D is the outside diameter of the hub. If the shaft and ring are of different materials, the maximum tangential stress at the bore is

$$S_t = \frac{e}{\frac{1}{E_1} + \left[\frac{V_1}{E_1} + \frac{1-V}{E}\right]\left[\frac{1-\left(\frac{d}{D}\right)^2}{1+\left(\frac{d}{D}\right)^2}\right]} \quad \text{[2]}$$

where E and E_1 are the moduli of elasticity of shaft and ring, respectively, V and V_1 are Poisson's ratio for shaft and ring,

and e, d, and D are the same as in Equation [1]. The maximum radial stress, also at the bore, when the material in the shaft and ring are the same is

$$S_e = \frac{Ee}{2}\left[1 - \left(\frac{d}{D}\right)^2\right] \quad \ldots\ldots [3]$$

When the material in the shaft and ring are different, the maximum radial stress is

$$S_e = \frac{e}{\frac{1}{E_1}\left[\frac{1 + \left(\frac{d}{D}\right)^2}{1 - \left(\frac{d}{D}\right)^2} + V_1\right] + \frac{1 - V}{E}} \quad \ldots\ldots [4]$$

where the symbols have the same meaning as in the preceding equations.

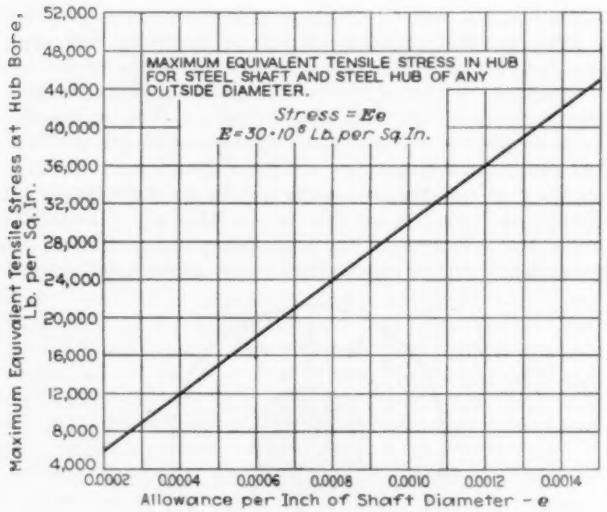


FIG. 2

The statement will be found quite often that the tangential stress, which is always greater than the radial stress, is the stress upon which the safety of the fit depends. In any case where more than one principal stress exists, however, these stresses must be combined according to one of the "theories of strength," and the equivalent stress thus found will be the criterion of strength of the part.

For steel and other ductile materials the maximum shear theory is commonly accepted at this time as being most nearly correct. According to this theory the maximum equivalent tensile stress in the ring due to the combined effect of S_i and S_e is

$$S = Ee \quad \ldots\ldots [5]$$

This stress, which is at the bore, is also the maximum stress in the fit; therefore, this equation determines the fit allowance that can be used when the same ductile material is used in both ring and shaft. It will be noted that when the frequently specified allowance of 0.001 in. per inch of shaft diameter is used the maximum equivalent tensile stress for steel is 30,000 lb. per sq. in., which approaches the yield point of annealed medium carbon steel. If the yield point is exceeded, a permanent set will be produced and a light fit will result. Stresses much above the yield point will cause the outer ring to fracture.

For brittle materials, such as cast iron, it is generally agreed that the maximum stress theory is the correct theory of strength. The critical stress according to this theory is the maximum principal stress, which, as has been seen, is the tangential stress at the bore of the ring as given by Equation [2]. If the shaft is steel and the ring is cast iron and it is assumed that Poisson's ratio is the same for steel and cast iron and that the modulus of elasticity of cast iron is half that of steel, this equation reduces to the simpler form of

$$S = \frac{Ee\left[1 + \left(\frac{d}{D}\right)^2\right]}{3 + V + (1 - V)\left(\frac{d}{D}\right)^2} \quad \ldots\ldots [6]$$

This stress is the greatest stress in the fit. For an allowance of 0.001 in. per inch of shaft diameter, this maximum stress is between 9000 and 15,000 lb. per sq. in., depending upon the ratio d/D. For ordinary cast iron this stress is near the breaking strength, but for the high-strength cast irons now being produced it is well within the limits of safety.

For a complete analysis the stress due to the fit must be combined with the stress at the bore due to the transmission of power, which stress is difficult to determine and is usually quite low, and for very high speeds with the stress due to centrifugal

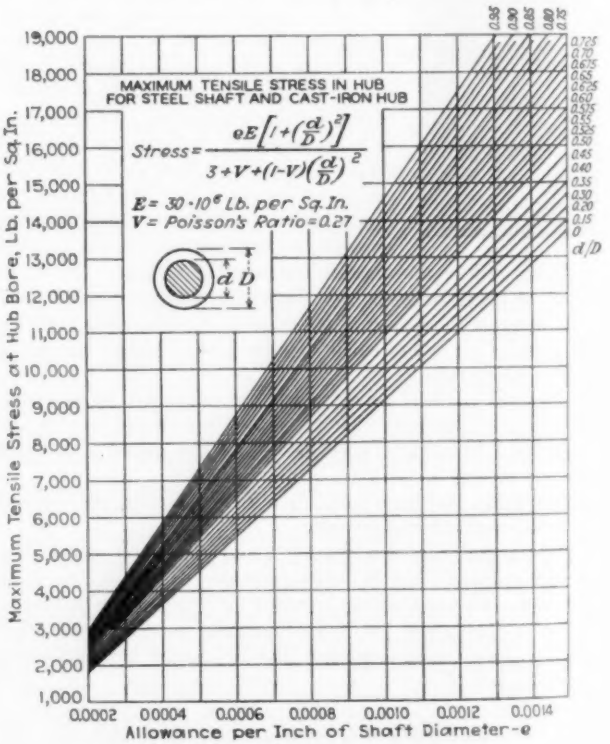


FIG. 3

force, which tends to relieve the pressure between the fitted parts and destroy the effectiveness of the fit. The stress due to centrifugal force[2] will usually be of no consequence, however.

In most instances it will be found that an allowance greater than 0.001 in. per inch of shaft diameter is unnecessary to provide capacity for transmitting the torque.

If a thin ring, such as a bushing, is shrunk or pressed on a shaft or into a thicker ring, the equations given heretofore,

[2] S. Timoshenko and J. M. Lessells, "Applied Elasticity," first edition, p. 303.

which are derived from the theory of thick cylinders, need not be employed. In this case the theory of thin cylinders will be used. This theory assumes that all the deformation occurs in the thin ring and that the tangential stress is uniform over its cross-section. This tangential stress determines the safety of the ring and is $S = Ee$ whether the ring is on a shaft or inside another ring. In the first case S will be a tensile stress and in the second case a compressive stress. This formula is the same as Equation [5].

Figs. 2 and 3, which are charts plotted from Equations [5] and [6], respectively, are useful for quickly determining the critical stresses. If the ring is a gear, a flywheel, or any part in which there is metal outside the actual hub, such as arms and a rim, an equivalent value of D larger than the hub diameter must be used in Equation [6] and the chart. The error that results from estimating this equivalent outside diameter will be usually only a few per cent.

Torque Capacity

The torque that a press fit or a shrink fit can transmit is obviously the product of the unit pressure existing between the shaft and the ring, or hub, the coefficient of friction between the two surfaces, and the area of the surfaces in contact.

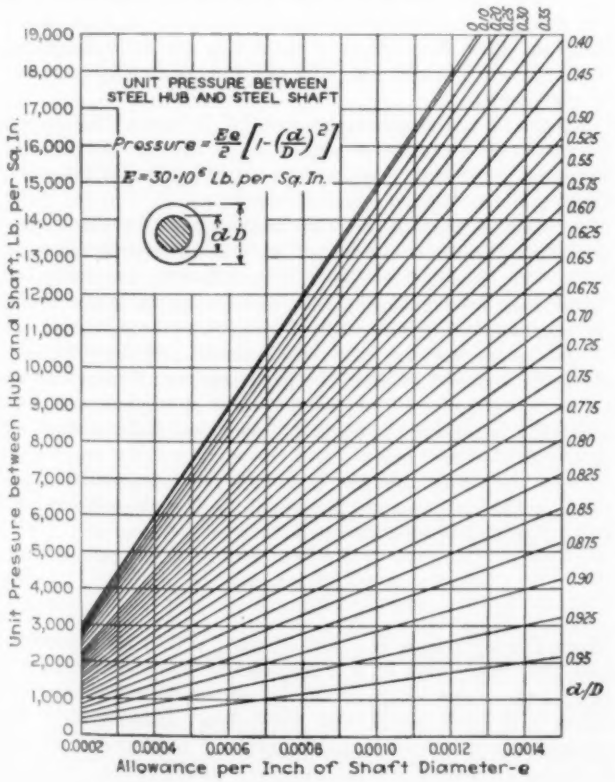


Fig. 4

The pressure has the same value as the radial stress at the bore and may be determined from Equation [3] or Equation [4]. If the shaft and ring are of steel, Equation [3] must be used. The unit pressure is therefore

$$P = \frac{Ee}{2}\left[1 - \left(\frac{d}{D}\right)^2\right] \quad \ldots\ldots\ldots\ldots\ldots [7]$$

If the shaft is steel and the ring cast iron, Equation [4] becomes

$$P = \frac{Ee\left[1 - \left(\frac{d}{D}\right)^2\right]}{3 + V + (1 - V)\left(\frac{d}{D}\right)^2} \quad \ldots\ldots\ldots\ldots [8]$$

if the same assumptions are made as in deducing Equation [6]. The charts, Figs. 4 and 5, plotted from these two equations will be found convenient in determining pressures. As with Fig. 2, an equivalent hub diameter must be used for D when the diameter of the actual hub is not the outside diameter of the outer member.

If the shaft is hollow, Equations [7] and [8], and also Equations [5] and [6], are not exact. If the diameter of the hole is less than 25 per cent of the shaft diameter, however, the

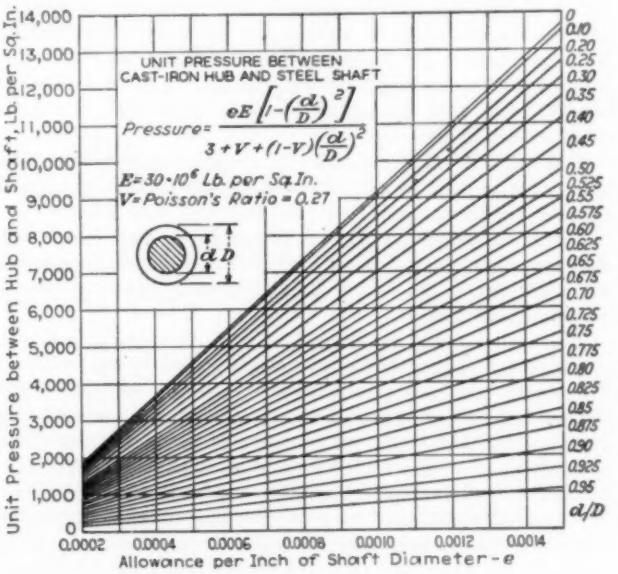


Fig. 5

shaft can be considered solid and these formulas be used with a maximum error (when d/D is zero) of about 7 per cent. The error decreases as d/D increases.

A number of factors affect the coefficient of friction between the fit surfaces, and among these the most important are the finish of the surfaces, the roundness of bore and shaft, the straightness or taper of bore and shaft, and, in the case of a press fit, the lubricant.

Views held on the effect of the finish of the fit surfaces are in direct contradiction. One contention is that the better the finish the lower the coefficient of friction. The opposite view borne out by the experiments of N. N. Sawin[3] is that the coefficient is larger with finer finish because the minute ridges on the metal are smaller and the actual contact area more nearly equals the theoretical area of the surfaces in contact. Values that the author calculated from the results of more than a hundred press fits in which one of the members was finished by grinding and the other either by grinding, finish-turning, or rough-turning show practically no effect of this difference in finish. It is to be expected, however, that there is a limit beyond which the ridges on rough-finished parts will shear off during assembly and a low tonnage will result. It is probable, also, that the force required to press the members apart will be less if the finish is poor than if it were good, even though the

[3] N. N. Sawin, *American Machinist*, May 31, 1928, p. 889.

force required to press them together were the same in both cases. For this reason and because grinding allows closer tolerances and more accurate measurement and seems to produce greater uniformity of results, ground surfaces are preferable.

After a shaft is pressed into a hub, the lubricant in time tends to squeeze out and the metal surfaces to freeze together, and a greater force will be required to press the shaft out if the fitted parts are allowed to stand for several days. A greater force will be required also to press a shaft out of a ring that has been shrunk on than if a press fit has been made with the same allowance. This fact is due, probably, to some lubricant remaining between the surfaces of a press fit and to the wearing of a path in the metal when the shaft is forced into the hub.

One of the laws of friction for partially lubricated surfaces is that the coefficient of friction increases with increase of pressure per unit area. The pressures between fitted surfaces in practice range from about 2000 lb. per sq. in. for very light fits to more than 15,000 lb. per sq. in. for heavy fits. For this range variations in pressure seem to exert no effect on the coefficient of friction between the surfaces—probably because the pressures are so high.

The hardness of the metal, its chemical composition, and whether it is cast, forged, or rolled appear to have much less effect on the friction coefficient than the other variables.

Actual coefficients of friction in pressing a shaft into a hub may vary from about 0.040 to 0.340. The coefficient in pressing the shaft out of the hub is usually greater with a minimum value of about 0.10.

The author found coefficients of 0.10, 0.11, 0.11, and 0.14 in pressing $5^1/_4$-in. diameter oil-quenched steel shafts of about 175 Brinell hardness into hardened cast-steel spiders of 160 to 260 Brinell hardness. When the shafts were pressed out after the fits had stood for 24 hours or more, the respective coefficients of friction were 0.14, 0.14, 0.12, and 0.15. Coefficients in pressing apart shrink fits were 0.16 and 0.17, about 25 per cent greater than the corresponding values for press fits made with the same shafts and hubs. In another series of 15 press fits for which the author recorded data, the coefficients of friction in pressing in the shafts ranged from 0.075 to 0.25, with an average value of 0.15. In these cases the shaft was $6^1/_{16}$ in. in diameter and of medium carbon steel quenched in oil. The ring was a pinion of the same material treated in like manner. The hardness of both shaft and pinion was about 170 Brinell. As in the cases cited heretofore the lubricant used in making the press fits was a mixture of white lead and light machine oil and the fit surfaces were ground. In all cases, also, great care was taken to obtain correct measurements of shaft and bore, readings being taken on a number of diameters in several planes and the micrometer calipers being checked against each other and against a standard gage.

In tests[4] with a tapered, ground, hardened steel shaft of $6^1/_2$ in. mean diameter and a hardened steel pinion with a ground bore the coefficients of friction in forcing out the shaft were 0.15, 0.17, and 0.21. The coefficients in forcing out the shaft after shrink fits had been made with the same allowances were 0.21, 0.22, and 0.23, respectively. The average shrink fit coefficient is about 25 per cent above the average press fit value.

In extensive researches on force fits N. N. Sawin[5] pressed long, tapered steel shafts through thick steel and cast-iron rings, thus gradually increasing the fit as the shaft was forced through the ring. The shafts and ring bores were ground, and when the fits were made, rape oil was used as a lubricant. While few numerical values were published, certain test data sent to the author yielded coefficients of friction for hardened steel shafts and rings of 0.054 to 0.22, with an average of 0.11. The diameters of the shafts used were 0.315 to 8.2 in. In some cases the coefficient of friction decreased as the shaft progressed through the ring; in other cases it remained practically the same.

Records of over two-hundred press fits published by C. F. MacGill[6] gave coefficients of friction for steel shafts and steel hubs ranging from 0.077 to 0.33 and for steel shafts and cast-iron hubs from 0.053 to 0.30, the average of all values being 0.17. The finish of the surfaces was not mentioned.

Coefficients of friction were computed by the author from data on 123 press fits, about half of which were made with both bore and shaft ground and the remainder with either the shaft turned and the bore ground or the shaft ground and the bore rough or finish-bored. Shafts were of heat-treated medium carbon or alloy steel, and outer members, which were gears or pinions, were of heat-treated medium carbon cast, rolled, or forged steel. The values of the friction coefficient varied from 0.03 to 0.25, the lower values being undoubtedly due to yielding of the soft steel around the bore because of too large fit allowances. The average coefficient where either bore or shaft was not ground was 0.091, while for ground shafts and bores the average was 0.086, about 6 per cent lower. It would seem, therefore, that the method of machining the fit surfaces has little effect on the coefficient of friction and the tonnage necessary to make the fit.

Except for the classic experiments of Wilmore,[7] the author knows of only one series of tests of fits in torsion. Four shrink fits were made[8] with shafts $2^1/_2$ in. in diameter, and the coefficients of friction ranged from 0.25 to 0.33.

A general average of the coefficients of friction realized when pressing in shafts would be probably in the neighborhood of 0.15. For pressing out shafts the coefficients are somewhat higher than for pressing in, and for twisting the shafts in the fits the values are greater still. Therefore, in the determination of the torque that a press fit will transmit and the force that should be required to press the parts together, it is reasonable to use a value of 0.10 for the coefficient of friction in torsion and 0.075 in making the fit, and these values have been found to give satisfactory results.

Since the torque that the fit is capable of transmitting depends on the coefficient and the area of the fit surfaces in contact, this torque may be expressed as

$$T = \frac{\pi}{2} f P L d^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots [9]$$

where T is the torque in pound-inches, f is 0.10 for a press fit and 0.125 for a shrink fit, P is the pressure obtained from Figs. 4 or 5, L the actual length of the fit, and d the shaft diameter. While it is fairly certain that the values suggested for f are conservative, they are based largely on coefficients realized in pressing shafts and hubs together; consequently, it is advisable to allow a safety factor in Equation [9]. However, for the common fit allowances and the usual proportions of bore, hub length, and hub thickness it will be found that there is ordinarily a generous factor of safety; in fact, in medium and large-size machinery it is surprising how seldom a key is necessary to transmit the power.

[4] This work was done by J. G. Ritter, Mechanical Engineer, Westinghouse Electric and Manufacturing Company.

[5] N. N. Sawin, loc. cit.

[6] C. F. MacGill, *A.S.M.E. Journal*, November, 1913, p. 1657.

[7] John J. Wilmore, *American Machinist*, February 16, 1899.

[8] These experiments were made by S. Timoshenko and L. S. Jacobsen, then of the Engineering Department, Westinghouse Elec. & Mfg. Co.

In a similar manner the average force in tons required to force the parts together may be stated as

$$F = \frac{\pi}{2000} f_1 PLd \ldots\ldots\ldots\ldots\ldots [10]$$

In this equation 0.075 may be used as an average value of f_1. Values of f_1, however, may vary from a minimum of about 0.05 to a maximum of about 0.30, so that the tonnages also will vary within wide limits. P, L, and d are the same as for Equation [9], and it will be observed that the force of pressing is in direct proportion to L, the length of the fit. In case a key has been used, the area πLd must be reduced by the area of the keyway.

With the equations presented every important question concerning a fit can be determined in advance. Furthermore, if the force required to make a press fit be recorded, an excellent estimate can be had of the torque that the fit will transmit. It is to be hoped also that more experimental data will soon be available on coefficients of friction for fits subjected to torque; and if so, perhaps designers will be inspired with more confidence in press fits and shrink fits and the use of keys will be confined to the few cases of necessity.

Bibliography

Timoshenko and Lessells, "Applied Elasticity," first edition, pp. 249–255.

Sanford A. Moss, *General Electric Review*, April, 1917, p. 320.

Discussion

D. V. Waters.[9] This paper is one that we can use in our every-day life. However, where mating parts are made in quantities not sufficient to justify the use of gages, it would seem that the safest plan would be to use keys, as we cannot always depend upon the man in the shop giving us the fit we should have. It is true that the stress concentration due to keyways is undesirable, but it may be somewhat minimized if they are cut with a disk-type cutter instead of an end mill, and if care is taken that the keyways do not extend clear to any shoulder that may be turned upon the shaft.

This is not intended in any way as a criticism, as for parts made with gages to interchangeable limits a good shrink or press fit is no doubt entirely adequate to transmit the torque required.

E. O. Waters.[10] Designers have grown used to employing the press fit and shrink fit as a means of securing rigidity in an assemblage of parts, without regard to the possibility of transmitting torque. It is fortunate that the author of this paper has seen fit to call attention to this possibility, and there can be no doubt that his straightforward analysis of the stresses and pressures involved, leading up to a simple method for calculating the torque capacity of one of these tight fits, will prove very useful in future design and construction.

However, it appears that the author, in his enthusiasm for press and shrink fits, has treated the keyed fastening with somewhat less than justice. Is it generally true that the elimination of keys and keyways reduces manufacturing costs? It would seem self-evident that, if the fit is relied upon to transmit the torque, the manufacturing tolerances must be held within much closer limits, and special equipment will be needed to make the assembly. Moreover, the process of disassembly becomes more difficult.

In the second place, what may be called the "space efficiency" of a keyed joint is much greater than that of a press fit. Assume for example, that a steel hub of diameter $2d$ is keyed to a steel shaft of diameter d with a steel key of width $^1/_4 d$. Then, if S_s represents the shear stress in the key, the torque transmitted per inch of length is

$$^1/_4 d \times {}^1/_2 d \times S_s = \frac{d^2 S_s}{8}$$

Compare with this a friction fit in which the shear stress at the bore of the hub also equals S_s. Then, in the author's notation,

$$S_s = {}^1/_2 Ee \text{ and } S_s = S_s[1 - (d/D)^2] = {}^3/_4 S_s$$

With a coefficient of friction of 0.10 and a factor of safety of 2 against slippage (roughly equivalent to the factor of safety against overstress that would probably be used in the case of the key), the torque per inch of length is

$$^1/_2 \pi \times \frac{0.10}{2} \times {}^3/_4 S_s \times d^2 = \frac{d^2 S_s}{17}$$

If the hub is made of cast iron with an assumed maximum tensile stress of $^1/_2 S_s$, then

$$S_s = \frac{1 - (d/D)^2}{1 + (d/D)^2} \times {}^1/_2 S_s = 0.3 S_s$$

and the torque per inch of length is

$$^1/_2 \pi \times \frac{0.10}{2} \times 0.3 S_s \times d^2 = \frac{d^2 S_s}{42.5}$$

In one case, the friction fit would have to be over twice as long, and in the other, over five times as long as the keyed fit, to transmit the same torque. This would indicate that the force or shrink fit is best suited to transmitting torque when the hub takes only a small portion of the maximum torque capacity of the shaft—a condition excellently exemplified in the street-railway application cited by the author.

A. M. Wahl.[11] The author deserves credit for presenting a fairly simple means for evaluating shrink and press-fit allowances, together with convenient graphs for the use of the practical designer. In looking over the paper, the writer noted the statement on the second page to the effect that, if the yield point of the material were exceeded, a light fit would result. Assuming that we have a ductile material, so that yielding occurs under practically constant stress, it is not quite clear to the writer just why this loss of fit would occur. Possibly an explanation of this phenomenon would involve plastic after-effects or the shape of the stress-strain diagram on removal of load. The writer was also wondering whether this loss of fit due to yielding had actually been observed in practice, although it appears to be the general opinion among engineers that this will occur. Of course, where brittle materials, such as cast iron, are involved, it is clear that too high a stress will result in fracture, with consequent loss in fit. If this loss of fit actually occurs for steel shrink or press fits, when the allowance is such that the yield point is exceeded, the writer would be interested in obtaining a rational explanation.

Sanford A. Moss.[12] The formulas for stress which the author uses are all correct, regardless of any comparison with Lamé, Clavarino, and Birnie. However, as an actual fact, the formulas pro-

[9] Development Engineer, in charge of Machine Design Division, Western Electric Company, Chicago, Ill. Mem. A.S.M.E.

[10] Associate Professor of Mechanical Engineering, Yale University, New Haven, Conn. Assoc-Mem. A.S.M.E.

[11] Research Engineer, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Jun. A.S.M.E.

[12] Mechanical Engineer, Thomson Research Laboratory, General Electric Company, Lynn, Mass. Mem. A.S.M.E.

posed by those three gentlemen are also all correct and in agreement with Mr. Baugher's formulas. All of the formulas take proper account of the lateral contraction expressed by Poisson's ratio. Lamé's formulas only go as far as giving internal stresses. The Birnie final form uses the stresses as given by Lamé or Mr. Baugher, but adds them together so as to obtain a criterion for failure according to the maximum strain theory. The Clavarino final form is for a cylinder with internal pressure with heads, and also on the basis of the maximum strain theory. Mr. Baugher gives reasons which seem to be correct for using the maximum shear theory instead of the maximum strain theory. So far as is known, this was applied to force fits for the first time in the *General Electric Review* paper which Mr. Baugher mentions. A rational derivation of all of these formulas is given in a paper, "Increase of Bore of High-Speed Wheels by Centrifugal Stresses," Trans. A.S.M.E., 1912. This paper gives an explicit and easily followed demonstration of two fundamental principles applying to any sort of stresses in thick cylinders, called Lamé's laws. These are:

(1) The sum of the values of radial and tangential stresses at any radius has a constant value.

It follows that the longitudinal strain (which is the sum of the lateral contractions due to radial and tangential stresses) is the same at all radii.

(2) The difference between the values of the radial and tangential stresses at any radius varies inversely as the square of the radius.

This statement of Lamé's laws is based on the convention that tensile stresses are positive and compressive stresses negative. On this basis, all of the author's compressive stresses should have a minus sign, which seems to me to be the most consistent way of handling the matter. The writer has not seen Lamé's original paper, but has understood that he does not give a thorough derivation of his laws. The laws are correct, however, based on the use of Poisson's ratio, and, as stated, are rationally deduced in the 1912 A.S.M.E. paper.

Mr. Baugher is of course aware that the transmission of torque by means of a shrink fit, as he proposes, is standard practice in many cases, and the advantages he claims for it have been found to be true. However, Mr. Baugher introduces a number of numerical values which have not been before brought together. It would be desirable if he would give a value of f for his Equation [9], which has a proper factor of safety for a specified case, so that Equation [9] will give a value of torque which will be transmitted with a factor of safety proper in Mr. Baugher's judgment.

JOHN MANSA.[13] The author presents formulas and graphs for determining the maximum tensile stress in a hub and the radial pressure per unit surface of a fit. The formulas are clearly explained and the graphs are very convenient to use. The paper is therefore a help for the designing engineer.

However, when one reviews his friction data, it will be seen that there is a rather discouraging variation in the observed friction coefficients. The writer is of the opinion that this might be because the calculation of the friction coefficients have been based on assumptions which do not represent true conditions.

An example will show more clearly what is meant than an abstract treatment.

In Fig. 6, S is a shaft fitted in a hub H. The fit is transmitting a torque T. Mr. Baugher assumes that each unit of the fit surface is transmitting an equal part of the torque T. As the internal torque in the shaft is T at the beginning of the fit (section A-A) and zero at the end of the fit (section B-B), then the average torque in the part of the shaft within the fit is $^1/_2\ T$. The angular distortion of section A-A relatively to section B-B will be

$$\frac{^1/_2\ T \times L}{G \times \pi/_{32} \times d^4} \text{ (radians)}$$

where T is the total torque transmitted through the fit, L the length of the fit, d the diameter of the fit, and G the shearing modulus of elasticity for the shaft material.

This proves that for the assumed torque distribution there is a definite angular deformation of the shaft between the sections A-A and B-B. If now, and that will practically always be the case, the elasticity of the hub is different from that of the shaft, then a relative movement between the shaft and the hub must take place somewhere in the fit. But a sliding in some parts of the fit will destroy the joint, and therefore the assumption of uniform torque transmission cannot be correct.

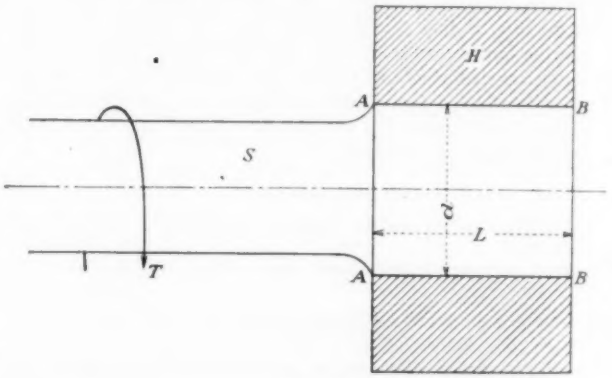


FIG. 6

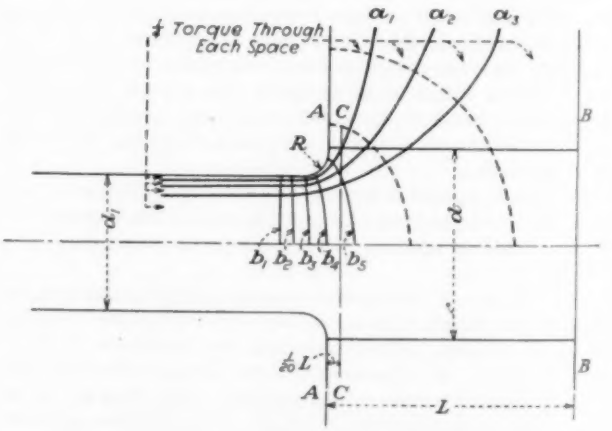


FIG. 7

The correct torque distribution must be determined by assuming no relative movement between shaft and hub in any part of the fit. This problem is practically the same as to find the stress distribution in a torque-transmitting shaft of varying diameter. A solution of that problem is given by F. A. Willers.[14]

Fig. 7 gives the approximate picture of the "torque flow" through a fit from a shaft to a hub. The lines a are equimomental lines; one-quarter of the torque is flowing between the surface and the line a_1, another quarter of the torque is flowing between the lines a_1 and a_2, and so on. The lines b are called equiangular lines; they are connecting points with the same an-

[13] Oil-Engine Designer, Ingersoll-Rand Company, Phillipsburg, N. J. Mem. A.S.M.E.

[14] *Zeitschrift für Mathematik und Physik*, vol. 55, 1907, pp. 225–263. Porter has used the same method for determining the stiffness of a shaft with varying diameter. Trans. A.S.M.E., paper APM-50-8.

gular distortion. It will be seen that for this special fit one-quarter of the torque is transmitted through one-twentieth of the surface (located between section *A-A* and *C-C*). If a joint like this is twisted to the point of sliding, then the friction coefficient calculated by Mr. Baugher's Equation [9] (in the following referred to as the "apparent friction coefficient") will be five times smaller than the true friction coefficient, existing between section *A-A* and *C-C*.

If the true friction coefficient always was five times the apparent, then the error would be of no importance. However, the relative values of the fillet radius R, the diameters d and d_1, the length L, and the outside diameter of the hub D have an overwhelming influence on the ratio between the apparent friction coefficient and the true one. The ratio varies from 1 to 10, or even more for fits of commonly used design.

From this it is obvious that for the calculation of a fit the knowledge of the torque distribution over the surface of the fit is as important as the knowledge of the friction coefficient. The large variation of the friction coefficients determined by the author might more or less be due to the disregarding of the torque distribution in the different fits tested by him.

The writer has referred only to torque-transmitting fits. Of course, similar conditions take place when a fit is subject to a force in axial direction.

CHARLES JABLOW.[15] The paper is a concise and useful treatise for any engineer in making or using press or shrink fits. Some of the suggested methods of calculation have been given the acid test over a period of years and this has resulted in entire satisfaction.

In reading the paper there are several points that suggest themselves. One thing is that it is not only important to know the torque that a fit will transmit, but of greater import is it to arrive at the true value of torque that the fit will be called upon to withstand. Many loose fits, of course, can be traced to the various factors enumerated. The writer wishes merely to point out that the normal duty of the fit will be only a fraction of the occasional duty, and therefore it is desirable to predetermine the maximum possible rates of acceleration or retardation and the inertia of the connected masses. The writer has in mind a pinion fit of a car or locomotive motor during an emergency brake application or perhaps a flashover in the motor. Under such conditions, the normal duty of the fit falls far short of requirements.

It is hardly conceivable to imagine a key working in parallel or as an aid to a fit. The writer's experience includes sheared keys that failed at about six times the maximum torque for which they were designed. Had a fit alone been used in this application, it is probable that the shrink fit would have been adequate. In spite of the admission that keyways are not always cut accurately in the plane of the axis of the shaft, many designers are prone to insist upon their use, and in the case of an electric motor, where a loss of a fit without a key might result in excessive overspeed, causing failure of the motor, these keys may be justified.

The lubrication of a press fit has been referred to, but few consider the lubrication in a shrink fit. It has been our experience with a ground shaft and ground bores that where extreme care was taken with the ring gage to check the shaft and a plug gage to check the pinion, the resulting lubricant remaining in the bore due to the prussian blue check resulted in just a sufficient film of lubricant to seriously affect the holding power. With the method of heating formerly used in the particular case referred to, this lubricating film was not removed. Subsequent heating with a gas torch in the bore, to the same temperature as previously used, resulted in very much higher tonnages. It is therefore important to notice that the character of the machine fit is not the all-important thing it seems.

It is the practice among certain manufacturers to specify tonnage instead of fit allowance. If, as the author indicates, the lubricant is squeezed out in the fit, this would seem to be a great fallacy, and it is more important to make sure that the fit allowance is had. It is no secret among men operating a press that they will obtain the tonnage required almost in spite of the fit allowance.

The author is to be commended for clarifying the points involved in making press and shrink fits.

[15] Railway Equipment Engineer, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

AUTHOR'S CLOSURE

The point raised by Mr. D. V. Waters regarding the use of keys where mating parts are made without gages is well taken. The author's experience, however, has been largely with miscellaneous gears, quite often fitted without keys and for which gages usually were not used, and there have been practically no cases of trouble resulting from insufficient fit allowances.

What Professor Waters says of the increased cost brought about by closer tolerances and special equipment necessary in employing press and shrink fits is probably true in many instances; yet it is the author's belief that, in the majority of cases where keys are used, a press fit or a shrink fit also is used, as Professor Waters brings out, to secure rigidity.

The second point raised by Professor Waters is very interesting, although the author cannot agree with the conclusions. Several errors seem to have been made in deducing the equations, so that a true comparison cannot be made with the equations as presented.

First, in the expression $d^3S_s/8$ in his first equation, S_s is the shearing stress in a steel key (see Fig. 8); whereas in the third

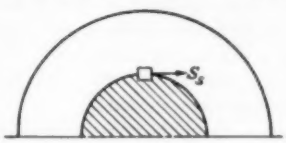


FIG. 8

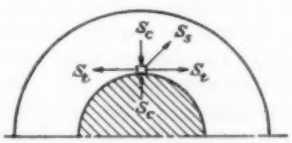


FIG. 9

equation S_s is the maximum shearing stress in a steel hub (see Fig. 9). A comparison of the two equations becomes, therefore, a comparison of the allowable shearing stress in key steel with the allowable shearing stress in the steel of which the hub is made. Obviously, the two allowable values are seldom the same; and as key stock is usually low-carbon steel, the advantage will most often be decidedly in favor of the material in the hub, especially if it is heat-treated. Therefore it cannot be said that for a steel shaft and hub the "space efficiency" of a press fit is only half that of a keyed joint, since for many cases it may be equal or greater, depending upon the materials used. Second, a safety factor of two is used in the third equation, but none is used in the first equation. This factor alone accounts for the difference in torque capacity represented by the two equations.

These discrepancies apply also to the comparison of the first and fifth equations. In addition, the assumption made in deducing the fourth equation—i.e., "cast iron with an assumed maximum tensile stress of $^1/_2$ S_s"—is open to question, since the failure of cast iron in torsion is actually a tensile failure. Most published data give the maximum tensile strength of cast iron as 0.83 to 0.90 of the maximum strength in shear. If the lower value were used, the fourth equation would become $S_s = 0.5\ S_s$, and the right-hand side of the fifth equation would become $d^3S_s/25$. If the safety factor were then omitted, as in

the first equation, the right-hand side of the fifth equation would become $d^2S_s/12.5$. A fair comparison may then be made between the first and fifth equations. It must be remembered, however, as before, that the values of S_s in the two equations are not the same.

The author is convinced, also, that when more comprehensive results are available on the behavior of fits in torsion, the safe value of f will be found to be several times the figure of 0.10, which is recommended in this paper and used in Professor Waters' third and fifth equations. If this proves to be true, the third equation will become at least equal to and perhaps greater than the first equation, even though the other points are neglected.

Therefore it seems to the author that Professor Waters' equations really show that the "space efficiency" of a press fit is greater than that of a keyed joint when the hub is steel and is nearly equal to that of a keyed joint when the hub is cast iron. There is considerable proof in practice that this is true for a steel hub.

The author was fortunate in being able to consult Dr. A. Nádai,[16] an authority on the plastic state of metals, with regard to the question raised by Mr. Wahl. The answer to this question is, as Mr. Wahl suggests, the plastic flow which occurs when the material in the hub is stressed beyond the yield point. If a straight bar is stressed above the yield point and the load is then held constant, the bar will continue to stretch a small amount. The same action takes place in a plastically overstressed hub of a press fit. Therefore what the author has called a decrease in the coefficient of friction is really a decrease in the pressure between the two parts. If, in such a case, Equation [10] be reversed in order to determine the friction coefficient f_1 from actual data observed in making a fit, the theoretical value of P obtained from the pressure chart will be greater than the pressure which actually existed between the fitted parts, and the error will show in a decreased value of f_1. The author has observed such a result a number of times when the fit allowance was so high that there was little doubt that the yield point of the metal in the hub had been exceeded.

The comments of Dr. Moss are a valuable contribution in clarifying the questions which the author was unable to explain, regarding the formulas given by the various authorities. In a number of texts on machine design and mechanics of materials, however, there does seem to be a lack of agreement concerning the form of the equations attributed to the different authorities.

Regarding the value of f which has a proper safety factor for use in Equation [9], the author believes that 0.05 represents the minimum value that need be used on the basis of present knowledge of press fits. The use of this value, of course, is equivalent to incorporating a factor of safety of 2 in the equation, the other values remaining as recommended on the fourth page. With what may be termed standard proportions of machine parts in use at present, the factor of safety above the torque to be transmitted usually will be considerably greater than 2 for a value of f equal to 0.10 and a value of e equal to 0.001 in. per in. of shaft diameter. As stated before, the author believes that when more data are available on the behavior of fits in torsion, the safe value of f will be found to be much greater than 0.10.

Every one who has compiled results of a large number of press fits has observed a great variation in the coefficient of friction. The author does not feel, however, that Mr. Mansa's explanation offers a complete solution, because he does not believe that the case of axial force in making or disassembling a fit—from which practically all the data given in this paper are taken—is similar to the case of twisting. The former is a case of constantly increasing area of contact—in pressing two parts together—accompanied by a proportionately increasing force, while the latter is a case of non-uniform distribution of torque.

Mr. Mansa's discussion of the correct torque distribution is of great importance. The author is familiar with Willers' method of determining the stress distribution in a shaft of varying cross-section. In fact, Timoshenko and Jacobsen applied this theory in 1924 to the case of press and shrink fits in the unpublished report to which the author referred on the fourth page. This report brings out several interesting facts. One is that slippage starts at one end of the fit—at A-A in Fig. 6, for example—and progresses with increase of torque until the entire fit slips. This result, of course, is to be expected. The torque required to produce the initial slip does not increase as the length of the fit is increased beyond a length of about half the shaft diameter. The torque required to produce complete slip does increase, however, directly in proportion to the length of the fit. The coefficients of friction at the instant of complete slip, as computed from an equation similar to [9], were found to be in the neighborhood of 0.30, as noted on the fourth page. According to Willers' theory, the torque required to produce initial slip will decrease, as Mr. Mansa suggests, as the fillet radius—R in Fig. 7—is decreased. Since in the majority of fits in practice the hub is pressed directly on a straight shaft so that there is no fillet, it is to be expected that initial slip of such fits takes place at very low values of torque. There seem to be no serious results of such conditions, however, and hundreds of cases can be cited where the fits are performing satisfactorily.

As the analysis given by Mr. Mansa assumes that there is no slip between hub and shaft, it would seem that a solution by this method would yield only the torque necessary to produce first slip, which in most cases is probably quite low in comparison with the ultimate torque capacity of the fit. Practical experience proves that it is not necessary to limit the torque to be transmitted to the initial slip value. Therefore, in view of these facts, it seems to the author that there is considerable justification for a simple statement of the ultimate torque capacity of a fit, as expressed by Equation [9], in which the value f_1 is admittedly an average coefficient of friction.

It is unnecessary for the author to emphasize the importance of Mr. Jablow's statement with regard to the correct determination of the maximum torque that a fit will be called upon to transmit. In many applications the maximum torque can be easily computed, but in others it cannot. Unless it is known quite exactly, however, similar safety factors must be used for the fits as are used in determining the design of the other parts of a machine.

[16] Research Engineer, Westinghouse Electric and Manufacturing Company. See paper "On the Mechanics of the Plastic State of Metals," presented at annual meeting of the A.S.M.E., December, 1929, and others.

MSP-53-11a

Haynes Stellite Cutting Tools

By W. A. BECKER[1] and E. E. GORDON,[2] KOKOMO, IND., and W. A. WISSLER,[3] LONG ISLAND CITY, N. Y.

This paper describes briefly the physical properties of Haynes Stellite and gives the more important details of several types of machining operations on which it is being successfully used. Its most important property is red hardness. Measurements of hardness were made by an improved method up to 1100 deg. cent. The resulting data show that the cobalt-chromium-tungsten alloy is harder than high-speed steel at temperatures above 500 deg. cent. This is believed to account for its superior cutting quality.

When used for milling, three typical jobs on cast iron show 87, 100, and 27 per cent increased production, respectively, on high-speed steel. On boring cylinder blocks the increase was 64 per cent. Turning operations in cast iron show 33, 35, and 60 per cent increase in production, and on steel, 123 and 93 per cent.

In general, Stellite bits and blades can be used economically on all operations on cast iron, semi-steel, malleable iron, bronze or similar metals, and on mild steel if the section is heavy and the tools are rigidly supported. It has not been successful on automatic or semi-automatic machines for cutting steel.



W. A. Becker



E. E. Gordon



W. A. Wissler

HAYNES STELLITE is a non-ferrous alloy consisting essentially of cobalt, chromium, and tungsten, invented and patented by Elwood Haynes, who is also credited with having built one of the first successful automobiles. The early experiments were directed toward the development of a non-tarnishing alloy for use in the manufacture of cutlery, but it was soon found that the addition of tungsten conferred a high degree of hardness; the resulting alloys were tested and found satisfactory as cutting tools in the Haynes automobile plant. The superiority of the alloy over cutting tools then in use was very striking, and its introduction into the machining industry progressed more or less rapidly until the war, at which time the alloy took its place as an important factor in the rapid production of war supplies. Its widespread use in subsequent years needs no description.

It is obvious that the properties of these alloys will vary considerably, depending upon their relative compositions. Some of them are soft enough to be forged and rolled to a very limited degree; these grades are used for knives and similar purposes. As more tungsten is added the hardness increases, reaching its maximum in grade No. 3, which is used largely for machine tools. Grade No. 1, furnished as welding rod, is almost equal to No. 3 in hardness, but has been especially adapted for welding on machine parts subject to abrasive wear. This paper will be confined to No. 3, and all data and figures given refer to this grade of metal, which must be cast to shape and finished by grinding.

Aside from the cutting quality of the tool, which at the present time must be determined by an actual cutting test, there are a few physical properties that are of sufficient interest to the subject at hand to be tabulated herein. Comparative strength figures, determined by a cantilever method duplicating rather closely the conditions under which a lathe tool operates, show that Stellite is admittedly weaker than high-speed steel; this is, however, compensated for in the design of tool holders and tools, and causes but little trouble in operation.

TABLE 1 PHYSICAL PROPERTIES

Hardness cold:	Stellite No. 3	High-speed steel, hardened and drawn
Rockwell	C-61	C-63
Brinell	600	627
Tensile strength, lb. per sq. in.	38,000	.....
Compressive strength, lb. per sq. in.	311,000	382,000
Cantilever strength (1/2 in. sq., 1 in. overhang), lb.	2500	5000

Haynes Stellite is usually thought of as being an extremely hard alloy, but its hardness when measured cold is not unusually great, as will be seen in Table 1. This fact makes it difficult to account for the superiority of the alloy as a cutting tool until it is realized that the cutting edges of tools in operation are heated to surprisingly high temperatures. It is very seldom that a visual examination of a lathe tool in operation gives any indication of a red heat, but Mr. E. G. Herbert, who has carried out extensive researches in England on the mechanism of cutting, has given definite determinations of cutting-tool temperatures.[4] By using a set-up in which the tool and the billet are made the

[1] Sales Engineer, Haynes Stellite Company. Mr. Becker spent five years in the employ of several of the large machine-tool and motor builders, actively engaged in research on the fundamentals of milling, boring, turning, etc., studying from both the practical and theoretical angles. Immediately following this work, in 1920 he became a sales engineer for the Haynes Stellite Company, connected first with the Detroit office and then in the Chicago territory.

[2] Sales Engineer, Haynes Stellite Company. Mr. Gordon has been a sales engineer for the Haynes Stellite Company since 1922, working in both the Cleveland and Detroit offices. Immediately prior to this he was engaged for six years in production work, time study and research on problems relating to the most efficient use of steel and alloy tools in one of the large automobile shops. The practical experience gained by this work was supplemented by technical training along the lines indicated.

[3] Metallurgist, Union Carbide and Carbon Research Laboratories, Inc. Mr. Wissler became metallurgist for the Haynes Stellite Company in 1917 and was closely associated with Elwood Haynes during the development period of the company. In 1923 he was transferred to the Union Carbide and Carbon Research Laboratories in charge of research and experimental work on alloy cutting tools. Prior to the connection with the Haynes Stellite Company he had secured degrees in chemistry and electrometallurgy and had held several positions more or less closely connected with the non-ferrous alloy industry.

Contributed by the Special Research Committee on Cutting of Metals and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[4] *American Machinist*, Feb. 18, 1926, p. 281.

elements of a thermocouple, and using very sensitive instruments for recording by photographic methods, Mr. Herbert determined that cutting-edge temperatures vary from 250 to 700 deg. cent. while cutting dry and from 100 to 525 deg. cent. when using a coolant, the temperatures being higher with greater cutting speeds. Obviously, if the tool does its work at such high temperatures, the effective hardness is the hardness of the tool within that range of temperatures. Mr. Herbert carried on experiments correlating the cutting quality and red hardness of different high-speed steels heat-treated by different

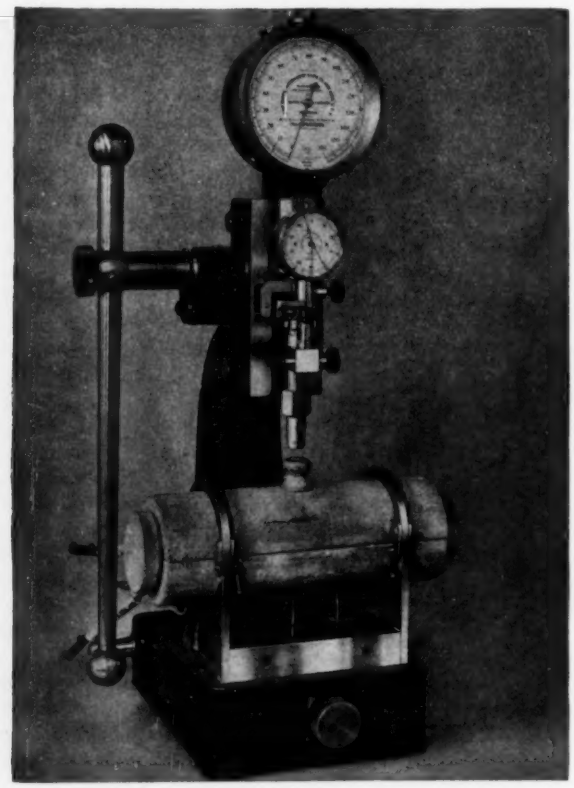


FIG. 1 MONOTRON HARDNESS-TESTING MACHINE

methods, and concluded that high-speed steel tools could be evaluated by a "figure of merit" consisting of a certain hardness at a certain temperature, the temperatures being somewhere in the neighborhood of 600 deg. cent.[5] For these experiments the pendulum hardness tester was used.

In view of the importance of the red hardness property in cutting tools, the Union Carbide and Carbon Research Laboratories has investigated hardness from ordinary temperatures to as high as 1100 deg. cent. After quite a number of experiments it was decided that the Monotron hardness-testing machine (Fig. 1) would be most readily adaptable for the purpose. This machine can be equipped with a furnace so designed that the hardness of a bar may be determined without removing it from the furnace, thus eliminating heat losses with the exception of those occurring by conduction through the penetrator, and by making the determination as quickly as possible these losses can be reduced to a minimum. Using this apparatus, data have been secured from which curves (Fig. 2) have been drawn showing the average hardness of Haynes Stellite and several makes of high-speed steel from room temperatures to 1100 deg. cent. At temperatures of 500 deg. cent. and higher, Haynes Stellite is harder than any steel tool or ferrous alloy as yet examined. This property is believed to be the dominating factor in the values of Haynes Stellite—supporting evidence being offered in that the successive development of plain carbon steel, Mushet steel, chromium-tungsten and chromium-tungsten-cobalt high-speed steel has resulted in every case in the production of a tool having greater hardness at higher temperatures.

The large majority of Haynes Stellite cutting tools are furnished in the form of bits and blades of solid Haynes Stellite, but for certain purposes it is advisable to use a welded tool consisting of a tough steel shank and a Haynes Stellite tip. It is recommended that solid tools be used in every case where it is possible and economical and that welded tools be used only on jobs where there is considerable overhang and when the size of the tool would result in extremely high costs if it were made solid. On practically all production jobs today tools of cross-section 1 in. sq. and under are made solid. Solid tools possess maximum cutting quality. They are cast in a special manner with chilled cutting surfaces. The excessive heat of the welding operation incident to the manufacture of welded tools destroys some of the outer chill and slightly impairs the cutting quality. Extensive tests in large production shops have proved conclusively that solid tools of Haynes Stellite are 10 to 15 per cent more efficient than welded tools.

TABLE 2 RECOMMENDED GRINDING PRACTICE

For Machine Grinding

	Ingersoll	Gisholt	Sellers	Lumsden
Carborundum...	401-A V-S	401-A V-S	36-N.L. 8 Alox.Vit.	24-R.L. 5 Vit
Hampden.....	3346-K.Sil.	3346-H.Sil.	3336-J.Sil.	3336-J.Sil.
Norton.....	3846-K.Sil.	3846-H.Sil.	3836-K.Vit.	3836-I.Sil.
Precision.....	4623-K.Sil.	4623-H.Sil.	36-K.Sil.	3618-J.Sil.

Hand Grinding

Carborundum.....	46-L-Alox.Vit.
Hampden.....	46-M-Vit.
Norton.....	46-M-Vit.
Precision.....	46-L-Vit.

Speeds: 3000 to 4000 ft. per min.

All cutting tools must be properly ground, and it is natural to expect that the proper practice for alloy tools would differ

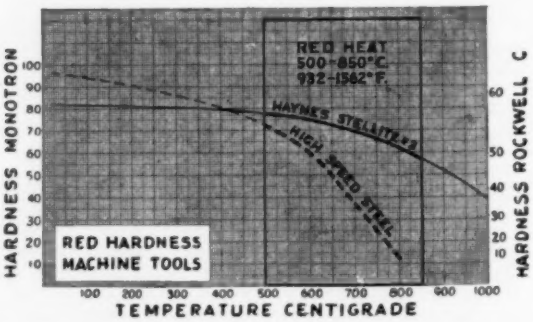


FIG. 2 AVERAGE HARDNESS OF HAYNES STELLITE AND SEVERAL MAKES OF HIGH-SPEED STEEL

from that for high-speed steel. Experience gained over a period of years in our own and customers' shops enables us to make the recommendations listed herewith. In addition to using the proper wheels, care should be exercised that the tools are not overheated and checked in the grinding operation. The generation of sudden heat should be avoided and the tool be heated gradually. If heat is suddenly applied, the alloy has a tendency to check and crack.

No one tool alloy has as yet been found to be the most economical on all types of operations. General rules are always of some value, although they always have exceptions, and it is

[5] Cutting Tools Research Committee, Institution of Mechanical Engineers, London, Dec. 2, 1927.

believed the following should be borne in mind in regard to Haynes Stellite tools. Machining may be divided into four general divisions. One of these consists of the machining of cast iron, semi-steel and malleable iron. The second division consists of the machining of mild steel when the operation is done under conditions having a considerable degree of rigidity, by which is meant that the steel casting or forging is of a rigid section, the number of tools used is small, and the machine is heavy and in good condition. The third type may be said to include the machining of light steel parts on automatic or semi-automatic machines and the machining of alloy-steel parts. The fourth class would consist of a miscellaneous heading, covering such materials as chilled iron, brass, bronze, aluminum alloys, hard rubber, fiber, mica, and similar materials. These materials are grouped not because of similarity, but because of their minor importance. It may be said that Haynes Stellite cutting tools are nearly always successful on the first and second classes, that they have met with but little success in the third class, and that they have come into wide use on some of the materials in the fourth class.

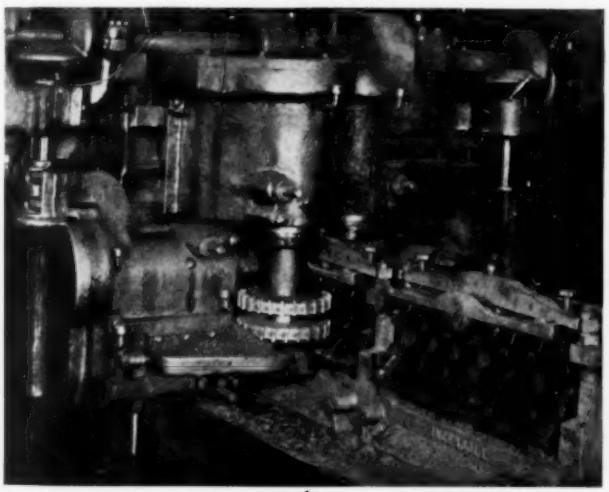


FIG. 3 MILLING CYLINDER BLOCK

Specific applications will be discussed under three general headings—milling, boring, and turning. Many of these jobs are being done exclusively with Haynes Stellite, and it was found necessary to go back a number of years and present the data obtained when the alloy was introduced on the job. In some instances these records are eight years old. In such cases there has been shown the comparison between Haynes Stellite and high-speed steel at the years indicated and there has been tabulated also the performance of the former at the present time. The percentage increase of production is shown based on the original figures, inasmuch as it is not possible to estimate the exact superiority under present conditions. It will be noted that a marked increase in the rate of production has taken place during the periods shown. This is due partly to the extensive experimental and development program carried out jointly by the Haynes Stellite Company and the Union Carbide and Carbon Research Laboratories, and to a lesser degree to improved machinery, better material to work upon, and to increased knowledge and skill as to how to secure the best results. Inasmuch as the authors cannot with accuracy estimate the increased production given by high-speed steel tools in the same period of time, it has been preferred to base the increase in production on the original figures and to show what Haynes Stellite is doing on these jobs at present.

The authors also have refrained from going into tool cost, and in some instances pieces per grind. Tool cost is figured in different ways in different shops, and a quotation of such figures might be misleading. If tool costs were to be determined with a high degree of accuracy, it would be necessary to consider the following items: first cost; scrap value, if any; number of times reground and total cost of regrinding; cost and loss of production due to changing tools. These items divided by the total number of pieces produced per tool will give an accurate figure on tool cost per unit of production. It is also difficult to give in all cases the number of pieces per grind. Some users will continue to operate tools that other men would change, and in other cases the material being machined varies so greatly that figures from day to day are not consistent. It is believed that the data given in this paper will enable production men and engineers to estimate reasonably closely the savings they may expect to make in their own shops.

MILLING OPERATIONS

Haynes Stellite used for milling has been most successful on cast iron, including all grades from grey to semi-steel, and production figures will be confined to examples on this material. In the last few years, however, owing to improvement in strength and cutting quality, a few milling operations on steel have been, and are being, successfully done with Stellite. Although at the present time the extent to which steel is milled with Stellite is very limited, there is reason to believe that applications of this kind will increase greatly in the near future.

Milling has been divided into three classes—heavy, medium, and light; the first two classes being done on jobs such as cylinder blocks, cylinder heads, and crankcases and the latter on chain-case covers, manifolds, bearing cups, and other small parts. A typical example of each of these classes of operations is given.

Table 3 contains data on the rough milling of cylinder blocks. It will be noted that the tools were introduced on a showing of an 87 per cent increase in production. Since that time a secondary increase of 27 per cent has been made, which would give an increase of 138 per cent for Haynes Stellite in 1930 over high-speed steel in 1922.

TABLE 3 HEAVY MILLING, CAST IRON

Rough Cylinder Block, Top and Bottom, Manifold Bearing Cap Slots and Valve-Cover Pads; Machine, Line Type Mill

	High-speed steel	Haynes Stellite	
	1922	1922	1930
Depth of cut, in.	1/8	1/8	1/8
Surface speed, ft. per min.	57	110	110
Feed, in. per min.	4	8	10.5
Pieces per hour	5.7	10.7	13.6
Increased production, per cent	..	87	...

An example of medium milling is given in Table 4, which shows roughing and finishing the top and bottom of a cylinder

TABLE 4 MEDIUM MILLING, CAST IRON

Rough and Finish Cylinder Block, Top and Bottom; Machine, Drum-Type Mill

	High-speed steel	Haynes Stellite
Depth of cut, in.	1/8	1/8
Surface speed, ft.	60	120
Feed per minute, in.	9	18
Pieces per hour	17	34
Increased production, per cent	..	100

TABLE 5 LIGHT MILLING, CAST IRON

Manifold; Machine, Vertical Mill

	High-speed steel	Haynes Stellite
Depth of cut, in.	1/8	1/8
Surface speed, ft. per min.	119	148
Feed, in. per min.	8 1/4	15 1/4
Machining time	1 min., 1 sec	33 sec.
Floor to floor time	2 min., 5 sec.	1 min., 37 sec.
Pieces per hour	29	37
Increased production, per cent	...	27

TABLE 6 MISCELLANEOUS MILLING OPERATIONS

Part	Operation	Machine	Cutting speed, ft. per min.		Feed, in. per min.		Pieces per grind		Floor to floor time, min. and sec.		Increased production per cent
			Haynes Stellite	High-speed steel	Haynes Stellite	High-speed steel	Haynes Stellite	High-speed steel	Haynes Stellite	High-speed steel	
Cylinder head.....	Rough and finish top and bottom	Double-spindle vertical	114	92	15	12	173	30	6'0"	7'30"	25
Cylinder block.....	Profiling base	Single-spindle vertical	88	63	5.6	3.7	80	30	9'0"	13'11"	46
Transmission case.	Mill top and side	Line type	126	68	8	5	200	80	1'25"	2'13"	56
Cylinder block.....	Rough and finish mill top and bottom	Double-spindle rotary	104	73	10	7	120	80	1'30"	2'8"	42.5
Cylinder block.....	Finish mill top, bottom and sides	Line type, eight-spindle	120	47	7	3.5	250	50	4'5"	7'55"	93
Crankcase, lower..	Rough and finish mill	Line type	98	54	16	9	240	100	1'55"	2'55"	52
Flywheel housing..	Mill joint surfaces	Double-spindle rotary	104	73	10	7	100	50	2'30"	3'34"	42.8
Transmission case.	Rough mill top and two sides	Single-spindle rotary	125	94	12	9	500	200	3'0"	4'0"	33
Cylinder head.....	Rough and finish mill bottom	Double-spindle rotary	126	59	17.5	12	268	200	1'25"	1'54"	34
Cylinder head.....	Rough mill top, bottom and side	Planer type	85	50	5	3	70	30	7'19"	11'30"	57
Transmission case.	Generate flange	Single-spindle rotary	100	50	12	6.0	50	30	6'0"	10'0"	67
Crankcase.....	Mill top and bottom	Two-spindle line type	128	70	9	5.0	100	20	4'40"	8'30"	82.5
Crankcase.....	Mill gear end	Plain horizontal	126	75	9	4.5	150	80	2'0"	3'34"	78.5
Cylinder block.....	Rough mill top, bottom and sides	Ten-spindle line type	125	57	8	4.0	100	50	3'33"	6'51"	93
Cylinder block....	Rough mill bottom	Two-spindle line type	75	66	10.5	6.8	100	50	3'0"	4'20"	44
Crankcase.....	Mill joint	Single-spindle line type	125	62	11	5.5	50	20	3'30"	6'0"	71
Cylinder block.....	Mill top bottom and side	Three-spindle line type	94	68	6	4.5	100	50	7'0"	9'15"	32
Electric flatiron....	Mill bottom	Single-spindle rotary	79	66	7.5	6.0	125	40	0'32"	0'40"	25
Solenoid frame....	Mill pad	Plain horizontal	79	41	5.8	2.5	125	30	2'6"	3'52"	84

	Cutting speed, ft. per min.		Feed, in. per min.		Pieces per grind		Increased production, per cent
	Haynes Stellite	High-speed steel	Haynes Stellite	High-speed steel	Haynes Stellite	High-speed steel	
Average	106	65	9.7	6.1	153	64	55.7
Increase percentage	63		59		139		

TABLE 7 MILLING OPERATIONS, PRESENT PRODUCTION WITH HAYNES STELLITE TOOLS

Part	Operation	Machine	Cutting speed, ft. per min.	Feed, in. per min.
Cylinder block...	Rough and finish mill top, bottom bearing cap slots, manifold and valve-cover pads	Line type	110	10.5
Cylinder block...	Rough and finish mill top, bottom, valve-cover and manifold pads	Line type	120	9
Crankcase.....	Rough and finish mill top and bottom	Drum type	120	18
Cylinder block...	Rough and finish mill bottom and valve-cover pad	Line type	110	8
Cylinder block...	Rough and finish mill top	Rotary	91	12
Cylinder block...	Rough and finish mill top and bottom	Drum type	92	17
Cylinder block...	Rough and finish mill top and bottom	Drum type	91	14.4
Cylinder block...	Rough and finish mill top and bottom	Drum type	104	13.8
Cylinder block...	Rough and finish mill ends	Drum type	102	12
Cylinder block...	Rough and finish mill ends	Drum type	110	15.5
Cylinder head....	Rough and finish mill top and bottom of cylinder head	Line type	90	11.5
Cylinder head....	Rough and finish mill cylinder contact faces	Drum type	124	9.3
Cylinder head....	Rough and finish mill top, manifold and cover pads	Line type	142	13.3
Cylinder head....	Rough and finish mill cylinder contact and surface	Rotary	86	13.5
Cylinder head....	Rough and finish mill top and cover side	Rotary	132	14
Crankcase.....	Rough mill top face	Line type	99	13
Crankcase.....	Rough and finish mill lower face	Rotary	111	15
Gearcase cover...	Rough and finish mill face (1 cut)	Line type	122	8
Crankcase.....	Rough and finish mill	Line type	109	9
Water manifold..	Rough and finish mill six pads	Double-spindle horizontal	172	24.3
Manifold.....	Rough and finish mill contact face	Double-spindle horizontal	126	13.8
Manifold.....	Rough and finish mill contact face	Single-spindle vertical	114	18
Manifold.....	Rough and finish mill contact face	Double-spindle horizontal	104	13.8
Bearing cap.....	Rough and finish mill sides	Double-spindle horizontal	232	14
Clutch housing...	Rough and finish mill face and recess	Drum type	166	31
Cylinder head...	Rough and finish mill pump end	Drum type	225	28
Gearcase cover...	Rough and finish mill contact face	Double-spindle horizontal	166	15.5

block. It will be noted that the speed was increased from 60 to 120 ft. per min. and the feed from 9 in. to 18 in. per min., resulting in a 100 per cent increase in production.

Table 5 represents comparative time-study data on a light milling operation on a cast-iron manifold. In this case the cutting speed was increased from 119 ft. to 148 ft. and the feed was increased from 8¼ to 15¼ in., representing a reduction of 46 per cent in machining time and 27 per cent increased production.

Table 6 is a tabulation of miscellaneous milling operations, covering practically every phase of milling on which Haynes Stellite tools are used. These figures represent operating conditions when Haynes Stellite was demonstrated and adopted as standard cutting tools on these jobs. Cutting speeds, feeds, and production vary according to the kind of material machined, size and shape of the casting, type of machine, cutter set-up and other operating conditions. A comparison with other tools indicates average increases of 63 per cent in cutting speed, 59 per cent in feed per minute, 139 per cent in pieces per grind and 55 per cent increased production.

Table 7 is introduced to show speeds and feeds at which similar milling jobs are being run today. It is impossible to summarize these data, but the individual figures will indicate what may be expected on jobs of these types.

The successful application of Haynes Stellite tools on milling operations, and in fact all machining operations, depends upon the proper ratio of speeds and feeds and methods of application. Hardness of material machined, kinds of fixtures used, condition and strength of machine, nature of castings, proper selection of cutters, and proper cutting and clearance angles all have an important bearing on the final results. As a rule 3 deg. on the face and 5 deg. on the diameter are used.

TABLE 8 BORING, CAST IRON

Cylinder Block; Machine, Six Spindle Vertical

	High-speed steel	Haynes Stellite	
		1922	1930
Depth of cut, in.	⅜	⅜	⅜
Surface speed, ft. per min.	45	66	75
Feed, in. per min.	2½	4½	7¾
Pieces per hour	14	23	34
Increased production, per cent	..	64	..

Boring Operations

The rough-boring of cylinder blocks is no doubt the largest single item of boring operations on which Haynes Stellite tools are used. There has therefore been selected a typical example of boring cast-iron cylinder blocks of a popular make of automobile. Table 8 lists comparative data showing that the cutting speed was increased from 45 ft. to 66 ft. per min. and the feed was increased from 2½ in. to 4½ in. per min., representing an increase in production of 64 per cent. More recent figures, as given, show an increase in production of 143 per cent over high-speed steel tools in 1922.

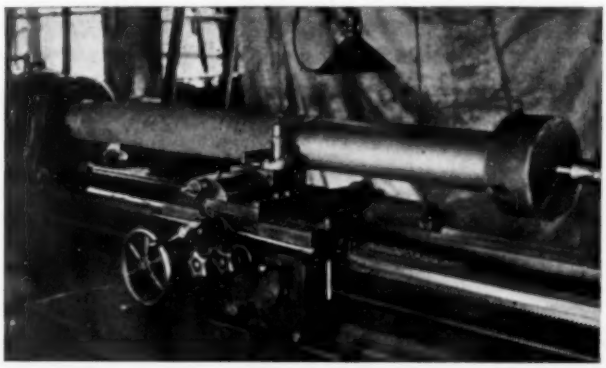


Fig. 4 Turning Ink Roll

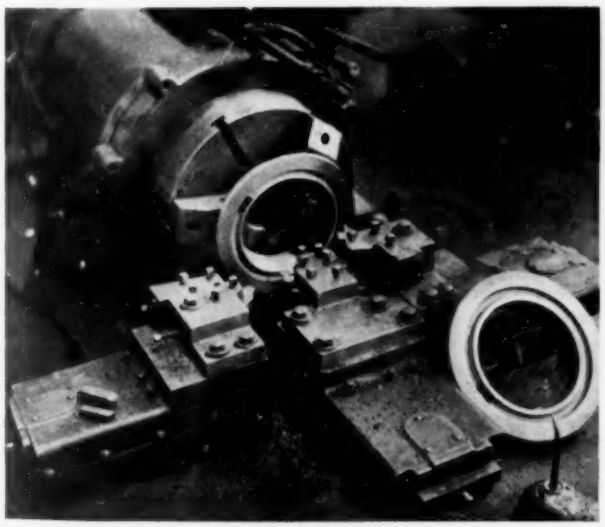


Fig. 5 Turning Bearing Retainer

Table 9 gives the operating data on eight cylinder-boring jobs as they are being run at the present time. As a general rule

TABLE 9 ROUGH BORING, CYLINDER BLOCKS
Haynes Stellite Tools

Blades in bar	R.p.m.	Cutting speed, ft. per min.	Feed, in. per min.
6	50	52	4½
6	100	104	7.3
6	90	98	7.6
6	63	63	8.5
4	48	41	2.9
4	65	55	3.3
4	100	74	6¾
4	85	75	7¾

cutting angles of 6 deg. without any back slope are recommended for this kind of work. Cutting speeds range from 41 ft. to 104 ft. per min. and feeds from 2.9 to 8.5 in. per min.

Turning Operations

Turning operations with Haynes Stellite tools cover a larger range of work than either milling or boring. Milling and boring are generally done on machines designed especially for that purpose. Turning operations are done on various types of screw machines, lathes, and semi-automatics. As is true on milling operations, speeds and feeds for turning with Haynes Stellite tools largely depend upon the hardness of the metal machined, condition of the machine, shape of the casting, class of work, number of tools in the set-up, nature of the cut, and other operating conditions.

In regard to the materials being machined, it is of course well known that Stellite is successful on cast iron under almost any conditions, assuming only that the machines used are in a reasonably good condition. It should, however, not be forgotten that the first use of Haynes Stellite cutting tools on a



Fig. 6 Flywheel Milling

large scale was in turning steel shells during the war. Many steel jobs can be successfully and economically machined with these tools, especially if the work consists of roughing cuts with round-nosed tools when the work is rigidly supported and it is not necessary to machine sharp corners. The specific applications shown will include cast iron, steel, and bronze.

Table 10 gives comparative results on turning a cast-iron ink roll. On this job the speed was increased from 55 ft. to 87

TABLE 10 TURNING, CAST IRON
Ink Roll, 6⅜ In. Diameter by 71 In.; Machine, Engine Lathe

	High-speed steel	Haynes Stellite
Depth of cut, in.	⅛	⅛
Surface speed, ft. per min.	55⅓	87
R.p.m.	35	52
Feed per revolution, in.	0.050	0.050
Pieces per grind	1	2
Machining time, min.	41	27
Floor to floor time, min.	56	42
Increased production, per cent	..	33

ft., with a resultant saving in machining time of approximately 34 per cent. The cutting angles used on this tool were 5 deg. top side slope, 6 deg. side clearance, and 6 deg. front clearance.

Table 11 is a rough turning, boring, and facing operation on cast-iron bearing retainers on an automatic lathe. With Haynes Stellite tools it was possible to increase the speed of the job from 80 ft. to 115 ft., using the same feed as previously used with other tools. This increase in cutting speed resulted in an increase in production of approximately 35 per cent. The cutting angles used on this job were 10 deg. side clearance and 6 deg. front clearance, with no top side slope.

TABLE 11 TURNING, CAST IRON

Bearing Retainer, 11 1/8 In. Diameter by 2 In.; Machine, Semi-Automatic Lathe

	High-speed steel	Haynes Stellite
Depth of cut, in.	3/32	3/32
Surface speed, ft. per min.	80	115
R.p.m.	28	40
Feed per revolution, in.	0.023	0.023
Pieces per grind	150	250
Machining time	2 min., 44 sec.	1 min., 55 sec.
Floor to floor time	3 min., 9 sec.	2 min., 20 sec.
Pieces per hour	19	26
Increased production, per cent	..	35

Table 12 is a turning operation on a flywheel on a vertical turret lathe. The cutting speed of the roughing operation was increased approximately 50 per cent and of the finishing operation approximately 40 per cent. The feed per revolution of the roughing operation was increased 50 per cent and of the finishing operation 100 per cent. The increase in speed and feed on this job resulted in an increase in production of 60 per cent. The cutting angles on these tools were 5 deg. top side slope, 6 deg. side clearance, and 10 deg. front clearance.

TABLE 12 ROUGH AND FINISH TURNING, CAST IRON

Flywheel, 15.075 In. Diameter by 4 3/4 In. Wide; Machine, Vertical Lathe

		High-speed steel	Haynes Stellite
Depth of cut, in.	Rough	3/16	3/16
	Finish	1/32	1/32
Surface speed, ft. per min.	Rough	47	71
	Finish	79	110
R.p.m.	Rough	12	18
	Finish	20	28
Feed per revolution, in.	Rough	0.032	0.050
	Finish	0.015	0.032
Pieces per grind		20	48
Machining time		12 min., 20 sec.	7 min., 20 sec.
Floor to floor time		13 min., 20 sec.	8 min., 20 sec.
Pieces per hour		4.5	7.2
Increased production, per cent		...	60

Table 13 gives the comparative results of a turning operation on S.A.E. 3115 steel gear blanks on a vertical turret lathe.

TABLE 13 TURNING, S.A.E. 3115 ROLLED STEEL

Gear Blank, 25 1/2 In. Diameter by 3 1/4 In.; Machine, Vertical Lathe

	High-speed steel	Haynes Stellite
Depth of cut, in.	3/16	3/16
Surface speed, ft. per min.	47 1/2	95
R.p.m.	7	14
Feed per revolution, in.	0.027	0.033
Pieces per grind	20	40
Machining time	18 min., 30 sec.	7 min., 30 sec.
Floor to floor time	20 min.	9 min.
Pieces per hour	3	6.7
Increased production, per cent	..	123

This job is representative of the type of operation referred to in the second of the four general subdivisions of Haynes Stellite machining operations. It consisted of machining a steel part rigidly supported, of heavy section, where the number of tools was small and the machine was heavy and in good condition. The cutting speed in this case was increased 100 per cent and the feed was increased approximately 22 per cent, resulting in an increase in production of 120 per cent. The cutting angles used on this job were 22 deg. front clearance and 20 deg. side clearance. The tool was hollow-ground on top parallel to the cutting side with a 1/32-in. land.

Table 14 gives the comparative results on turning and chamfering an S.A.E. 1020 rear-axle shaft. On this job the cutting speed was increased from 66 ft. to 110 ft. and the feed was increased from 0.015 in. to 0.021 in. per revolution. This increased speed and feed resulted in an increased production of 93 per cent. The cutting angles used on the turning tool were 10 deg. top-side slope and 7 deg. side and end clearance.

TABLE 14 TURNING, S.A.E. 1020 STEEL

Straight-End and Chamber Rear-Axle Shaft 1 15/64 In. Diameter by 6 7/16 In. Long; Machine, Special Lathe

	High-speed steel	Haynes Stellite
Depth of cut, in.	1/8	1/8
Surface speed, ft. per min.	66	110
R.p.m.	16	280
Feed per revolution, in.	0.015	0.021
Pieces per grind	15	35
Machining time	2 min., 36 sec.	1 min., 6 sec.
Floor to floor time	3 min., 6 sec.	1 min., 36 sec.
Pieces per hour	19 1/2	37 1/2
Increased production, per cent	..	93

Table 15 shows the comparative results on a rough turning operation on a bronze bushing. Cutting speed was increased from 78 ft. to 131 ft., and in this case the same feed was employed. On this job production was increased 50 per cent by the use of Haynes Stellite tools. The cutting angles used on this tool were 10 deg. top-side slope, 10 deg. front and side clearance, and 5 deg. drop grind or negative back slope.

TABLE 15 TURNING, BRONZE

Bushing, 2 1/2 In. Diameter by 3 In.; Machine, Turret Lathe

	High-speed steel	Haynes Stellite
Depth of cut, in.	1/16	1/16
Surface speed, ft. per min.	78	131
R.p.m.	112	187
Feed per revolution, in.	0.021	0.021
Pieces per grind	9	40
Machining time	1 min., 15 sec.	45 sec.
Floor to floor time	1 min., 30 sec.	1 min.
Pieces per hour	40	60
Increased production, per cent	..	50

Summary

It would be difficult and perhaps of little value to attempt to make a mathematical summary of the data presented in this paper. Figures of this type but covering a much larger number of jobs indicate that the use of Haynes Stellite on operations now being done with steel tools will result in increased production and increased life per grind within minimum and maximum figures as follows:

	Increase in production, per cent	Increase in number of pieces per grind, times
Cast Iron:		
Milling	25 to 100	2 to 5
Boring	25 to 110	2 to 6
Turning	20 to 140	2 to 5
Mild Steel:		
Milling	25 to 35	2 to 3
Boring	30 to 120	1 1/2 to 5
Turning	15 to 125	1 1/2 to 6

Discussion

A. K. Dean.[6] The papers on cutting tools by d'Arcambal and by Becker, Gordon, and Wissler are each so comprehensive and authoritative that the writer does not propose to attempt to refute any statement, but can only heartily agree with each.

This paper will be confined to a discussion of some practical methods of procedure in the selection and adaption of the proper cutting tool for the job, and will try to emphasize one or two points in the light of experience.

All will agree that there are four distinct steps in cutting material: (1) Carbon steel, (2) high-speed steel, (3) Stellite, (4) cemented tungsten carbide. While each class may be termed an advancing step as to hardness over the preceding class, yet no one of the groups has replaced any of the others. There is still a place for each.

[6] American Laundry Machinery Company, Rochester, N. Y.

The initial cost and the ultimate life must be considered in this selection. There is no justification for using a material which costs $450 per lb. if a carbon steel will do the work nearly as fast

The writer's first introduction to high-speed steel was in a railroad shop about the year 1904, and the excitement caused recently by the advent of cemented tungsten carbide was mild as compared to the rush then to change from carbon to high-speed steel; and the results obtained on tire turning, cylinder boring, driving-box bronze turning and boring, etc., led to a sweeping change for all classes of work.

In our brass shop where we had 15 brass finishers making injector nozzles, blowoff cocks, drain cocks, gage cocks, lubricator parts, and other red and yellow brass parts, we soon discovered that carbon steel was far superior to high-speed steel and has not yet been replaced.

About a year ago the writer asked a man whose duty it was to select machine tools and cutting steels for a large manufacturing concern, with shops located in several cities, what he was doing with cemented tungsten carbide. His reply was that they had spent about $8000 for Carboloy, Widia, and other makes, but that he had found as he visited the different shops that generally it was on the shelf and the better understood high-speed steels were in use.

That is a good demonstration of how not to begin the change from one cutting material to another, which differs so radically in character. Trying to gorge a shop with a large quantity generally spells failure so decisively that it is hard to get a good start again.

In 1914 the writer has his first introduction to Haynes Stellite, and noting its remarkable performance turning steel high-explosive shell bodies in munition plants, he tried to adapt it to general work, principally turning cast-iron gear blanks, pulleys, and general lathe work, most of which had intermittent cuts. The results were so poor that not until 10 years later, in 1924, was he induced to again try Stellite. This time the results were highly satisfactory, no doubt largely due to the improvement in Stellite, but also in large measure due to a much more intelligent approach to the job.

This time our procedure was as follows: The selection of a department manned by a foreman who was interested in getting the highest possible production, as a laboratory or proving ground; and then by taking one job at a time and carefully selecting the proper size and shape of tool coupled with correct speeds and feeds, we learned how to adapt it to other jobs, so that now this material has earned a permanent place in our shop, but has by no means replaced high-speed steel, which is still supreme on many jobs.

We are using this material in two forms: (1) Square cast bars, principally for turning 65 per cent carbon spindle steel, a step-up from 80 f.p.m. to 140 f.p.m. (2) Machine steel shanks with Stellite tips. These tipped tools are splendid in design for general cast-iron turning, and in cost figure lower than high-speed steel when labor of forging and heat treating is considered.

The writer first saw a demonstration of cemented tungsten carbide about three years ago, and it was a really remarkable performance. Among other things, he saw it turning glass, porcelain, hard rubber, manganese bronze, etc. He also saw the operator chuck a sash weight (and any one who is familiar with the foundry practice in vogue in making sash weights knows that they are not meant to be machined and that there is no cutting material that will successfully machine one except cemented tungsten carbide), so that it ran eccentric $^3/_8$ in., and take a cut $^1/_4$ in. deep, so that at each revolution the cutting tool had to cut into the scale and sand inclusion at a cutting speed of 350 f.p.m., and after taking a cut 14 in. long, the tool showed no abrasion or dulling. This was the most impressive test because best understood.

We have now spent two years fitting cemented tungsten carbide to some 12 or 15 jobs. During that time we have abandoned 12 or 15 other jobs as not suited for cemented tungsten carbide and have gone back to high-speed steel.

All of this cutting and trying has been done in this same departmental laboratory or proving ground, and we now feel as though we have learned the fundamentals of this latest addition to cutting tools, and are ready to make many further applications as soon as normal production will permit.

While our experience as to cutting tool design, machine tools, and care of tools in operation and grinding checks generally with that of Mr. St. Clair, we wish to emphasize one or two points.

1 Design of a cemented tungsten-carbide tool.
   (*a*) Tools of ample size should be used. Owing to the relatively high initial cost compared to high-speed steel, there is a temptation to attempt too small a tool for the job. This is false economy and generally spells defeat.
   (*b*) Do not be afraid to do a little experimenting on your own account. While sales engineers for cemented tungsten carbide generally recommended a tool that is flat on top and with a large radius at cutting edge, we have obtained better results in some cases with top and back slope and a radius of as little as $^1/_{32}$ in. This top slope gives better shearing action and cuts down the power required, and is thus easier on the machine tool. Any cutting tool should be set slightly above the center in a turning operation, but this is especially important with cemented tungsten carbide. If tool is below center, the chip pressure tends to pull the cutting edge rather than push it, and fails to take advantage of its compressive strength. Care should be taken, however, so that work will not rub the tool, with 4 to 6 deg. clearance.
2 The machine tool.
   (*a*) It is not necessary to await the purchase of new rigid superspeed tools to cash in on cemented tungsten carbide. Much can be done with our present tools. We have made successful applications of cemented tungsten carbide to jobs on lathes that have been in constant heavy duty use for eight years, and we are showing a 30 to 50 per cent saving over any previous performance. However, any one purchasing new tools should do so with a view to getting cutting speeds, rigidity, and power that will stand up under heavy cuts with cutting speeds of 250 to 300 f.p.m. on cast-iron turning and not less than 125 f.p.m. on planing.

Early in November the writer saw a planer on the test floor of a prominent manufacturer of planers, which was the largest tool ever turned out by that manufacturer, with a table 10 ft. wide and 30 ft. long, arranged for cutting speeds of 20 to 120 f.p.m., and in the same shop he saw a planer at work on regular production work, cutting cast iron at 120 f.p.m.

The manufacturers of cemented tungsten carbide have said that they were not ready for planer application, but from what we can see around the corner, they are about ready now, and any planer with a cutting speed of less than 125 f.p.m. may soon be in the obsolete class.

With regard to the power requirements on machine tools, when using cemented tungsten carbide, that is, the high-speed in-put required, this does not seem to be in direct ratio to the results, as the cemented tungsten carbide will hold a much keener cutting edge, and thus the removal of the metal is a clean cutting oper-

ation requiring less power than if a tool with a more blunt edge were used.

Owing to its ability to resist sand inclusion and hard spots in cast iron, cemented tungsten carbide is an ideal material for the new semi-automatic machine tools with complicated set-ups where tool grinding is an expensive operation, due to disturbing an expensive set-up. In cases of this kind, the high initial cost is justified even if there is not much gain in speeds and feeds.

As to the care of tool in operation, our practice when starting a new job on cemented tungsten carbide is to double the feed and speed as standard for Stellite, and then make gradual changes from that point until we get best results. Experience shows this to be about the correct speed and feed for new rigid tools, but on old machine tools, higher speeds with lighter feeds seem to work better.

As to grinding, we have found hand grinding to give best results owing to the more sensitive control. Any attempt to force the grinding operation beyond the cutting capacity of the wheel only results in rapid wheel wear.

On the 12 per cent cobalt material we are using a 60-grain I-grade wheel for roughing, and finishing on a 120-grain I-grade wheel. On the 50 per cent cobalt material, we are using much harder wheels.

Honing the cutting edge between grinds has increased time between grinds by as much as 300 per cent.

We believe 5000 f.p.m. surface speed to be the maximum, and better results are generally obtained below this speed.

All our applications of cemented tungsten carbide so far have been turning operations, but we are planning on some boring jobs, not so much to increase quantity as to improve accuracy. Boring with our present method generates so much heat in the part that it is hard to handle in the unloading operation, and thus varies in size when cool.

If we duplicate our turning experience on boring operations, we will find the heat in the chip rather than in the work. On our heaviest cuts, such as $^3/_{16}$ in. depth of cut and $^1/_{10}$ in. feed at 200 f.p.m., on cast iron, the work was not over 110 deg. fahr., while the chips were red.

In closing the writer wishes to say that every job is a law unto itself, but these laws are constantly changing with new developments in cutting steels. One of our largest production jobs is planing gray-iron with about 20 per cent steel. For this job nothing works as well as 18 per cent tungsten high-speed steel. Another job is turning steel pipe, and here again high-speed steel is still on top, although both Stellite and cemented tungsten carbide have been tried.

Authors' Closure

By W. A. Wissler. Mr. Dean's discussion outlined a very rational method of undertaking the introduction of a new cutting material into one's own shop; such a plan is particularly applicable and desirable if the tool in question is a new and as yet unproved product. Usually it is most satisfactory and economical to proceed slowly until the new material has definitely proved its value. The final answer may, however, be obtained more quickly if the experience and knowledge of the maker's service department are fully utilized.

As Mr. Dean states, no new tool metal produced since the beginning of the steel age has displaced entirely any tool already in use. The introduction of a new tool either has permitted old operations to be done more economically or has made possible the working of materials theretofore considered unmachinable. There is a decided possibility that tungsten-carbide tools will permit the use of steels much harder than at present.

The difference in results obtained from Haynes Stellite tools in 1914 and 1924 can be explained partly by the improved machine tools available to the latter date, but largely by the fact that intensive and continued research will improve any product. During the period mentioned, the major improvements in Haynes Stellite tools consisted of an increase in strength and a great increase in the ability to cut cast iron and similar materials. Both of these improvements were the direct result of research and development directed toward those ends. If Haynes Stellite produced in 1931 were compared with that made in 1924, additional increases in quality and uniformity would undoubtedly be noticed.

MSP-53-11b

Tool-Steel Tools

By A. H. d'ARCAMBAL,[1] HARTFORD, CONN.

Metal-cutting tools offered the trade today are vastly superior in quality to tools of a decade ago. The factors that govern the life of metal-cutting tools made from carbon, alloy, and high-speed tool steels, namely, design, steel, and treatment, are discussed in this paper, particular stress being laid on the importance of correct design.

The important role played by cutting fluids or coolants as affecting the life of the tool, as well as the necessity of employing the proper grinding wheels and correctly using them, are also covered in some detail.



THE quality of metal-cutting tools such as drills, reamers, taps, dies, cutters, hobs, etc., in use today is vastly superior to that found in tools manufactured a few years ago. Marked improvement in machine-tool design, tougher materials to machine, together with the demand for greater accuracy and economy, all have been contributing factors in the development of more efficient small tools.

This paper covers only cutting tools made of carbon, alloy, or high-speed tool steels. Tungsten-carbide tools and Stellite tools are being covered in other papers.

The following three factors govern the quality of metal-cutting tools: (1) Design, (2) quality of steel used, and (3) hardening treatment. It will be noted that two of these three factors, namely, raw material and hardening treatment, are metallurgical problems. For this reason plants manufacturing metal-cutting tools should maintain a well-equipped metallurgical department. This department not only should prepare specifications for the different types of steels employed, inspecting all material when received, but also have full control over all hardening operations. Improved steels and treatments will be developed from time to time by a properly conducted metallurgical department, keeping pace with the increasing demand for better cutting tools.

Design

To cover properly this factor of design it is necessary to consider the various types of tool-steel tools most commonly used.

Drills

Twist drills have been standardized so well in regard to angle-of-lip clearance, point angle, etc. (see catalogs of drill manufacturers), that time will not be taken to discuss tools of this description in detail.

The high-speed steel twist drill of today is in every sense of the word a production tool. In special demonstrations at machine-tool exhibits twist drills have performed in a truly remarkable manner. One-inch high-speed steel drills have penetrated cast iron at the rate of over 100 in. per min. and steel at over 50 in. per min. with practically no effect on the drill point after drilling a number of holes under such grueling conditions.

Reamers

Reamers are used to produce smooth round holes that are as close to the desired size as it is possible to obtain. Many types of reamers are in use today, the following being the types most commonly employed:

Hand reamers (solid and adjustable)
Shell reamers (solid and adjustable)
Chucking reamers (solid and adjustable)
Expansion chucking reamers
Expansion hand reamers
Jobbers reamers
Locomotive taper reamers
Bridge reamers
Taper-pin reamers
Taper reamers for standard tapers.

Number and shape of lands, relief, back taper, finish, all are factors that must be carefully controlled. It is the usual practice to design reamers of various types with staggered flutes, or what is more commonly called uneven spacing of flutes, resulting in a freer cutting tool with chattering reduced to a minimum. Smooth true holes usually are obtained from reamers so designed.

The most efficient type of adjustable-blade shell reamer is of the serrated-blade design. The blades in this type of reamer are held securely over their entire length. Adjustment to the maximum diameter can be made without reducing the strength of the reamer. There is an increasing demand for expansion chucking reamers. Reamers of this type should be provided with a lock nut, the same to be located inside the reamer so as to permit its use on bottoming holes.

The helical or fast-spiral taper-pin reamer is rapidly supplanting the straight and slow spiral-fluted types. These fast-spiral reamers are very free cutting, producing smooth round holes. One or two designs on the market can be used successfully when either driven by machine or used by hand. Other designs possess such an extreme spiral that they can be used successfully only under power.

The duplex-spiral taper reamer (see Fig. 2), a fairly new design of tool, is meeting with favor by the trade. In this interesting design, one or two flutes cross the path of the other flutes resulting in the production of extremely smooth and round tapered holes. One manufacturer offers a spiral-fluted taper reamer with one straight flute; this design of reamer also produces very satisfactory holes.

The four-grooved high-speed steel twisted-type locomotive taper reamer has lost its popularity during the past two or three years, the majority of railroad shops now using the milled-type high-speed steel locomotive taper reamer with six to eight spiral flutes, depending on the diameter of the tools. While the twisted type of reamer is somewhat less expensive than the milled type, the latter reamer not only gives considerably more service, but produces a more accurate hole.

As in the case of taps, the size cut by a reamer is dependent to a considerable extent on the type of material being machined.

---

[1] Consulting Metallurgist, Pratt & Whitney Company. Graduated from University of Michigan in 1912 with degree of Bachelor of Chemical Engineering and entered service at Detroit Copper and Brass Rolling Mills, Chalmers Motor Co., and Dodge Brothers as Chief Chemist. Was employed at Wright-Martin Aircraft Co. for a year as Chief Metallurgist. Joined Pratt & Whitney Co. in 1919.

Contributed by the Special Research Committee on Cutting Metals and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

As a proof of this statement holes were recently reamed in several different kinds of materials, using a 2-in. serrated-blade reamer with a peripheral speed of 180 ft. per min., and a feed of 0.011 in. per revolution. The following results were obtained:

Machinery steel and heat treated S.A.E. No. 6150 steel	The 2-in. standard cylindrical plug gage just entered hole
Annealed high-speed steel	Gage fit sloppy. (Holes from 0.001 to 0.002 in. larger than gage)
Cast iron and bronze	Plug gage would not enter hole

Taps

The different types of taps most widely used include machine-screw taps, hand taps, nut taps, tapper taps, pulley taps, boiler taps, pipe taps, coupling taps, and staybolt taps.

The following variables affect tap life and so must be kept under careful control: (1) Number of flutes, (2) shape and depth of flutes, (3) width of lands, (4) chamfer, (5) relief, and (6) back taper.

Taps designed for work in steel will not always give satisfactory service when used on other metals such as brass or aluminum. Care also should be taken to see that taps designed for through-hole jobs, such as spiral-pointed taps that force the chips ahead of the tap, are not used on blind holes. Difficult blind-hole jobs that cannot be threaded satisfactorily with the regular design of straight-fluted tap often can be successfully tapped with a spiral-fluted tap, the direction of spiral being the same as the cut. Taps of this design bring the chips out of the hole, thus preventing chipping of the threads and breakage.

Demand for greater accuracy has resulted in the use of ground-thread taps by the majority of the automotive plants, as well as by other large manufacturers. These taps are ground on the outside diameter, root diameter, and in the angle of the thread. The shanks also are ground concentric with the threads. These taps produce threads that, under satisfactory operating conditions, will meet the N.S.T.C. Class 3 tolerance. Many Class 2 jobs can be produced satisfactorily with cut-thread taps. Unless great accuracy is desired, the high-speed steel cut-thread tap should be specified, due to these cut-thread taps in the majority of cases threading at least as many holes per grind as the more expensive ground-thread tap.

It must be borne in mind that the same tap will not produce the same size thread on all different types of material. The type and condition of the machine and tap holder used also will affect the size of the thread. The use of sulphur-base oils will result in a different sized tapped hole than when soluble oil is used, this being especially marked in the case of bent-shank tapper taps used in full-automatic tappers. With these facts in mind, it is not difficult to understand the reason for the tap manufacturers' refusal to guarantee the size of the tapped hole.

It is not uncommon today to find high-speed steel taps being run at drill speeds in many automotive plants. Taps which will operate successfully under such severe conditions must be correctly designed, made from a good quality steel, and be properly hardened.



Fig. 1 Representative Group of Tool-Steel Tools

Dies

The most commonly used types of dies are: Inserted-chaser

dies, adjustable round split dies, spring screw-threading dies, solid square-bolt dies, solid square-pipe dies, and hexagon re-threading dies.

The inserted-chaser die is the type most generally employed today for threading bolts, screw-machine parts, etc. The adjustable round split die or button die finds wide use in the threading of smaller sizes of threads. Dies are so well standardized as regards design that it is not necessary to go into details at this time.

Cutters

Rapid strides have been made during the past few years in cutter design, due principally to the machine-tool builders bringing out heavier and more powerful milling machines. The modern milling machine provides quicker change of speeds and feeds, improved mechanical control, etc. Increased demand for parts made of heat-treated alloy steels also has been a contributing factor in the vast strides made in the improvement in cutter quality. A few of the more popular types of cutters are: Plain milling cutters, side milling cutters, half-side milling cutters, staggered-tooth milling cutters, inserted-blade cutters, end mills (shell and shank type), formed milling cutters, thread milling cutters, metal slitting saws, screw slotting cutters, involute gear cutters, and helical cutters.

Number and shape of teeth, helix angle, rake, clearance, hole diameter, etc., are all important factors that govern cutter design. Plain milling cutters for heavy production work are provided today with a high helix angle, rake of 12 to 15 deg. sometimes greater, with just enough clearance to prevent rubbing back of the cutting edge. Helical mills of this design will take cuts at least $^1/_2$ in. in depth in ordinary steel, the abundant chip room found in this design taking care of chip disposal. The high helix angle results in the production of smooth surfaces due to the shearing action. Fairly recent developments in cutter design have brought out many efficient types of cutters.

Half-Side Mills

These cutters have spiral teeth, the face of the teeth being undercut, resulting in freer cutting than is obtained with the older type side mills. These half-side milling cutters give much greater service than can be obtained from the usual design of side mills due not only to a longer life between grinds because of the shearing action, but also to the greater number of grinds possible because of the deep recess on the cutting side. Half-side mills are ideal for straddle-milling jobs and similar operations.

Staggered-Tooth Milling Cutters

Cutters of this design have teeth cut at alternate right- and left-hand spiral angles. This construction, together with the considerable angle of undercut, produces cutters which can remove large amounts of metal without excessive chatter and vibration, producing a smooth, clean finish. These cutters are very efficient for milling deep cuts or slots in metal. The side teeth act as chip removers.

End Mills

Shell-and-shank-type end mills offered the trade today are of the deep-recessed type. Mills of this design have more chip clearance than the older design, as well as possessing considerably greater life due to the large number of re-grinds possible.

Inserted-Blade Cutters

A large number of railroad shops in this country have installed during the past few years modern milling machines, resulting in the demand for inserted-blade cutters of a more rugged and efficient design. Cutters with correctly hardened high-speed steel blades securely locked in heat-treated alloy-steel bodies, provided with large bores, are giving remarkable service in slab milling, channeling, shoe and wedge jobs, and similar operations. In properly designed slab-milling cutters the sections are so designed as to give either a continuous or herring-bone spiral, permitting cutters to be resharpened as a gang instead of individually, thus insuring a continuous smooth surface on the work. The majority of inserted-blade cutters offered the trade are so designed as to permit easy replacement by the customer of worn-out blades with new blades.

Metal-Slitting Saws

Just a few years ago there was offered the trade only one simple design of metal-slitting saw to be used on all types of material. Today there are several designs, the type to be used depending on the nature of the material being machined. Saws designed for slitting steel will not give the best of service when used on non-ferrous metals such as copper, aluminum alloys, non-metallic materials, and the like. Saws used on non-ferrous and non-metallic materials have fewer teeth, greater rake to cutting edges, and higher polished surfaces. A still different design works more efficiently on cast-iron jobs. Number and shape of teeth, amount of concavity, rake, and finish, all are factors that must be carefully controlled.



Fig. 2 Duplex Spiral-Taper Reamer

Form Cutters

Form milling has today been placed on a production basis due to the introduction of spiral-fluted form cutters eccentrically relieved and provided with rake-tooth form. The shearing action due to the spiral-flute form results not only in a longer life for the cutter, but also produces a smoother finish, all with less strain on the machine tool. Pieces so thin as to bend under the cut when straight-fluted form cutters are used, can be form-milled successfully with the spiral-fluted cutter.

Spiral-fluted thread milling hobs also are rapidly replacing the straight-fluted thread hobs on many operations. Increased speeds and feeds together with a longer life between grinds are obtained through the use of these spiral-fluted thread hobs.

There is an increasing demand for high-speed steel ground-thread milling hobs where great accuracy is necessary. As high-speed steel cut-thread hobs mill approximately the same number of pieces per grind as the expensive ground-thread hobs, the latter should be used only where the accuracy of the products demands this additional expense. Improved methods of threading hobs and the employment of modern heat-treating methods result in the production of high-speed steel cut-thread hobs of marked accuracy as regards thread form, lead error, etc.

It should be remembered that fine-edged metal-cutting tools designed for use on ferrous materials such as cast and malleable iron, machinery steel, alloy steels, etc., not always will give satisfactory service when used on non-ferrous materials such as brass, bronze, copper, aluminum alloys, and the like. General-purpose tools such as are usually listed by most tool manufacturers work fairly well, however, on most of the materials mentioned. Where production requirements warrant the use of a tool designed especially for the job, considerable savings

can frequently be realized through the use of such special tools. The increased inventory cost is often found to be more than compensated for by the greater efficiency obtained through using tools properly designed for the material being fabricated.

Quality of Steel Used

The second factor that governs the life of metal-cutting tools is the quality of the steel used in the manufacture of the tool. During the past 10 to 15 years many interesting types of steels have been introduced by the steel mills, the big majority of these steels not finding wide favor due to tools made from the same not performing more efficiently than tools made from steels of the more common types. The various types of steel most commonly used today in the manufacture of metal cutting tools are as follows:

PLAIN CARBON TOOL STEEL

Carbon Per cent	Manganese Per cent	Phosphorus Per cent	Sulphur Per cent	Silicon Per cent
1.10–1.20	0.30 max.	0.020 max.	0.020 max.	0.30 max.

ALLOY CARBON TOOL STEELS

Carbon Per cent	Manganese Per cent	Phosphorus Per cent	Sulphur Per cent	Silicon Per cent	Chromium Per cent	Vanadium Per cent	Tungsten Per cent
1.05–1.15	0.30 max.	0.020 max.	0.020 max.	0.25 max.	None	0.15–0.20	None
1.10–1.20	0.30 max.	0.020 max.	0.020 max.	0.25 max.	0.40–0.60	None	None
0.85–0.95	1.20–1.40	0.020 max.	0.020 max.	0.25 max.	0.40–0.60	None	0.40–0.60
0.85–0.95	1.60–1.80	0.020 max.	0.020 max.	0.25 max.	None	None	None
0.85–0.95	1.60–1.80	0.020 max.	0.020 max.	0.25 max.	None	0.15–0.25	None

The first type of alloy steel mentioned is excellent for tools where breakage is the usual cause of failure. This vanadium tool steel is also recommended for so-called shock tools.

The second type under the heading "Alloy Carbon Tool Steels" is at times used for tools where a slightly greater depth of hardening than can be obtained from the plain carbon tool steel is desired. Chromium also silkens the grain to quite an extent, thus producing a much more velvety type of fracture after correct hardening than is to be found in the simple carbon tool steel.

The last three types of alloy carbon tool steels are commonly known as oil-hardening, non-deforming steels. Tools that will permit of a minimum size change in hardening usually are made from one of these three types. Experience has shown, however, that the usual types of water-hardening tool steels stand up better in service than the oil-hardening brands.

High-Speed Steels

The big majority of high-speed steel metal-cutting tools in use today are made from high-speed steel of the following average composition:

Carbon	Chromium	Vanadium	Tungsten
%	%	%	%
0.70	4.00	1.00	18.0

This type of steel is the simplest to harden of all the different kinds of high-speed steels offered the trade. Tools made from it give satisfactory service when used under normal conditions, regardless of the type of material being machined. There has been an increasing demand during the past two or three years, however, for 18 per cent tungsten high-speed steel containing from 5 to 10 per cent cobalt, especially for use in lathe tools, boring tools, tool bits, etc. Tools made from this cobalt steel must be ground all over after the hardening operation, due to the deep soft skin produced in hardening. Turning tools made of this high-cobalt high-speed steel give excellent service on all different types of materials, but especially when used on heat-treated alloy steels and other materials that are difficult to machine.

There continues to be a slight demand for the so-called low-tungsten, high-vanadium type of high-speed steel (13 per cent tungsten, 2 per cent vanadium), especially for turning tools of various types. This steel is considerably more difficult to harden properly than the more commonly used 18 per cent tungsten material, its general use therefore being somewhat restricted.

The high-carbon, high-chromium type of steel is being used to a great extent today for blanking and lamination dies, as well as for parts where resistance to wear is of primary importance. This material contains from $1^1/_2$ to $2^1/_4$ per cent carbon and

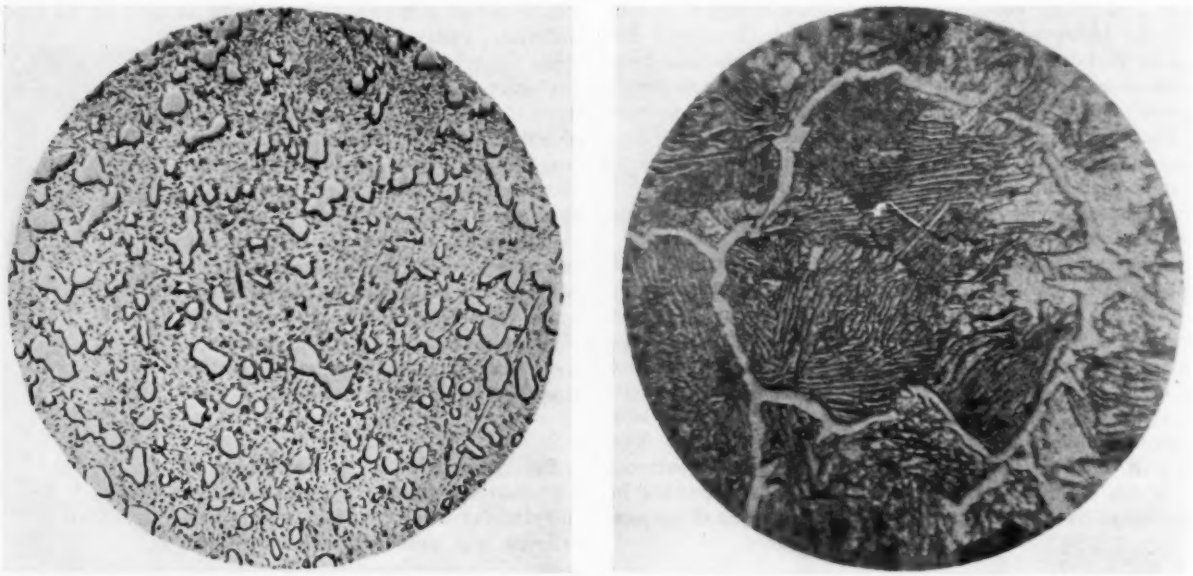


Fig. 3 Microstructures of Plain Carbon Tool Steel—at Left, in Proper Condition for Machining and Hardening; at Right, Finish Worked at Too High a Temperature by Mill and Improperly Annealed

11 to 13 per cent chromium. This high-chromium alloy does not possess the toughness, red hardness properties, and cutting qualities of high-speed steel, so is not recommended as a substitute for high-speed steel for metal-cutting tools.

Practically all carbon, alloy, and high-speed tool steels are melted today by the electric-furnace process, using the Heroult three-electrode type furnace. Ten years ago the crucible process was the one employed by the mills in this country for the melting of these high-grade steels. It is the opinion of the author that the electric-furnace-melted material is of a uniformly higher quality than the crucible-melted steel, this being especially noticeable in the alloy and high-speed tool-steel types.

From time to time, the small-tool manufacturer is requested to furnish semi-high-speed steel tools. A careful study of a large number of the various types of steels introduced by the steel companies during the past few years shows no steel possessing the necessary properties for this purpose. Just because a low-tungsten steel tool ($1^1/_2$ per cent to 2 per cent tungsten) gives a red spark when touched to the grinding wheel is certainly no reason for expecting the tool to have high-speed steel properties. In fact, tools made from this low-tungsten tool steel cannot be run at any higher speeds, feeds, etc., than can plain carbon-steel tools and moreover do not possess as great toughness, due to the increased hardness penetration. These remarks regarding red-hardness properties and toughness apply equally well to many other so-called semi-high-speed steels tested.

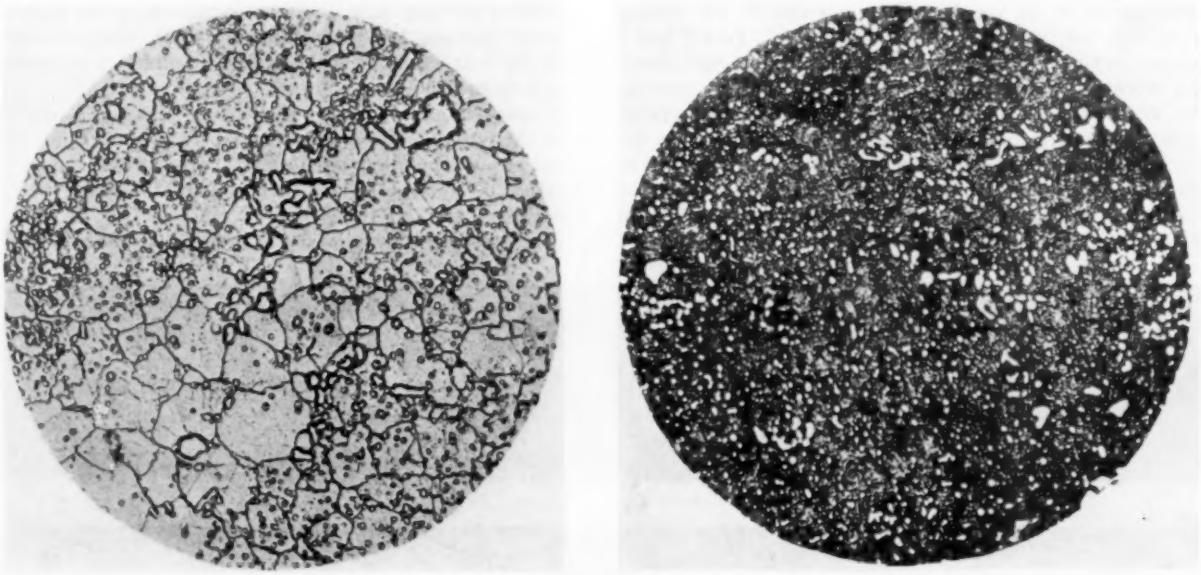


FIG. 4 MICROSTRUCTURES OF HIGH-SPEED STEEL QUENCHED FROM PROPER TEMPERATURE—AT LEFT, NOT DRAWN; AT RIGHT, FOLLOWED BY A 1100 DEGREE FAHRENHEIT DRAW

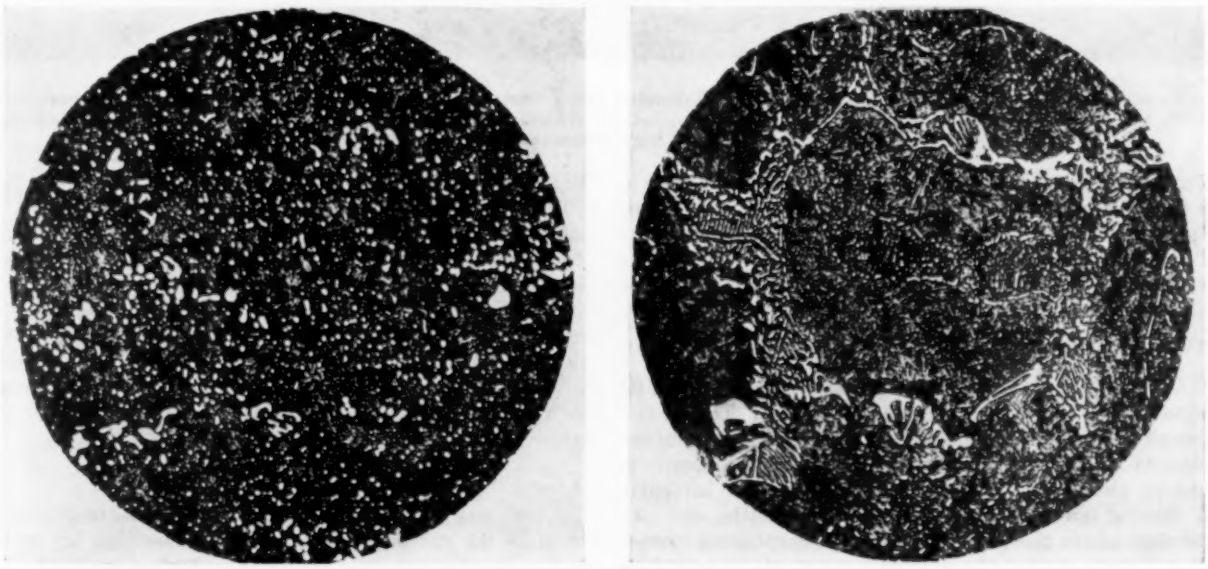


FIG. 5 MICROSTRUCTURES OF LONGITUDINAL SECTIONS CUT FROM $5^1/_2$-INCH DIAMETER STOCK—AT LEFT, HIGH-SPEED STEEL FORGED BLANK; AT RIGHT, HIGH-SPEED STEEL BAR STOCK

Incoming shipments of tool steel should be given a careful microscopic inspection. A lot of steel may be of the correct chemical composition, have the proper hardness, and possess a satisfactory fracture after the standard hardening treatment, but unless the structure as revealed by the microscope at magnifications from 100 to 1000 times is satisfactory, tools made from this material will not give satisfactory results. Figs. 3, 4, and 5 are representative of the important information one obtains only through the proper use of the microscope. Figs. 5 and 6 illustrate the necessity of using high-speed steel forgings instead of bar stock for the manufacture of cutters, hobs, etc., in the larger diameters. It is not possible at the present writing for the steel mills to produce high-speed steel-bar stock in the larger sizes that is free from harmful segregations, etc. The fracture of tools made from these larger bars is extremely coarse as compared with the velvety type of fracture obtained when forgings are used.

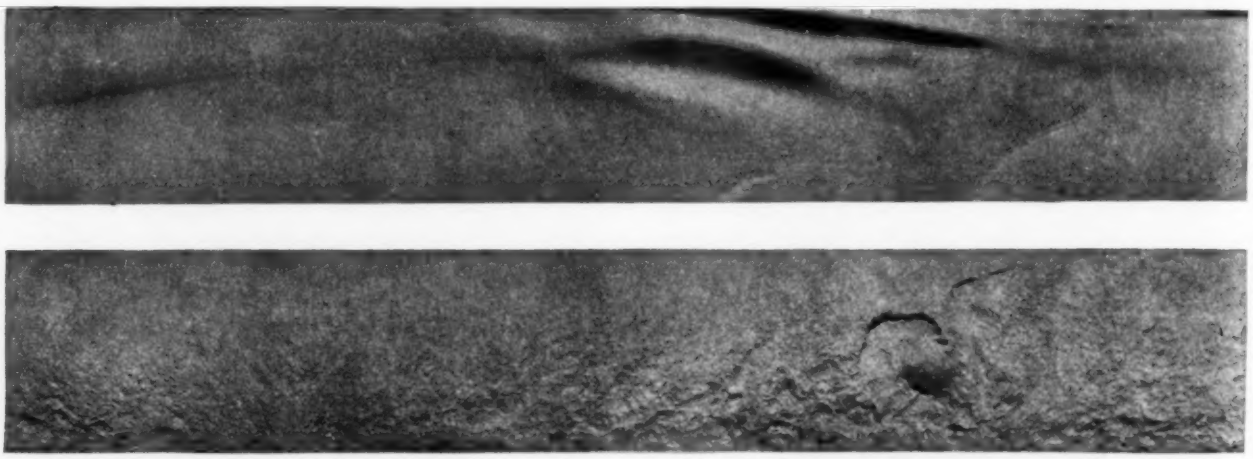


FIG. 6 ABOVE, FRACTURE OF A 5 1/2-INCH DIAMETER HARDENED HIGH-SPEED STEEL FORGING—NOTE FINE PORCELANIC STRUCTURE; BELOW, FRACTURE OF A HARDENED DISK CUT FROM A 5 1/2-INCH DIAMETER BAR OF HIGH-SPEED STEEL—NOTE COARSE-GRAINED WOODY STRUCTURE

The importance of using only steel of the best quality in the manufacture of high-grade metal-cutting tools cannot be stressed too strongly. So much depends on the kind of raw material used that every precaution should be taken to insure the use of only carbon, alloy, and high-speed tool steels of the highest quality obtainable.

HARDENING TREATMENT

Often one hears the statement, "a tool is no better than its heat treatment." A very true expression, indeed. The hardening room of a small-tool plant is one of the most important departments in the factory. Expense should not be spared in equipping this department with the best types of hardening and drawing furnaces, pyrometers, quenching baths, etc. All furnaces should be provided with automatic temperature control, time and program clocks for each operator should be furnished, quenching baths should be maintained at the correct temperature through efficient cooling systems, and all pyrometers should be frequently checked against master instruments, etc. Proper hardening instructions should accompany each order of tools, these instructions preferably being issued by the metallurgical department. Guesswork by the man at the fire thus is done away with. After hardening and drawing, the tools should be checked carefully for hardness, using special files provided for this purpose in conjunction with the Rockwell hardness tester.

The lead bath is used by most manufacturers in connection with the hardening of carbon-steel tools such as taps, dies, drills, reamers, etc. The temperature used varies from 1440 to 1500 deg. fahr., brine being the quenching medium most commonly employed. The oil bath, salt bath, or oven-type furnace may be used for the tempering of the tools after the quenching operation.

High-speed steel tools of the 18 per cent tungsten type should be hardened as follows: They should be preheated to a temperature of 1500 to 1600 deg. fahr. and then transferred to the high-temperature furnace, the temperature of which is approximately 2350 deg. fahr. As soon as the tool reaches the furnace temperature it should be quenched in oil.

Drawing should be from 1050 to 1200 deg. fahr., depending upon the type of tool, and then they should be air cooled. High-speed steels containing cobalt must be quenched from a temperature of approximately 2400 deg. fahr. to obtain the most satisfactory results.

High-speed steel containing the proper percentage of carbon and quenched from 2350 deg. fahr. will show the same hardness after a 1050 deg. fahr. draw as in the undrawn condition. Considering that high-speed steel tools properly hardened and high drawn possesses twice the strength of tools in the undrawn state, without any sacrifice of hardness, it is the practice of most manufacturers to give all high-speed tools the high-draw treatment. It is important that high-speed steel tools be heated to a temperature in excess of 2300 deg. fahr. when open-fire hardened, as the cutting efficiency of a high-speed tool increases with increased hardening temperature up to the temperature where slight coarsening of the grain takes place.

GRINDING OF TOOL-STEEL TOOLS

A few years ago an article appeared in one of the trade journals in which the statement was made that more tools are spoiled in the grinding operation than during hardening. While this statement is somewhat exaggerated, in the author's opinion, it is true that many tools are ruined daily due to improper grinding. The use of too hard a wheel, a loaded wheel, or improper cooling during the grinding operation, are all contributing factors to grinding checks, soft skin, and similar evils. Fig. 7 shows three tools purposely ruined through improper grinding, then given a hot-acid etch treatment to make the lacework of grinding checks more visible. It is more economical to use a few additional wheels of the proper grain and grade monthly than to attempt to economize through the use of harder wheels merely

because they do not wear away so rapidly. The grinding-wheel salesmen are in a position to instruct the manufacturer as to the different types of wheels best suited for the various grinding operations.

In connection with the grinding of high-speed steel tools, a few grinding-room foremen share the opinion that a straw or blue color appearing on the tool after the grinding operation will

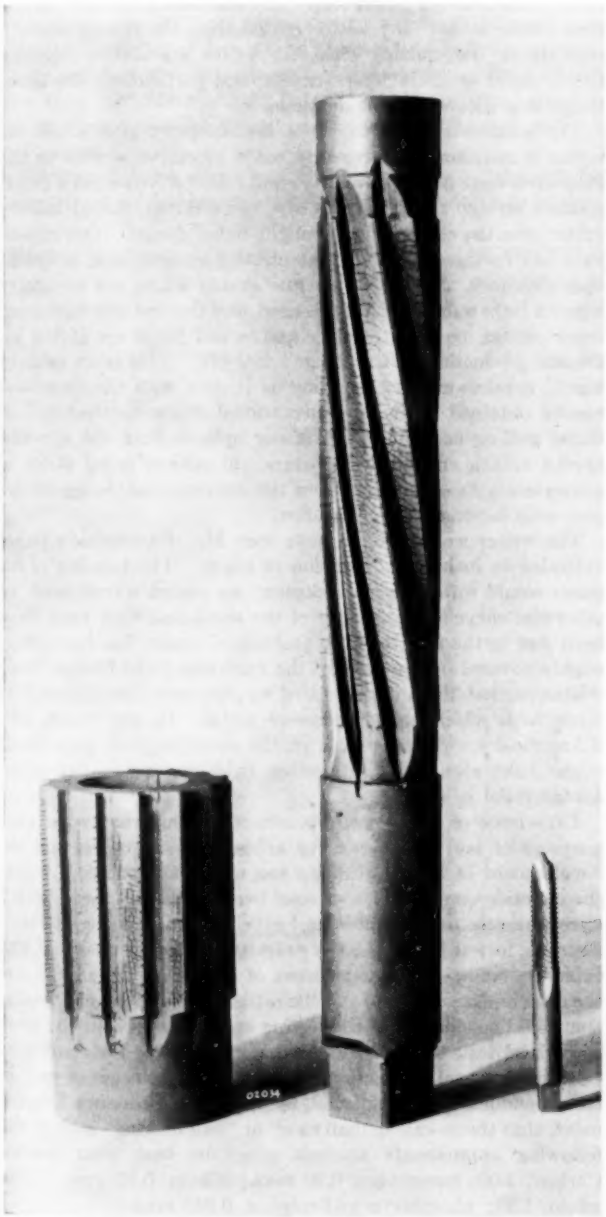


FIG. 7 CRACKS THAT RESULT FROM IMPROPER GRINDING

not affect the performance of the tool, considering that the tool had passed through that range when being high drawn. While it is true that a 1100 deg. fahr. draw is a much higher temperature than the blue color would indicate, frequently a high-speed tool showing color after the grinding operation can be filed readily along the cutting edges. This means that the cutting edges of the tools were heated to a temperature in ex-

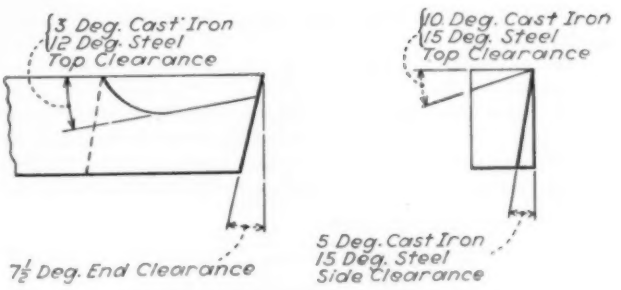


FIG. 8 LATHE TOOL FOR HEAVY WORK

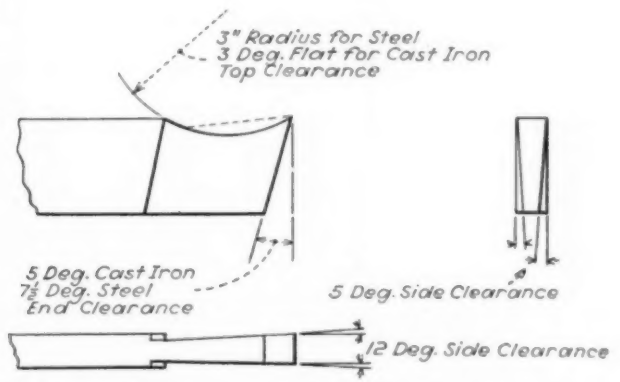


FIG. 9 PARTING TOOL FOR HEAVY WORK

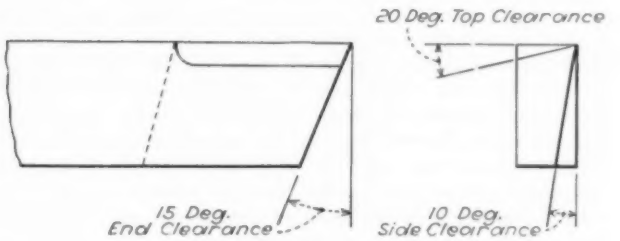


FIG. 10 LATHE TOOL FOR TURNING CARBON TOOL STEEL AND HIGH-SPEED STEEL

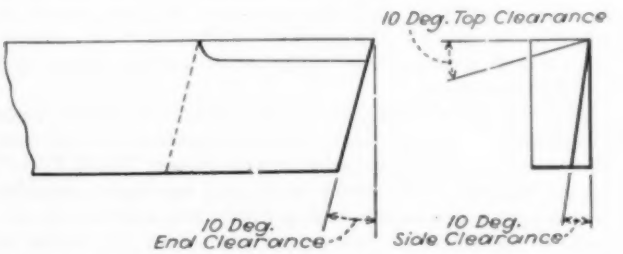


FIG. 11 LATHE TOOL FOR TURNING HIGH-CARBON HIGH-CHROMIUM ALLOYS

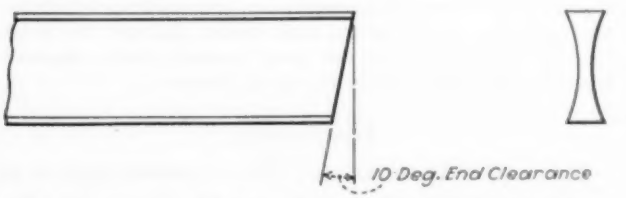


FIG. 12 JOHNSON CUT-OFF BLADE FOR GRIDLEY MACHINE

cess of the 1100 deg. fahr. draw during the grinding operation, thus ruining the tool.

Cutting Fluids

The cooling and lubrication of cutting tools is of such importance that The American Society of Mechanical Engineers appointed a sub-committee several years ago to prepare a report on this all-important subject. This committee has not completed

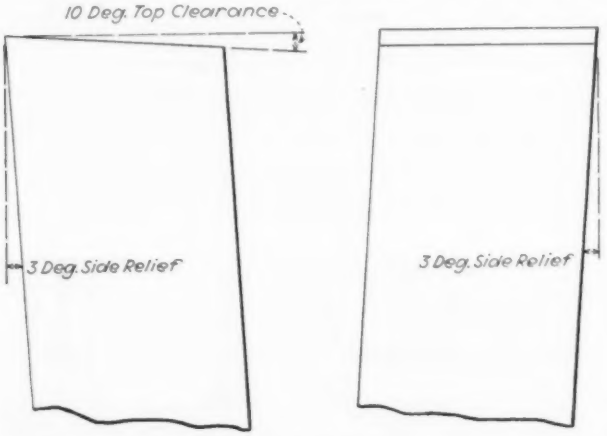


Fig. 13 Turning Tool for Carbon Tool Steel and S.A.E. No. 6150 Steel Used With Gridley Machine

its work as yet, but a progress report was presented about two years ago.

The failure of many metal-cutting tools is due to the coolant used. Some machining operations such as threading and milling require a good grade of sulphur-base oil as the cutting fluid; other types of jobs, a soluble oil; some special operations call for pure lard oil; aluminum-alloy jobs, kerosene plus a small amount of machine oil, etc. A generous flow of the cutting fluid should be directed properly on the work at all times. The kind of cutting fluid used not only affects the life of the tool, but has an important bearing on the finish obtained. On some threading jobs the size produced is dependent in some degree on the type of cutting fluid used.

Conclusions

The three factors that govern the life of tool-steel tools, namely, design, steel, and treatment, have all been covered in detail in this paper. The necessity of grinding tools correctly as well as the importance of employing the correct cutting fluid are also touched on.

Machine tools rightfully have been called, "The Master Tools of Industry." Of what use would these master tools be without their metal-cutting tools, unfortunately named "Small Tools." Many concerns, after careful study, will purchase a machine tool at a price running into four or five figures, and then use any make of small tool on this expensive machine that might be handy or that can be purchased at a deceptive price. "A machine is no better than its tool equipment," is a true statement, worthy of the careful consideration and study of the metal-working trade. While tremendous progress has been made in developing and improving metal-cutting tools during the past ten years, the author feels confident that proportionally greater improvements will be made during the coming decade.

Discussion

Frederick S. Blackall, Jr.[2] The d'Arcambal paper is an interesting survey of the high lights of the small-tool field. Because of the breadth of the subject, any brief comment must of necessity be specific rather than general.

It is refreshing to hear as eminent a metallurgist as Mr. d'Arcambal scouting the alleged merits of some of the so-called "alloy carbon tool steels." Where surface wear is a factor, many of these latter-day developments have distinct merit; but it is the writer's opinion and experience that for cutting tools any compromise between a true high-speed steel and a straight-carbon steel yields little if any better results than the straight-carbon steel itself. For cutting tools, the writer has always regarded the so-called semi-high-speed steels (and particularly the lower tungsten grades) as simple delusions.

The engineering department of the company with which the writer is associated has recently made extensive studies in the field of thread milling-cutter design. Mr. d'Arcambal's paper touches on the superiority of the spiral-fluted thread milling cutter over the conventional straight-fluted design. Our experience has further indicated that existing designs lack sufficient chip clearance, due to the narrow gashes which are necessary where a large number of flutes is used, and that cutters possessing fewer cutting teeth with wider gashes and lands are giving increased production and prolonged tool life. This is an entirely logical conclusion, corresponding as it does with the improved results obtained with the conventional coarse-toothed spiral-fluted milling cutter; and it is our opinion that the superior thread milling cutter of the future will possess spiral flutes in somewhat reduced number from the conventional design of the past with increased chip clearance.

The writer would like to have seen Mr. d'Arcambal's paper extended to include a discussion of gages. The heading of his paper would warrant such inclusion; its omission from such an otherwise encyclopedic survey of the small-tool field may have been due to the fact that the problem of design has been thoroughly covered by the work of the American Gage Design Committee or that the author desired to purposely limit himself to those tools which actually remove metal. In any event, Mr. d'Arcambal's wide experience in the metallurgy of gage steels might have shed some interesting light on one or two other controversial points.

Experience of recent years points to the superiority for gage purposes of tool steels over the milder carbon grades, and the development of thread grinding has made it possible to utilize the superior qualities of tool steel even in thread gages, which were formerly made almost exclusively of carburizing steels, a desirable forward step. A few years ago, several concurrent and extensive studies were undertaken of gage-wear factors. From the mass of data collected and the rather widely divergent results, one would conclude that the testing apparatus used in the more theoretical tests could hardly have approximated the conditions existing in actual production. It was the conclusion of certain tests, conducted, it is believed, by the Army Ordnance Department, that the so-called "ball race" or "ball bearing" steel of the following approximate analysis gave the best wear results: Carbon, 1.00; manganese, 0.30 max.; silicon, 0.25 max.; chromium, 1.40; phosphorus and sulphur, 0.040 max.

The disadvantage of this type of steel is its tendency toward a superficial surface softness—appropriately described as a soapy condition—which made gages of this material not only unpopular with the trade, but, in the writer's opinion, inferior to tools possessing a glass-hard surface. The painstaking if highly theoretical experiments of Messrs. French and Herschman, of the Bureau of Standards, produced an even more startling conclusion—namely, that a file-soft gage is superior to a file-hard gage.

Extensive study of the subject, corroborated by wear tests in production, which, in the writer's opinion, are the only wear

[2] Vice-President and General Manager, Taft-Peirce Mfg. Co., Woonsocket, R. I. Mem. A.S.M.E.

tests yet devised which yield accurate and anywhere near consistently uniform results, leads to stating the unequivocal opinion that the hardest gage is the best gage where wear factors alone are considered. Rockwell readings of C 63 to 65 for plain gages, C 60 to 62 for ordinary thread gages, and C 58 to 60 for pipe-thread gages appear to be desirable minima. A certain sacrifice of hardness is necessary in thread gages, where the element of toughness is necessary to prevent chipping.

For those who favor the chrome manganese steels for gage purposes, the second analysis under the heading "Alloy Carbon Tool Steels" in Mr. d'Arcambal's paper will be found to produce a very satisfactory steel, possessing the wearing qualities of ball race steel without the latter's tendency to soapy hardness. The chromium content should be kept nearer 0.40 than 0.60 for best results.

For thread gages, where lead changes must be minimized, the conventional oil-hardening tool steels prove quite satisfactory. The higher manganese, tungsten-free type has proved superior to the tungsten grade in our experience.

No discussion of gages at this juncture of engineering history should close without a fervent plea for the universal adoption of the gage design standards of the American Gage Design Committee, which have been recognized by this Society.

[NOTE: A discussion of this paper jointly with the paper of Becker, Gordon, and Wissler by A. K. Dean will be found with the discussion of the latter paper.]

AUTHOR'S CLOSURE

Mr. Blackall's comments were very much appreciated. The author agrees with Mr. Blackall that the value of this paper would have been materially increased should the discussion have included the all-important subject of gages. The author was requested, however, to confine his remarks to metal-cutting tools made of carbon, alloy, and high-speed tool steels.

MSP-53-12

A Survey of Surface Quality Standards and Tolerance Costs Based on 1929-1930 Precision-Grinding Practice

By R. E. W. HARRISON,[1] CINCINNATI, OHIO

In an earlier paper presented at a meeting of the Chicago Section of the A.S.M.E. on March 18, 1929,[2] the author reviewed the process of evolution through which the art of precision of cylindrical grinding had passed up to that date. This paper reviews some of the defects which exist in the present-day accepted standards of finish of mechanical components, and represents an endeavor to maintain continuity by carrying the discussion through to the present-day standards of finish attainable by the abrasive process. It is pointed out that it is useless to specify extremely close limits of accuracy unless the measurements are made across basically plane surfaces.

Size control and its effect on cost of production, as well as the length of useful life, is discussed, and better finishes are advocated as a means of reducing the ill effects of high initial wear. A scheme for the establishment of definite standards of finish is put forward, and a finish-calibrating instrument is described. Emphasis is laid on the fact that there is no available short-cut method of obtaining ultra-fine-finished surfaces with a minimum of initial wear, and the author points out that standardization of finishes can be attained at low cost with little disruption of present methods; also that the subject of longer useful life to mechanical components is closely interlocked with the provision of a better standard of finish and the attendant benefits derived from close adherence to true limits of accuracy.



THE urge for betterment which leads to perfection brings our past performances under the most critical review. Sometimes this is the result of failure measured in terms of longevity or the inability of some mechanical piece of work to fulfil its function; sometimes it is the urge of the designer to produce a more mechanically perfect and lasting job; sometimes it is the desire of the production engineer to excel in his art and produce something better than that which has gone before; and it is often the urge brought about by intensive competition which makes it a commercial necessity for one individual or concern to produce something better than that which has hitherto been presented in an endeavor to capture a market and its attendant profits.

Whatever the cause may be, the fact remains that the world is marching steadily forward. While the credo of many manufacturers is to make and produce within the widest permissible limits of accuracy and finish consistent with the established standards of efficiency for their product, there are others, and these latter probably constitute the majority, who, with a genuine desire to produce a better job, strive all the time to improve on past performance.

Ever since the first lathe and the first boring machine, and in fact the first tool of any of the now-established types, were put into operation, the finish and accuracy of their product have been matters of speculative discussion.

There of course has always been a compromise between the practical and the ideal. But although it is invariably conceded that perfection is unattainable, this fact has never prevented striving for that goal.

This paper represents an endeavor on the part of the author to record, in a useful way, some of the progress which has been achieved up to date.

SECTION A

1 Surface Irregularities on Ground and Lapped Work

The fact that so many engineers supplement their eyesight with eyeglasses and spectacles is in itself evidence of a growing

[1] Engineering Director, Cincinnati Grinders, Inc. Mem. A.S.M.E.; Chairman, Cincinnati Local Section. Mr. Harrison received his education at Grove House High School and Crescent College, both of Manchester, England, after which he attended Manchester College of Technology, where he took a regular course in mechanical engineering, including heat engineering. Mr. Harrison then entered the employ of the Churchill Machine Tool Co., Ltd., at Pendleton, Salford. Here he received training in erecting and machine-shop departments on regular schedule, and during 1911 he was appointed foreman of the turret-lathe and automatic-screw-machine department. Mr. Harrison held this position for about 18 months, and later was given charge of the jig-tool and work-progressing department, which position he held until the outbreak of the World War in 1914. In 1923 Mr. Harrison was appointed assistant works manager of the Churchill Machine Tool Co., Ltd., and was given the assignment of building and equipping the company's new works erected at Broadheath, Manchester.

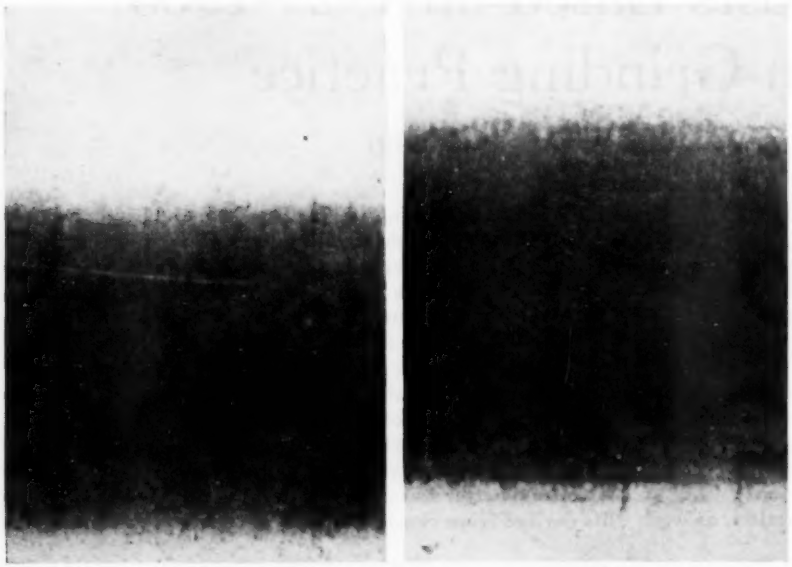
Mr. Harrison was responsible for purchases, supervision, and all contracts, and generally functioned as assistant to the general works manager. He was made special technical representative of the Churchill company to spend six months in the United States and Canada, surveying the two countries as a possible outlet for British machine tools. During the remainder of 1923, Mr. Harrison visited France, Holland, and Belgium in a similar capacity. In January, 1924, he was appointed commercial manager of the Churchill company with responsibility for sales. Mr. Harrison joined the engineering staff of the Cincinnati Milling Machine Co. in July, 1926, and on the formation of the company's subsidiary, Cincinnati Grinders, Inc., he was made a member of the board of directors and assigned the duties of chief engineer.

[2] "Precision Cylindrical Grinding," Trans. A.S.M.E., Sept.-Dec., 1929, MSP-51-18.

Contributed by the Machine Shop Practice Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

dissatisfaction with the limitations of the naked eye. Hence the author feels confident that no one will feel that the discussion is being made of too academic a nature if the microscope is accepted as a highly desirable and, in many cases, necessary tool in the inspection department; for without the aid of this tool, the naked eye is insufficient to gage qualities of surfaces called for by present-day standards.



A *B*

FIG. 1 THE SAME PIECE OF WORK
(*A*, as viewed with the naked eye; *B*, defects revealed by the camera.)

A simple experiment with a camera, photographing a piece of ground work, will be sufficient to convince the most skeptical that the sight of the average individual can be deceived to a surprising and disconcerting extent.

The outside diameter of an 8-in. diameter ball race, when viewed by the naked eye, appeared to have a highly satisfactory surface, as shown at *A* in Fig. 1. It, however, took on a quite chattered or broken appearance, as shown at *B*, when photographed. This photograph is not an enlargement and represents the actual size of the work.

It might be argued that in the quest for the true plane surface, one is getting out of the realm of practical workshop possibilities and introducing laboratory methods and discussions. But, in the author's opinion, the answer to this is the increased mileage available on the modern automobile due to the higher initial finish and accuracy built into the components generating the power and carrying the load, as compared with the relatively short total of mileage and noisy, inefficient performance of the automobile of even three years ago.

That there is an appreciation of the fact that minute surface irregularities result in rapid wear, irrespective of lubrication, and that there is a definite though now limited market for the near-perfect piece of mechanism, is borne out by the fact that there are a few automobile manufacturers who have built up a profitable business by building this type of mechanism only.

It is generally known and accepted that the method of removing metal which involves the use of a hardened tool, the nose of which penetrates the surface to be machined to the required depth, leaves a relatively torn and disrupted surface which has a high coefficient of friction and a relatively high initial wear.

It is not proposed in this paper to discuss the relative merits of machined and ground work, but to deal with the finer irregularities produced by the accepted finishing operation of today, i.e., grinding. Whether the work be hardened or soft, or cast iron, steel, or bronze, it has been proved that the cheapest way of securing accuracy within the closely specified limits and finish, up to an acceptable standard, is by means of the precision grinding machine.

This, then, is the starting point.

The modern grinding wheel, which is built up from an accurately sized, accurately spaced, and carefully bonded abrasive, is in itself a tool of precision, produced by its makers to within limits of accuracy controlled within thousandths of an inch. Examined by the naked eye after being mounted and trued in a modern precision machine, the working surface of this wheel presents a smooth and precise appearance. However, when this wheel and the work on which it is operated are viewed through the microscope, it will be realized that modern precision grinding is nothing more or less than a closely controlled milling operation in which each of the particles of abrasive presented to the work plows its quota of metal and leaves its resulting furrows.

While one may overlap and overlap the path taken by succeeding particles of abrasive, the resultant surface under a powerful microscope has much in common with a plowed field, Fig. 2. Sharp ridges of steel or iron exist on the surface of the work, and these must be removed before one can arrive at a plane surface which will have a maximum resistance to wear.



FIG. 2 PHOTOMICROGRAPH × 75
(Surface of cylinder ground to generally acceptable finish with 80-grit wheel.)

Some few years ago the research engineer of one of the auto-

mobile concerns making cars in the highest price class, in a fit of discontent with his company's product, inserted an electromagnet in the oil stream while a standard engine was undergoing its dynamometer test. The result of this simple little test was so startling that this concern almost ceased manufacturing until means could be found to solve the problem which was thus so suddenly thrown up to them. The electromagnet had drawn a considerable quantity of ferrous particles from the oil stream, and this was proof positive that the initial wear on all contacting members was astonishingly high. The remedy was obviously a better quality of finish than any which had hitherto been considered acceptable.

It was realized that a lapping compound composed of particles of iron, steel, and bronze was a dangerous element to introduce into any piece of high-class, high-speed machinery.

A world-famous maker of Diesel engines, of both the marine- and the land-operating type, found that reduction in initial wear due to bettered methods of finishing enabled him to control his sizes so accurately that he reduced the running-in time of his engines to one-third of that previously found necessary, and at the same time increased the power-producing efficiency of his product by 8 per cent.

A world-famous builder of locomotives made a test on a locomotive equipped with one cylinder which had been bored and another which had been bored and afterward ground. The locomotive was put on the road, and the cylinder heads were not removed for 14 months. Within two days of the inspection of this locomotive the process of finishing cylinders was changed to embody the improved method of finishing.

Much is heard about the appalling cost of rust or oxidation. Optimistic statisticians have even dared to put a figure on this wastage, which runs into billions of dollars. In the author's opinion it is not an exaggeration to say that this figure is at least equaled by the cost of largely avoidable wear. Hence there is what appears to be a sufficient and valid reason for the time given to the consideration of a subject, the economics of which involves such an immense sum of money.

Consider, for example, surfaces produced on cylindrical work, and discard for the moment the questions of accuracy as regards roundness and parallelism.

Fig. 3 shows a representative plan view of an automobile wristpin ground to a finish which is generally considered good and acceptable by manufacturers who judge their product only by the naked eye. The finish-grinding operation was performed with an 80-grit grinding wheel.

Fig. 4 shows the same piece reground with a 500-grit grinding wheel.

Fig. 5 shows the same piece after being put through a lapping operation in which the pin is held on a spider between abrasive disks that were mounted on vertical spindles.

Fig. 6 shows the same piece after being again passed through the grinding machine following on the lapping operation. It will be noted that the grinding operation with the 500-grit grinding wheel has not been continued long enough to remove all the marks left by the laps.

The hitherto frequent renewal of automobile wristpins resulted in intensive study being given to this member, with the result that a very large percentage of pins built into new automobiles and those used as replacements are either ultra-finish-ground with a 500-grit grinding wheel or are lapped.



FIG. 3 TYPICAL EXAMPLE OF A WORKING SURFACE ×100
(A good job of grinding, with no defects visible to the naked eye; ground with an 80-grit wheel.)

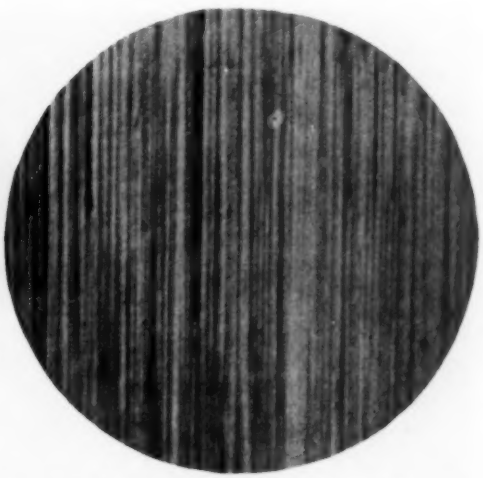


FIG. 4 A WORKING SURFACE PRODUCED WITH A 500-GRIT WHEEL ×100



FIG. 5 LAPPED SURFACE ×100

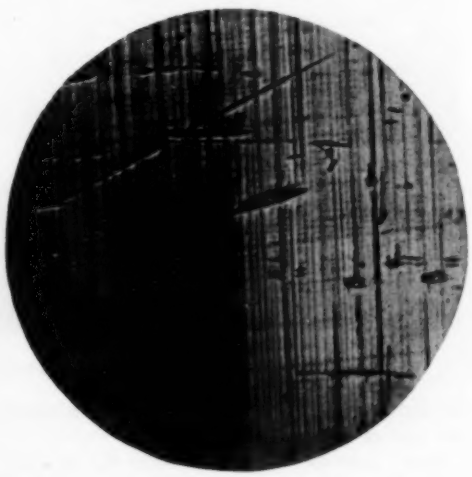


FIG. 6 APPEARANCE OF WORKING SURFACE AFTER LAPPING AND SUBSEQUENT GRINDING WITH 500-GRIT WHEEL ×100

The surfaces on automobile crankshafts also have come in for much critical review, and at the instance of one of the leading automobile manufacturers, Dr. C. B. Sawyer, of Cleveland, has made some interesting cross-sections which emphasize the necessity for better methods of finishing. (See Appendix 1.)

The method of obtaining these interesting specimens was to immerse the samples in a copper-cyanide plating bath and give an initial coating. This is followed by immersion in an acid copper-plating bath where a heavy deposition of copper is made. The samples are then removed and cut, using a hacksaw, the cutting being so performed that the copper is forced toward the iron or steel.

These samples clearly illustrate the amount of irregularity which must be removed from the surface of an automobile crankshaft before the pin can establish a smooth and unobstructed path. (See Appendix 1.)

Experience indicates that there is no royal road or shortcut to a really high finish. Such finishes must necessarily be built up by successive stages.

Grinding wheels designed for producing the highest class of finish are incapable of an attractively high rate of metal removal. Hence, a practice has grown up whereby the bulk of the grinding allowance is removed with a wheel of from 60 to 80 grit. An intermediate finishing operation utilizing a wheel of 120 to 250 grit is interposed to top off the ridges caused by the grain depth of cut of the roughing wheel, and the final finish is imparted to the work with a wheel which may vary from 250 to 500 grit, the final result depending upon the standard of finish necessary for the satisfactory accomplishment of the work.

Attempts have been made to effect a shortcut, but these have all resulted in the building up of a false finish. In other words, the ridges are turned over into the hollows, and in actual service the results obtained are unsatisfactory.

2 Scheme for Establishing Standard Finishes in General Engineering Practice

Through the period of years in which mechanical processes have been developing, no satisfactory method of gaging finishes, other than with the eye, has been developed. This means, also, that there is no definite standard of comparison on either a world, a national, or even a local basis.



Fig. 7 Rack of Plugs Used as Reference Standards of Finish

It is unfailingly true that, in those countries where high standards of workmanship have been established, engineers are more critical and demand a higher finish, but this in turn is again, and has always been, a matter of mental comparison, with no established basis on which to allocate a place to any one particular finish.

This matter has exercised some of the keenest minds in the industry and has been given very careful consideration by the American Standards Association.

In an attempt to obtain a solution to this problem in a practical workshop way, the author has adopted the following procedure:

A small rack, Fig. 7, carrying 1-in. diameter cylindrical plugs ground to the various progressive standards of finish has been placed in each of the following departments for reference: en-

gineering, grinding, inspection, standards, and laboratory.

The question at once arises as to how these standards are established, and this has been accomplished in the following way:

A finish-measuring device has been put into use, embodying an electromagnetic type sound detector or pickup as used in the better makes of electrically operated gramophones.

This instrument is connected up with a radio-amplifying unit and records minute vibrations caused by irregularities on the surface of the work to be inspected, reproducing them through a loud speaker.

with a piece to be tested, the loud speaker, and the millivoltmeter unit.

Fig. 9 shows a diagrammatic view of the instrument.

Attempts to measure the finish obtained on ground work by means of instruments based on the ability of the work to reflect from a measured source of light have proved inadequate, inasmuch as it is too easily possible to obtain false readings from surfaces which, while having the appearance of being highly polished, nevertheless contain comparatively deep furrows or scratches.

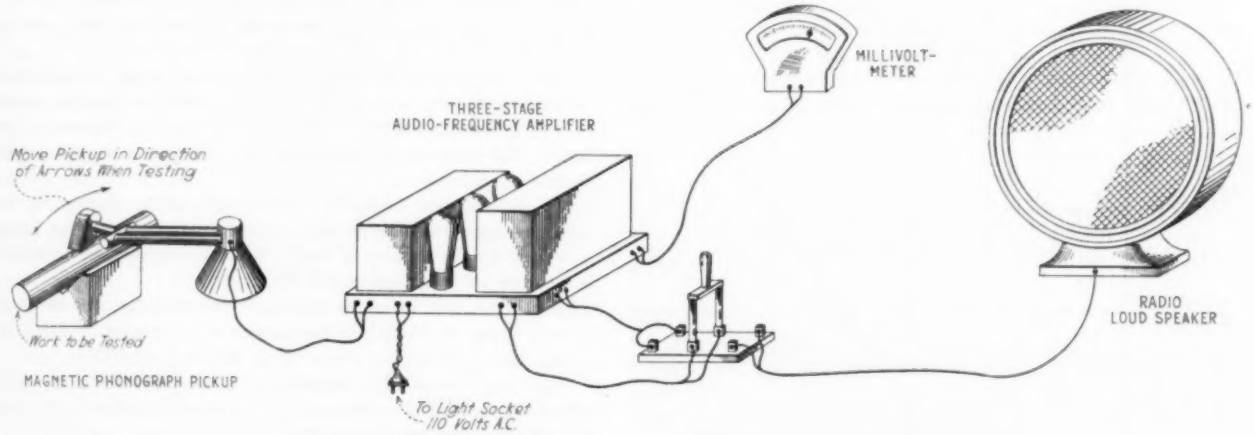


FIG. 9 DIAGRAMMATIC VIEW OF FINISH CALIBRATOR

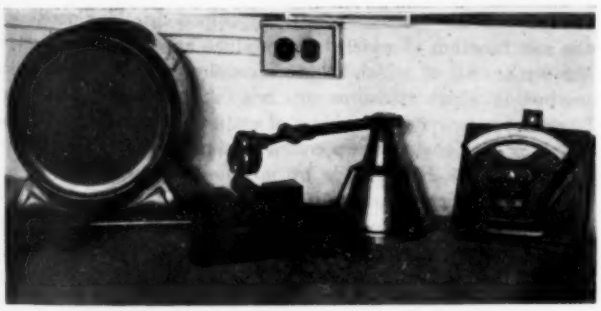


FIG. 8 FINISH-CALIBRATING INSTRUMENT WITH SOUND AND VISUAL RECORDERS

The contact with the work is made with a chisel-edged sapphire, and a millivoltmeter records the amplitude of a current which is directly proportional to the vibration, and which passes across the grid of a supplementary radio tube.

Work which is relatively roughly ground produces a deep, harsh vibration in the loud speaker and a relatively large movement of the needle on the millivoltmeter scale. A finely ground piece of work produces a keen high note characteristic of minute vibration, and results in a relatively small swing of the millivoltmeter needle.

From the foregoing description, therefore, it will be seen that there is available both sound and visual measurement on a definite scale to supplement the microscope and the naked eye.

The instrument is extremely sensitive and particularly well suited to picking out that class of unsatisfactory work which, in the every-day phraseology of the mechanic, is "highly polished and deeply scratched."

Fig. 7 shows the rack of finished samples, with three tiers each, containing soft, case-hardened, and oil-hardened steel samples.

Fig. 8 shows the work-finish-testing instrument in contact

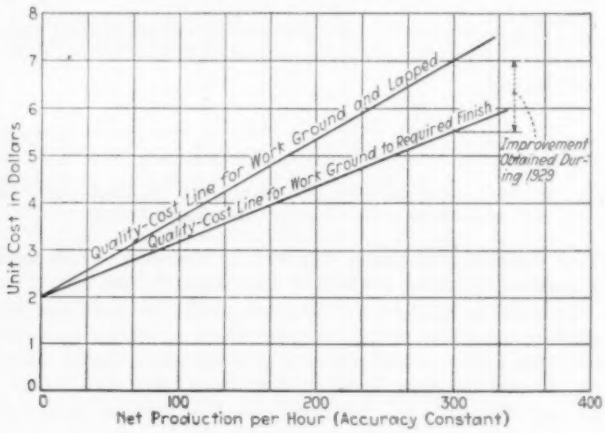


FIG. 10 QUALITY COST DIAGRAM ILLUSTRATING REDUCED COST OF IMPROVED FINISHES DUE TO IMPROVED METHODS

(Substitution of high-production grinding operation in place of lapping and polishing; use of heavier and mechanically improved machines; use of greatly improved fine grit grinding wheels; improved methods of truing grinding wheels.)

3 RELATIVE COSTS OF PRESENT-DAY STANDARDS OF FINISH

It being accepted that there is no available shortcut to high finish, it logically follows that the higher the finish the higher the cost on any given piece of work. However, much has been done in an endeavor to improve the standard, and the diagram in Fig. 10 serves in a general way to indicate what has been accomplished during the last year or eighteen months.

Case-hardened finish No. 3, which is equivalent to what is now generally known to the trade as a commercial finish, is accomplished on either the centerless or center-type cylindrical grinding machines, by rough and finish grinding the work with the same wheel, dressing the grinding wheel openly for the roughing operation and closely for the finishing operation.

Finish No. 2 is obtained in an identical way, but calls for the expenditure of from 15 to 20 per cent more time in the finish-grinding operation only.

Finish No. 1 is a refinement of finish No. 2 and calls for an additional grinding operation with a wheel of approximately 250 grit. This finish doubles the time required for the previous finishing operation, but does not in any way affect the time required for the rough-grinding operation.

The ultra-fine finish represented by the samples in the top row of the case calls for still another grinding operation, utilizing a ma-

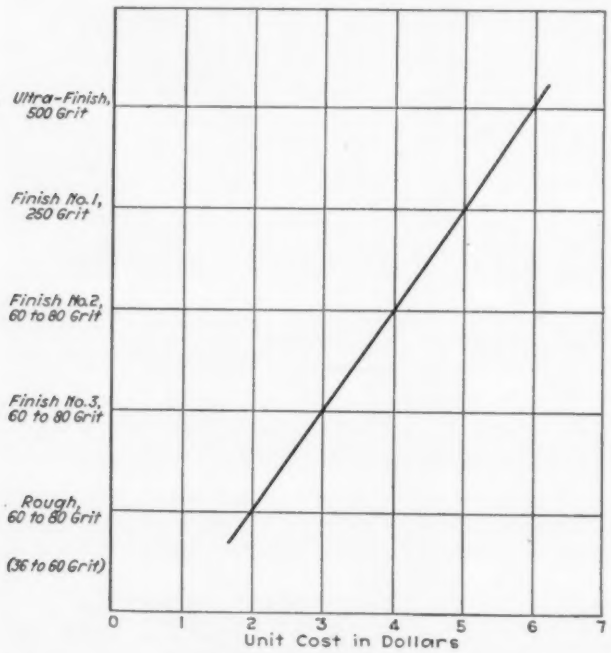


FIG. 11 QUALITY COST DIAGRAM INDICATING PROGRESSIVELY INCREASING COST OF VARIOUS GRADES OF FINISH

chine designed and closely adjusted for this work. This finish is obtained by using a wheel of approximately 400 grit.

To state the matter in another way, finish No. 3 calls for two grinding operations with the same wheel. Finish No. 2 calls for two grinding operations, also with the same wheel. Finish No. 1 calls for three grinding operations using two different wheels. The ultra-fine finish calls for four grinding operations involving the use of three different wheels. All the finish-grinding operations occupy about the same length of time and the cost of production is in proportion to the finish produced. This is illustrated graphically by Fig. 11.

SECTION B

1 ACCURACY CONSIDERED AS A COMMERCIALLY ATTRACTIVE FACTOR IN WORKS MANAGEMENT

The relationship between quality and finish, as discussed in the foregoing section of the paper, and accuracy, considered as a commercially attractive factor from the point of view of the works management, may at first appear to be somewhat obscure.

However, experience indicates that the two subjects are very closely related by that relatively indeterminate factor known as initial wear. Until one is in a position to commercially produce work with practically no initial wear, it might justly be said that he is being deceived in the belief that work is being held within the specified limits of accuracy stated on the drawing.

The effects of this knowledge of the existence of high initial wear can be seen on every hand.

(*a*) Designers frequently tighten up their limits in the hope that when the initial wear has taken place they will still have mechanisms which operate without shake or without losing gas, air, or oil pressure.

(*b*) Designers frequently call for extremely expensive hand-lapping operations, knowing that this, being a slow process, offers maximum insurance against the assembly of parts which would otherwise retain possibilities of high initial wear.

(*c*) Makers of twist drills and plug gages insure that their products have a maximum useful life before the fuzz is worn off them and they are reduced to the low limit.

(*d*) Engine builders make their pistons larger in diameter in relation to the bore to insure that when the engine comes off the dynamometer test it will not have too much piston slap.

(*e*) The preloading of anti-friction bearings is another effort in this direction.

(*f*) Builders of high-class machinery erect, dismantle, inspect, reerect, run, dismantle, inspect, reerect, and run again to check up the results of initial wear known as the bedding-in process on their shafts and bearings.

In other words, it might truthfully be said that the test run of any piece of high-class machinery is a species of teething period during which its whole future lies in the balance.

That the commercial hazards of this process are to some extent recognized is evidenced by the rapid increase in the use of high-class honing equipment.

A high-class company has built up an attractive and economically sound business in lapping machines. Another company has put on the market specially equipped grinding machines, the sole function of which is to produce an ultra-fine finish on the work. All of which is proof conclusive that accuracy of product in which measurements are taken between plane surfaces and not over rough ridges of material is recognized by the discriminating engineer as a commercial necessity.

The practice, however, is by no means general, and so far has been restricted to relatively few of the many thousands of manufacturers producing what they honestly believe to be a quality product.

2 THE COST OF ACCURACY IN GROUND WORK

One of the factors most closely affecting the manufacturing success of any modern organization is the intelligent use of the limit system, for in the specifying of close or wide limits the engineering department either can make or break the successful operation of any plant.

It is obviously most sound for the designer to specify the widest possible limits of accuracy consistent with the satisfactory functioning of the piece or mechanism under consideration.

There is evidence, however, that the question of initial wear does not always receive the consideration which its importance entitles it to, and it is becoming increasingly necessary for designers also to specify the class of finish on the bearings and sliding surfaces which operate under load.

The absence of a definite standard of finish has been a great obstacle, but it is hoped and anticipated that research work will shortly clear away this barrier.

Unfortunately, the subjects of finish and accuracy within closely specified limits have been somewhat confused in the minds of many designers, and this has led to the belief in many quarters that because, say, a locomotive is built up of parts with wide limits of accuracy as compared with, say, a printing machine or a machine tool, the finishes on locomotive work can be relatively rough.

No greater fallacy exists, and although it is readily admitted that wide clearances are necessary to enable the motion gear of a locomotive to accommodate itself to a non-precision roadbed, the effects of high initial wear, due to lack of finish, are so costly and serious that it might be said that the bearings and wearing surfaces of a modern locomotive, while operating with greater clearances, at the same time demand on the score of economics a higher degree of finish than almost any other piece of mechanism.

Where the load as measured in pounds per square inch is intensive, initial wear can easily develop into permanent and continuous wear if the mechanism is not started out right in the first place.

In these days when a variance of sizing within 0.0002 in. may mean the difference between automatic assembly and hand selection, and this in turn may mean the difference between success and failure to make a profit, and therefore for the firm to remain in business, it logically follows that the problem of accuracy and automatic sizing control has been put up to the machine-tool manufacturers in a very intensive way.

Conclusion

The conclusions which the author wishes to draw from the foregoing notes, data, and records of experiments are the following:

1 It is desirable that our efforts to work to close limits of accuracy be strengthened and improved by taking measurements across basically flat surfaces rather than across surfaces which have a relatively high factor of initial wear.

2 To enable this to be done, standards of finish should be established and these standards accepted in at least a national way.

3 The author submits for consideration a means whereby finishes can be calibrated and easy and accurate duplication of results be obtained.

4 A practical scheme applicable to most organizations carrying on engineering work is discussed and provides a basis on which refinement and special applications can be worked out.

5 A general guide to the requirements as regards the number of machine-shop operations to secure the different qualities of finish is discussed and submitted as representing the best available practice of today.

6 An endeavor has been made to show that there is at present no available shortcut to high-quality work with a low factor of initial wear.

7 The various means whereby surfaces with low initial wear can be obtained are indicated.

8 Finally, the author has shown that standardization of finishes can be attained at low cost and with practically no disruption of present methods; also, that limits of accuracy which control length of useful life are interlocked with the obtaining of a better quality and standardized finish.

Appendix 1—Microscopical Examination of Cross-Section of Surfaces

By DR. C. B. SAWYER,[3] CLEVELAND, OHIO

IN the recording of the shape of design of objects in general it has been found most practical to take a series of cross-sections of the object on different planes through it. The number of such cross-sections necessary is determined by the irregularity of the object and the detail required. Such cross-sections may be made by a draftsman and filed away as blueprints.

When it is desired to obtain a similar record of a surface filled with very fine scratches or surface irregularities, it is necessary, first, mechanically to section such surface and, subsequently, to magnify and photograph its cross-section. Where ground surfaces are concerned, two or three sections are generally adequate to give a very fair idea of the dimensions and nature of irregularities existing thereon. For many purposes a single cross-sectional photomicrograph is adequate. A method for obtaining photomicrographs at magnification up to 1000 diameters of cross-sections of surfaces produced by grinding, honing, polishing, and the like is here described.

So far as is known to the writer, the method was first employed by the famous metallurgist, Walter Rosenhain, in his study of the dimensions of slip planes produced on polished metallic surfaces during deformation of the metal. For a general description of the method it would be difficult to find better words than those of Rosenhain himself,[4] as follows:

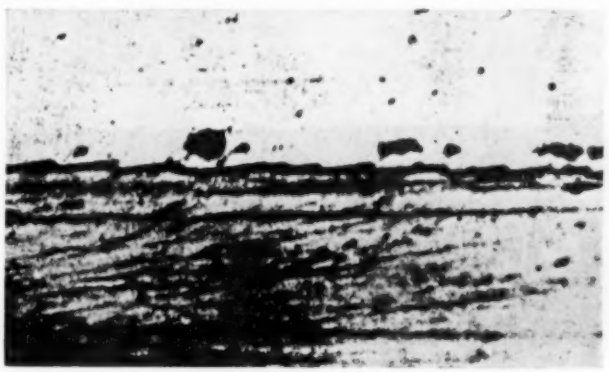


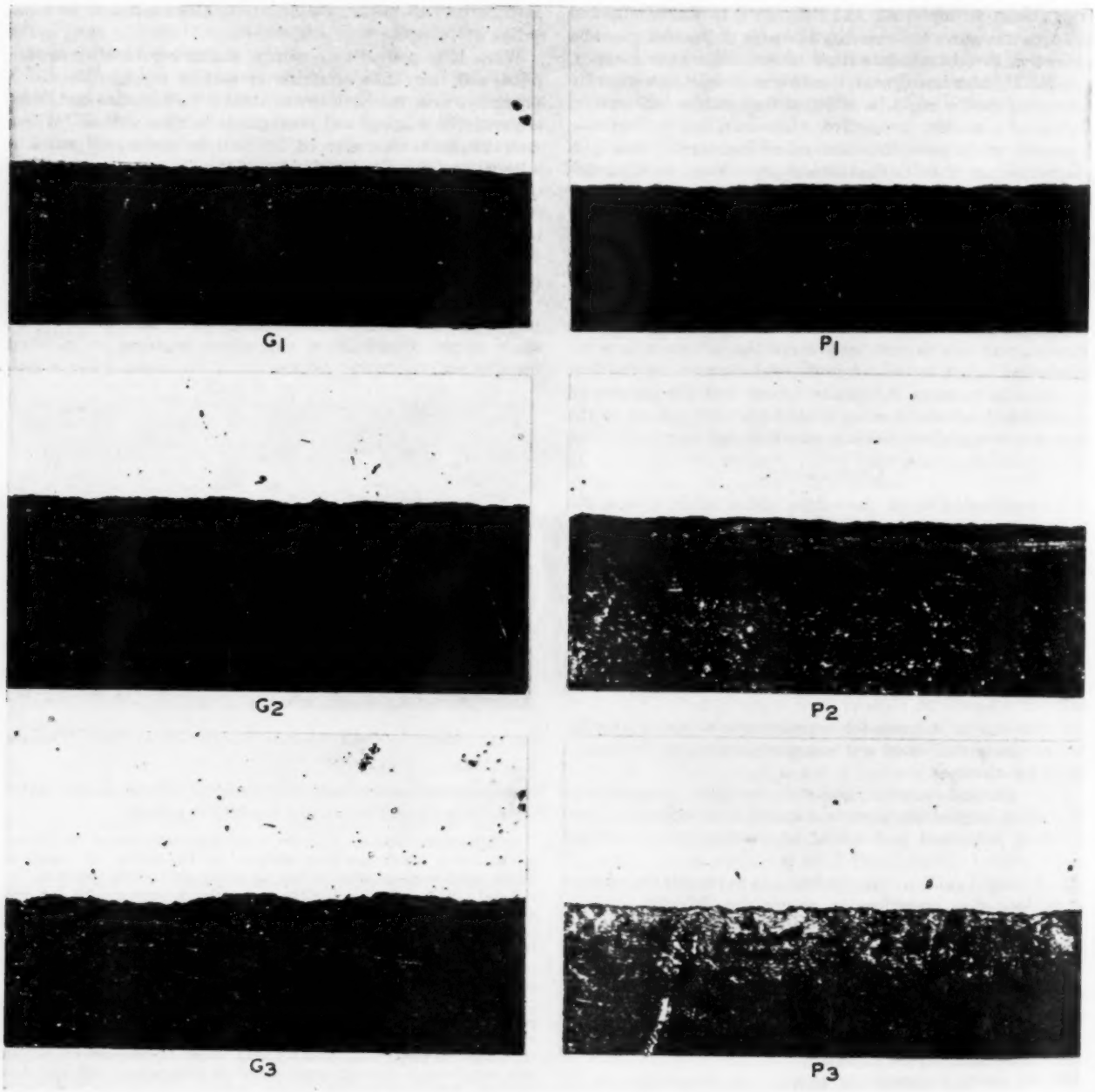
Fig. 12 Magnification Showing Slip Bands as True, Though Minute, Steps

> Still more conclusive, . . . is the demonstration of the nature of slip bands which has been obtained by the author by means of actual cross-sections of a surface on which slip bands had been developed. The difficulty in obtaining such a cross-section lay not alone in the minuteness of the surface features looked for, but also in the fact that, when a specimen is cut and polished, the edges of the surface are always more or less damaged and rounded off. Sharp images at high magnifications, therefore, could not be obtained by simply cutting a previously polished and strained specimen through at right-angles to the original surface and then polishing the cross-section thus obtained. The author overcame this difficulty by first coating the surface on which slip bands had been formed with a thick layer of electrodeposited copper. When subsequently a section was cut through the compound mass—in this case of iron and copper—the boundary on this section, between copper and iron, gave the exact outline of the section of the original surface. When observed at a high magnification, such a section shows the slip bands as true, though minute, steps. The appearance, under a magnification of 1200 diameters, is shown in Fig. 106. [See Fig. 12, which is Fig. 106 of the Rosenhain work.] It is of interest to note that the approximate depth of the minute slip-band steps seen in Fig. 106 is $^1/_{50,000}$ of an inch, which is about the length of a wave of sodium (yellow) light.

For the practical examination of iron or steel surfaces produced by grinding, it is first necessary to obtain a small sample of such surface, measuring perhaps an inch square, and as thick as general conditions may require. Such sample must have been preserved from rusting, corrosion, etc., after grinding. Before copper plating can be commenced, the surface of the sample must be freed from every trace of dirt and grease, without, however, damaging the surface. Cast-iron cylinders which have been subject to oil lubricants or preservatives present

[3] President, The Brush Laboratories Co.

[4] "Introduction to Physical Metallurgy," Rosenhain, 1914, p. 245.



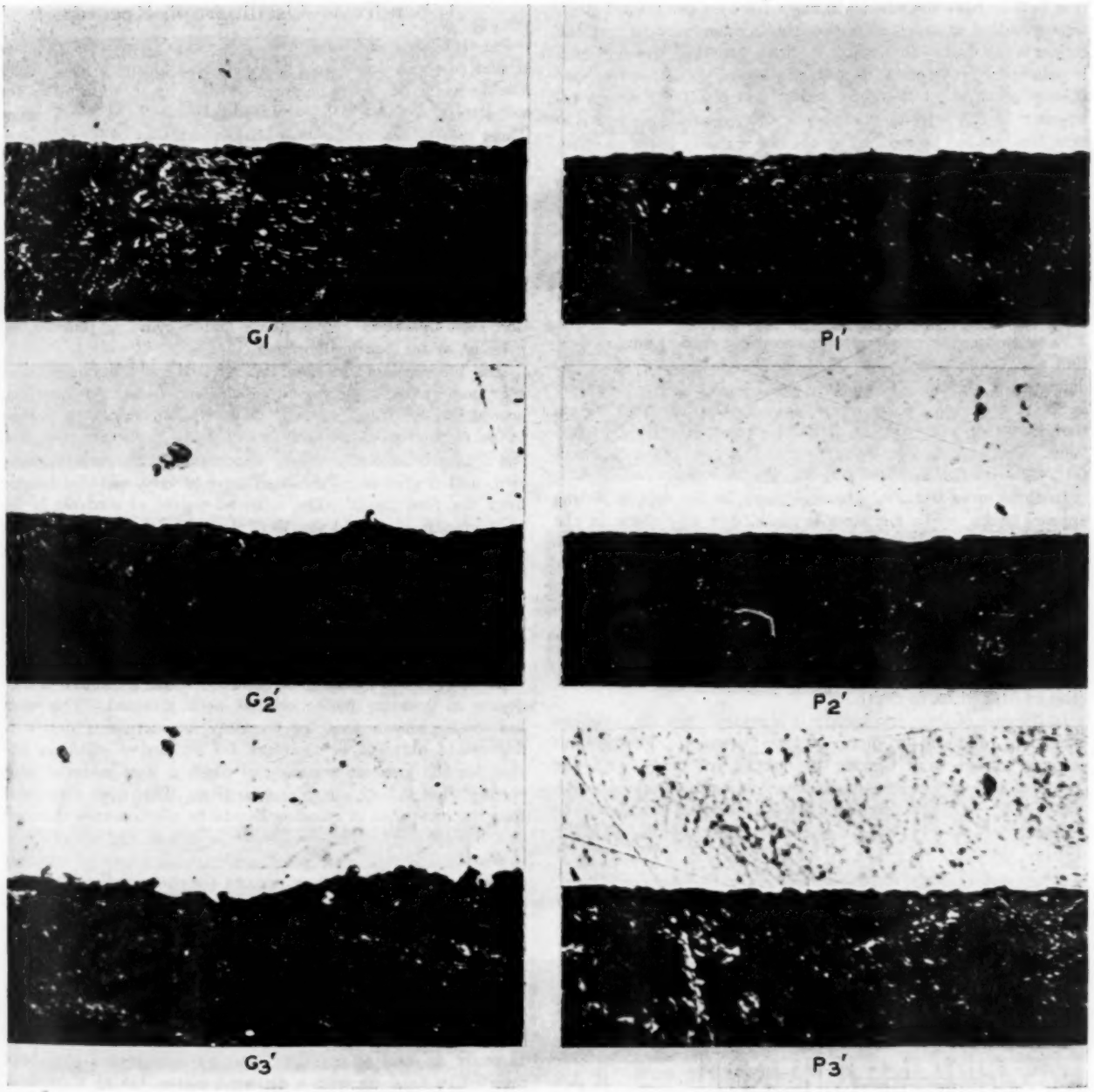
Ground Surfaces Polished Surfaces

FIG. 13 PHOTOMICROGRAPHS OF SURFACE CROSS-SECTIONS × 200

the most difficult cleansing problem, for the oil enters the graphitic pores and is very difficult to wash out. Long soaking in petroleum ether or pure ethyl ether is effective, or the sample may be placed beneath a reflux condenser so that freshly distilled clean ether is constantly dropping on its surface. It may be necessary to continue such treatment for 24 hours. Even when so cleaned the surface may still be covered with a very thin film of oil, and is best dipped in a hot (not boiling) dilute solution of sodium carbonate or sodium hydroxide. Such sample is then transferred quickly to distilled water before the caustic solution can evaporate. After rinsing in distilled water the sample is ready for copper plating. Non-porous materials, such as steel, are much more easily cleaned.

The copper plating is carried out with two baths. Were a steel sample to be dipped at once into the ordinary acid plating bath, "plating by contact" would ensue, and a non-adherent coating of copper would result, with corresponding solution of the steel sample and blurring of the minute outlines of scratches, etc. Consequently, it is necessary to start the plating operation in a cyanide bath in which "plating by contact" will not take place. Unfortunately, it does not seem possible to obtain, with a cyanide bath, a sufficiently thick coating of copper to serve the whole purpose. Therefore, the cyanide plating must be followed by plating in the ordinary acid bath.

A suitable cyanide bath is obtained by dissolving cuprous cyanide in potassium cyanide to form about a 5 per cent solu-



Ground Surfaces Polished Surfaces

FIG. 14 PHOTOMICROGRAPHS OF SURFACE CROSS-SECTIONS × 800

tion, with the addition of about one-half of 1 per cent of sodium carbonate. Plating is carried out at a temperature of about 50 deg. cent., and with the usual copper anode. The current density should be sufficient to produce a considerable evolution of gas. After about an hour, and while the surface is still smooth and uniformly salmon-colored, the sample is lifted from the copper cyanide and quickly washed in distilled water, before the adhering solution has time to evaporate. It is then at once transferred to the acid plating bath, where the plating is continued without danger to the surface, because of the protective coating of copper laid down in the cyanide bath. A suitable composition for the acid bath is that used for copper coulombmeters, as follows: 1000 grams of water, 150 grams copper sulphate, 50 grams concentrated sulphuric acid, and 50 grams alcohol.

Plating takes place at room temperature, and is greatly expedited if the solution is strongly and continuously stirred, as by an electric motor. Deposition continues until the thickness of copper overlaying the surface to be examined is about $^1/_8$ in. The time required varies from two days to a week, depending on the degree of stirring, which in turn controls the maximum current density at which adherent copper is deposited.

When the copper coating is of adequate thickness, the sample is removed, washed, and cut with a hacksaw along the plane desired. As the copper coating is not extremely adherent,

it is best to have the saw teeth move from copper toward steel, thus avoiding as much as is possible any tendency to tear the copper away from the surface. After so cutting, the specimen is polished by the usual metallographic methods. As copper is very much softer than steel, there will result at best an inequality in the level of the copper surface and the steel surface, but this can be reduced by the use of emery paper as much as possible in the polishing operations.

Results obtainable with this method are illustrated by the accompanying photomicrographs made several years ago in studying the bearing surfaces on two crankshafts. The surfaces on the one crankshaft were finished by grinding; on the other by "polishing." Three samples from the bearing surfaces of each crankshaft were copper-plated and prepared as described.

Each group of three samples was photographed at a magnification of 200 diameters (Fig. 13) and at a magnification of 800 diameters (Fig. 14). The first group of photomicrographs are at 200 diameters. They comprise samples G1, G2, and G3 from the ground surfaces and P1, P2, and P3 from the polished surfaces. The second group at 800 diameters is similarly designated, G1′, G2′, and G3′ and P1′, P2′, and P3′.

In these cross-sectional photomicrographs the copper plating appears white, while the steel is black, due to etching in the usual nital to accentuate the contrast between the steel and the copper.

As before stated, there is a slight difference in level between copper and steel, which results in a blurred appearance of the steel structure, as the camera was necessarily focused on the copper. Nevertheless, all boundaries, even at 800 diameters, are sharply defined and give the exact shape of scratches as little as 0.00001 in. in depth.

In the case of these crankshafts it is evident that the polishing operation produced a more uniform product. The ground surfaces varied from bearing to bearing, probably with the truing of the wheel. Sample G3 was probably produced with a freshly trued wheel, which became somewhat glazed by the time it reached G1. Such a partially glazed or filled wheel polishes somewhat as it grinds, and the surface produced by it approaches a polished surface.

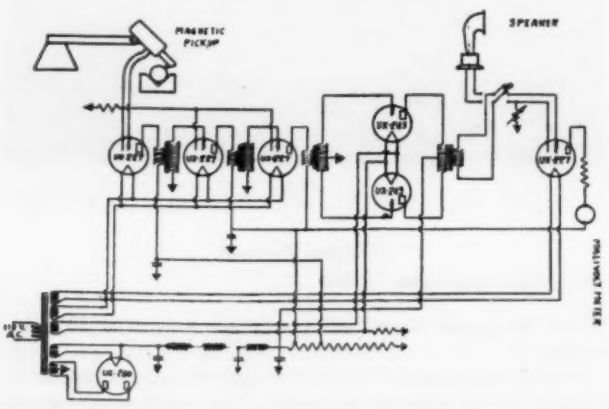


FIG. 15 CIRCUIT USED IN MAKING VISUAL AND SOUND RECORDS

Appendix 2—Circuit Diagram of Finish Comparator

The diagram (Fig. 15) represents the circuit used when making the visual and sound records. A further refinement and a permanent record can be obtained by connecting an oscillograph instrument to this circuit.

Appendix 3—Oscillograph Records

Fig. 16 shows oscillograph records made from a set of plugs similar to those shown in Fig. 7. It was ultimately found advisable to include a set of plugs of cast iron in addition to the soft-steel, case-hardened steel, and oil-hardened steel plugs. These records clearly show the varying equality of finish in each material, and are of great value in those cases where it is considered desirable for a record of the quality of surface to be kept for future reference.

Discussion

CHRISTOPHER H. BIERBAUM.[5] This paper is certainly timely and interesting, since further improvements of machine tools and their operation necessitate a higher grinding practice, especially so for bearing surfaces.

The photographic method for determining the roughness of a ground surface is not at all conclusive; it leads to many inaccuracies and incongruities, for the reason that it is subject to so many variations and interdependent factors. Of these the illumination is of primary importance; its intensity, direction, and degree of diffusion all have a very material bearing upon the final result. The inherent degree of contrast in the photographic plate and also that of the printing paper are very distinct factors, as well as the personal equation and chemicals used in their developing; all affect the final result in a marked degree. The procedure for determining the smoothness of ground surfaces presented in this paper overcomes all of the photographic difficulties and seems altogether practical and accurate.

In the present grinding of metallographic specimens a high degree of grinding perfection has been attained. This work has clearly shown that for economy an increasing degree of fineness of abrasive is necessary for successive grindings and that for the greatest accuracy of finish a final material must be used that is both sharp and cutting. This work also shows that the directions of grinding should be continuously changed; this, however, is not readily possible in the grinding of cylindrical surfaces. The value of more accurate and perfect grinding cannot be overestimated; it reduces the time for the running-in of bearings; it reduces the aggregate amount of wearing down that is necessary during such running-in, thereby very materially reducing the liability of introducing inaccuracies in the bearing surfaces during the period of running-in.

Many manufacturers have found that the highest degree of accuracy of finish in their bearings is an economic investment. This is especially true for bearing alloys that cannot be finished as easily as steel by grinding, and for that reason they have resorted to finishing with a diamond cutter, taking a depth of final cut of 0.001 in. or less, at a very high speed, and thereby leaving all of the bearing crystals upon the surface in an undisturbed condition,[6] leaving these hardest crystals flush with the finished surfaces, that is, in a state that most nearly approaches a run-in surface. By such procedure they have found that bearings can be produced which require very little or no preliminary running-in. This, together with the further improvement of the grinding of steel journals, should have a very marked effect upon the success of sliding-surface bearings.

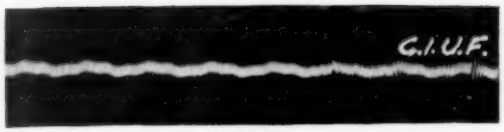
JOHN GAILLARD.[7] The paper makes it clear that a high quality of surface finish should not be confined to fits requiring

[5] Vice-President and Consulting Engineer, Lumen Bearing Company, Buffalo, N. Y. Life Mem. A.S.M.E.

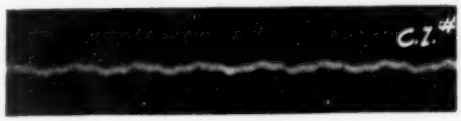
[6] A.S.M.E. Trans., Dec., 1919, pp. 799–802.

[7] Mechanical Engineer, American Standards Association, New York, N. Y. Mem. A.S.M.E.

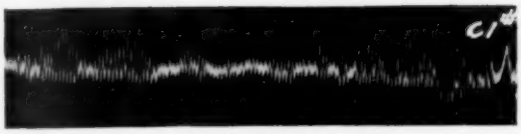
CAST IRON



Ultra-Fine Finish



No. 1 Finish

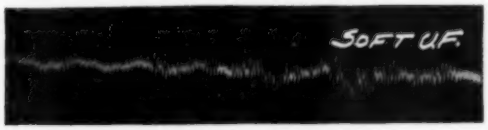


No. 2 Finish



No. 3 Finish

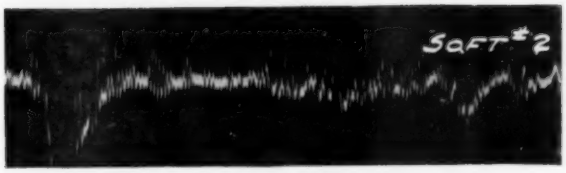
SOFT STEEL



Ultra-Fine Finish



No. 1 Finish

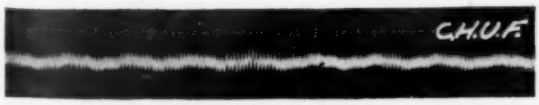


No. 2 Finish

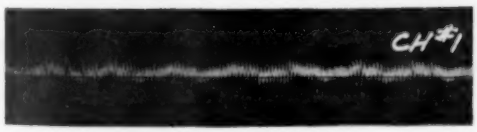


No. 3 Finish

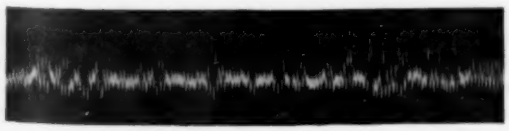
CASE-HARDENED STEEL



Ultra-Fine Finish



No. 1 Finish

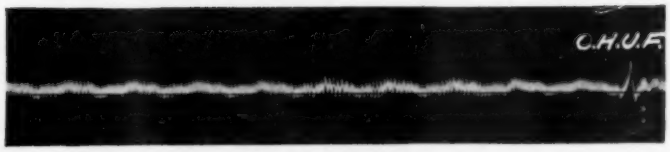


No. 2 Finish

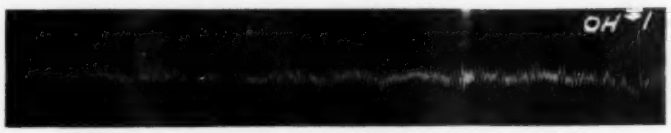


No. 3 Finish

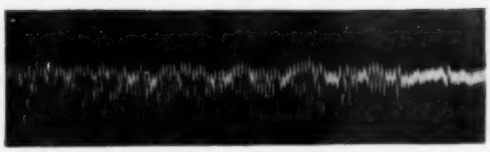
OIL-HARDENED STEEL



Ultra-Fine Finish



No. 1 Finish



No. 2 Finish



No. 3 Finish

FIG. 16 OSCILLOGRAPH RECORDS, FROM A SET OF PLUGS SIMILAR TO THOSE SHOWN IN FIG. 7

small allowances and close tolerances, but that it should also be applied to less exacting fits in cases where strong initial wear of the mating parts is to be avoided. This view accentuates the fundamental difference, too often disregarded, between the concepts "quality of fit" and "quality of surface." The quality of a fit depends on the ratio between the maximum and minimum permissible clearance or interference, or in other words, on the permissible variation in looseness or tightness. This again depends directly on the sum of the tolerances on the mating parts. The quality of a surface depends among other things on the maximum variation of its points from a true cylindrical plane. One could say that for a specific quality of finish all points of the surface concerned must lie within a definite "finish tolerance." For a surface of high quality this "finish tolerance" is of a smaller order of magnitude than the tolerances on the size of the parts. Dr. Sawyer's micrographs with a magnification of 800 times illustrate this. Although, in principle, quality of fit and quality of surface are different concepts, there is a practical relationship between the two. Close tolerances naturally demand a good quality of finish. The opposite is not necessarily true, however. Fits with large clearances and tolerances may be established without the parts receiving a high-grade finish. However, as the author points out and illustrates by practical examples, it often will be desirable, if not necessary, to give the parts a high-grade finish anyway in order to prevent rapid initial wear.

The paper shows that the art of producing good fits is progressing consistently along lines leading to greater refinement. Without being able to forego high speed of output, mass production has arrived at an accuracy of sizing parts and at a quality of surface finish that would have been impracticable in the workshop a few years ago even for serial production. The indications are therefore that the time is near when specification of a definite quality of finish by the designer will be regarded as just as necessary as the specification of tolerances on the sizes of the parts. It will not be sufficient to state, for example, that a part should be ground, but it also will be necessary to mention by class the quality of surface desired, and this again raises the problem of classifying surface finishes. This requirement is analogous with the necessity of standardizing classes of fit based on definite values of allowances and tolerances. So long as the nature of a fit is merely expressed by such terms as "snug," "wringing," "push," or "tight," no fit can be definitely specified. These terms have different meanings in different shops, and even with different individuals. If they have the same meaning, their interpretation is still dependent on the personal factor. A fit defined as a "wringing" fit by a husky fellow may feel like a "tight" fit to a man with less manual "torque."

The first step taken to eliminate the human factor in determining the class of a fit was to use a "standard" plug and ring giving practically a "metal to metal" fit together. That is, a physical example of a fit was created which could be imitated (or at least approximated) or varied from in a definite direction, as the case may be. The author's first step toward standardizing the class of a surface has been a similar one. Sets of plugs finished with different qualities of surface were established to serve as visual guides. However, in addition to this, the author struck upon the very ingenious idea of translating the frequency of unevenness of a surface per unit of length into frequency of sound waves per unit of time—that is, into the pitch of tone. This method has the great advantages of being easily applicable and of giving an immediate answer to the question what the quality of a surface is. The author must be congratulated on the simplicity and originality of this development. In addition to establishing a basis of measurement with consequent possibility of standard classification of surface finishes, it has the charm of containing a promise that in the future the surfaces of component parts may literally be tuned together for greater harmony in the assembled product.

As stated in the paper, the need for a basis of classification has been growing in recent years. Investigation of the nature of different surfaces was a basic requirement in trying to solve the problem. Different methods for doing this have been worked out. One of these, developed by Professor Berndt, of the Engineering Academy, Dresden, Germany, whom the writer visited this summer, may be of interest here. It complements Mr. Harrison's method in a remarkable manner and shows again how the existence of a problem may lead to its solution by independent experts at almost the same time. In the present case the solutions are different, but they show a similar basic feature.

Professor Berndt has developed a method of recording the minute irregularities of the profile of a finished surface. He lets a fine needle (so far he has used a gramophone needle) travel over the surface at such a slow rate of speed that the point follows all of its ups and downs. To do this, the part whose surface is being investigated is moved and the device carrying the needle is stationary. The needle is attached to a spring on which is mounted a small mirror. A beam of light is thrown on the mirror through a very fine slot and is reflected on a slowly moving film of sensitive paper. The beam thus traces a latent curve on this strip representing the magnified outline of the surface investigated. The strip is developed and oiled to become printable. The graphs thus obtained which were shown to the writer represented the outline of the surfaces investigated to the scale of either 400 to 1 or 1000 to 1.

One of Professor Berndt's students has used this method in the investigation of a large number of surfaces finished by different methods of machining, such as turning, grinding, lapping, broaching, and honing. Also the surfaces of parts tapped with cut taps and with ground taps were investigated. The study formed part of a "doctor of engineering" dissertation, and no details are as yet available for publication. The writer hopes to be able to report about the details in the near future when they are released.

Professor Berndt said that the needle method had given very good results, but that with increasing precision in surface finishing a point sharper than that of a gramophone needle was now required. Before developing this method, he had used two others. One of these, invented by a British expert, consists in making gelatin casts of the finished surfaces. These are cut to very thin slices, which are viewed under the microscope. This method appeared to be unreliable, as the casts warped when and after being removed and therefore did not represent true negatives of the surfaces. The interesting method of making the outline of the surface visible in cross-section described in Dr. Sawyer's paper, appended to Mr. Harrison's, marks great progress in this respect. The other method applied by Professor Berndt consists in measuring the maximum angle at which the surface still gives a fairly sharp image of a series of geometrical figures placed in a comparator, the surface under investigation deflecting the rays of monochromatic light sent out by the figures. Past a certain angle the image becomes decidedly blurred. This method has been checked by Professor Berndt with the needle method, with satisfactory results. Of course, the reflection method does not give details of the surface as the needle method does. In this connection it is interesting to note that Mr. Harrison refers to the occurrence in practice of "highly polished and deeply scratched" work possessing rather smooth surfaces located between pronounced ridges, which work gives a higher rating when subjected to the reflection test than it really merits from the viewpoint of quality of surface.

The practical solutions of the problem independently arrived at by Mr. Harrison and Professor Berndt are different and complementary. Mr. Harrison's method makes it possible to get an immediate but general answer regarding the quality of a surface. Practically, such an answer will probably be quite satisfactory in the large majority of cases. The application of Professor Berndt's method requires considerably more time and effort, but it gives a detailed and permanent record of the surface investigated. There will be cases in which one of the two methods should preferably be applied, and other cases where both should be used for the sake of completeness. Viewing in general the art of finishing surfaces and classifying their quality, both developments constitute a genuine contribution to the art by laying a basis for the substitution of judgment by measurement and consequent possibility of standardizing the practice.

The writer has followed the developments in this field with the deepest interest, both as a mechanical engineer and as a staff member of the American Standards Association (A.S.A.). When the time is ripe, in the opinion of the experts in this field to discuss the matter with a view to the possibility of establishing a national standard method of classifying surface finishes, the A.S.A. will be very glad to lend its assistance in organizing the work. For the present the writer wishes to congratulate Mr. Harrison on the great value to industry of his findings and on the exceedingly lucid manner in which his presentation has been made.

Sanford A. Moss.[8] Mr. Harrison properly distinguishes between truth of a surface and finish of a surface. A surface may have a low grade of finish and yet the high spots may be on a surface which is flat or true within a few tenths of a thousandth. On the other hand, the surface may have a high-grade finish according to Mr. Harrison's standards, and yet it may be out of round or out of truth by 0.0005 in. or even 0.001 in. Is it not possible that some of the benefits due to wearing-in are due to correction of errors of truth rather than errors of finish? In case of perfect film lubrication, there is supposed to be a layer of oil separating the surfaces. If the surfaces are perfectly true, it would seem that the finish is not a major item. There is of course another class of lubrication where there is not a perfect film, but merely a sliding of two greasy surfaces. For this sort of thing the finish is no doubt highly important. Of course, a first step toward finding out the cases where the finish is the major matter and the cases where truth rather than finish is the major matter, is the standardization of quality of finish, just as Mr. Harrison proposes. It is to be expected, therefore, that the pioneer work which Mr. Harrison has done will be followed up to a conclusion.

G. A. Bouvier.[9] The electrical method described in the paper, of calibrating the character of a surface, is not only new and novel, but apparently promises greater opportunity for general use in industry than methods suggested heretofore. While the author has stated that his work can only be regarded as preliminary, it is hoped that other investigators will take part in further research work. No doubt Mr. Harrison himself, if he has not already done so, will discover further advantages of the method.

A number of limiting factors have occurred to the writer which affect the method and which should be evaluated for a complete understanding of the subject.

Fundamentally, the proposed scheme consists of the generation of electrical energy by the movement of a phonograph needle over the surface whose character we are interested in. The phonograph needle, in this instance, is attached to an ordinary magnetic pick-up as used on phonograph records; in other words, a type of electrical generator. The movement of the needle over a hill-and-valley surface at a definite rate will of course result in the output of an electric current having a certain frequency relation to the number of hills and valleys over which the needle moves. Obviously, the power or energy generated will also have a definite relation to the character of the surface, because the hills and valleys not only differ in spacing but also in depth. The frequency would show up on the loud speaker as a complicated pitch or tone, and the energy as volume. Obviously, such a complex audible sound made up of overtone and discords can give to the investigator only a very rough indication of the real character of the surface. The writer would like to learn the results of further investigations of this method made with a recording oscillograph. This would supplement, in greater detail, the millivoltmeter readings which average the electrical output and hence give an index of the average roughness.

In ordinary ground work, as illustrated in Fig. 2, the scratches all run in the same direction. These scratches constitute the major surface imperfections, so that drawing the needle point at right-angles to them, as done by Mr. Harrison, would give the maximum energy output and the best index of surface roughness. On certain types of surfaces, however, such as the lapped surface shown in Fig. 5, the surface scratches possess no definite direction. On a surface of this type it appears that millivoltmeter readings should be taken for needle movements in two directions at right-angles to each other, and the results averaged.

It is apparent that such factors as the type of point, the weight on it, and the electrical constant of the amplifier will affect the results obtained. These factors must be standardized to make the device of general utility to the industry. The development of more knowledge concerning the limitations of this new method and the establishment of standards certainly will result in its ordinary and regular use in the shop for controlling the character of the finish on a definite standard basis instead of by the visual determinations now attempted. This method is especially suitable for measuring finishes so fine that methods employing the reflection of light cannot be applied. The writer hopes that Mr. Harrison will continue without interruption the development of this idea.

Luther D. Burlingame.[10] The presentation of this valuable paper emphasizes the fact that there has in recent years been a transference of skill from the workman to the machine so that precision work and that having a high degree of finish and accuracy, emphasized by Mr. Harrison, can be produced commercially. In the early days of grinding, the producing of such results depended on the skill of the workman.

In investigating historical matters relative to the origin and early use of grinding processes, it has come to attention that in those days grinding was done on an engine lathe with a grinding wheel mounted on the carriage, and personal skill was required in order to produce the desired grade of work; one of the workmen being so expert in this that he commanded a wage which placed him with the proprietors of the business, so that he came to the shop, as did they, in a tall silk hat. Skill in those days depended on knowing where to stand on the floor and where to tap the machine in order to secure satisfactory results.

This led Joseph R. Brown to invent the universal grinding

[8] Mechanical Engineer, Thomson Research Laboratory, General Electric Company, West Lynn, Mass. Mem. A.S.M.E.

[9] Development Engineer, Hawthorne Works, Western Electric Co.

[10] Legal Technician, Brown & Sharpe Mfg. Co., Providence, R. I. Mem. A.S.M.E.

machine, which he brought out in 1875 and which was first shown at the Centennial Exposition in 1876.

During the period since that time continued improvements have been made in the grinding art and in the machine connected therewith, so that today results can be obtained such as those pointed out by Mr. Harrison in his comprehensive paper.

Author's Closure

By Dr. C. B. Sawyer.[11] Mr. Gaillard's remarks concerning the appearance of a technique for studying production results at such time as the demand for better control arises in the producing industry itself seems well illustrated in the present case. The work on microscopical examination of cross-section of surfaces here described was completed some five or six years ago, and has lain dormant from that time forward, until brought to the attention of Mr. Harrison. It now appears that the demand for better production control of surfaces has resulted not only in the appearance of the two methods described in this paper, but also of Professor Berndt's, and doubtless of others to be heard from later.

With reference to the time required for copper plating surfaces, two days is probably the minimum so far attained, but new methods for copper plating are accelerating the rate of deposit, and doubtless such time can be further very considerably reduced. Probably the time required for copper plating and examination of the surfaces can ultimately be reduced to an amount at least comparable with that required for an ordinary metallographic examination of steel structure.

From the photomicrographs shown, it is apparent that the method of copper plating surfaces gives details in contours attainable by no other method so far suggested. It is therefore at once evident that the copper-plating method should be very useful in correlating or calibrating the indications obtained by other methods which may be much quicker.

Further, it is evident that with the lessening of time required for preparing copper-plated cross-sections, the method becomes more and more useful for routine control of machine-shop production. It approaches the type of control furnished by ordinary metallographic examinations. Such type of examination may be applied to specimens selected from a run of any specified number of units.

Mr. Harrison has asked me to make a few observations on the acoustic method described by him, and in this connection it is well to consider the characteristics of the human ear.

In the perception of quality of tone, the human ear is truly remarkable. It possesses the ability to resolve a complex tone into its various components, and for a true qualitative match of electrical impulses derived from ground surfaces of repetitive characteristics by Mr. Harrison's phonograph pick-up and amplifier, the ear is probably irreplaceable by any instrument such as an ammeter, or even recording oscillograph, because this latter requires time for the development of images. Unfortunately, on the other hand, the human ear is not especially sensitive to changes in tone intensity. At fair volume proceeding from a radio speaker, the ear can, under most favorable conditions, detect a minimum of a 10 per cent difference in intensity, but the attainment of this minimum requires instantaneous switching from one tone to another at least once a second, and preferably four or five times a second. Moreover, there should be a low level of extraneous noise. Such low level of sound would be difficult to obtain in an ordinary machine shop, and the requirement for quick comparison of tones predicates a more elaborate apparatus than that described by Mr. Harrison. Probably under most conditions of use with apparatus like Mr. Harrison's, the ear would do well to detect a 50 per cent difference in sound intensity, this arising in turn from like differences in the ground surfaces. In a sense, this is advantageous, as any real difference detected by the ear will certainly proceed from a very real difference in the surfaces.

With proper precautions, such as a soundproof room, two phonograph pick-ups adapted to be drawn at identical speeds over the samples under comparison, and mechanical motorized switching devices, Mr. Harrison's method could be refined to give very quick and accurate comparison of surfaces. However, for the great differences prevailing in ground surfaces at present, the method just as described by him is very useful, and it can be borne in mind that, when conditions warrant, refinements such as indicated can be worked out.

The question has been raised as to the effect of harmonics introduced by amplifier and speaker. In my opinion, the perfections lately incorporated in phonograph pick-ups, amplifiers, and loud speakers should eliminate cause for worry on this point when the application is limited to simultaneous comparison of samples.

For those further interested in the effect of sound on the human ear, reference can be made to "Speech and Hearing," by Harvey Fletcher, Ph.D., published by D. Van Nostrand Company.

By Mr. Harrison. The author readily agrees with Mr. Bierbaum regarding the limitations which apply to the photographic method of recording imperfections in machined surfaces.

He was particularly pleased to see the reference to the effect that, in the preparation of metallographic specimens, the material used for the final finishing operation must be both sharp and cutting. This is additional confirmation of the author's invariable claim that finish should never be obtained other than with a sharp and precise tool, as otherwise the surface produced has a false finish and is untrue either as a cylinder or a plane.

There are of course many applications where precision is not of prime importance, but in those cases where precision requirements are paramount, there can be no compromise, i.e., the final operation, either grinding or polishing, must be performed with a tool which is just as precise as the finest limit required on the work, and this tool must be held with maximum rigidity and guided with an accuracy which will permit the desired degree of tolerance to be maintained.

Mr. Bierbaum's remarks under the heading of artificially matured bearing surfaces also are of special interest, inasmuch as, in the author's experience, a considerable number of very accurate and high-finished spindles have been ruined in a short time by inserting them in bearings which have been scraped in such a way that the actual contact points between spindle and bearing have been the hard crystals in the bronze.

This has resulted in high unit pressure, disruption of the oil film, and all the troubles which go with excessive heating and carbonization of the lubricating oil.

The diamond boring tool seems to offer the solution, as it will perform the same function as a properly graded grinding wheel, and instead of exposing the hard crystals, will leave them flush with the surface where they properly belong and where they can exert their best influence in supporting the spindle.

Mr. Gaillard's remarks regarding the experimental work carried out by Professor Berndt are also extremely interesting, and it is a remarkable coincidence that Professor Berndt should be engaged in an endeavor to find a solution to this problem simultaneously with ourselves.

It has been the author's endeavor to devise an instrument capable of practical workshop application, the principle on which this instrument is designed being capable of expansion to take care of the greater refinements which are desirable and permissible in a laboratory investigation.

[11] President, Brush Laboratories Company, Cleveland, Ohio.

The main virtue which might be claimed for the author's scratch test is the ability of the instrument, both visually and orally, to detect and record the spurious finish, i.e., the one with high luster but serious mechanical imperfections.

It is the author's hope that the American Standards Association will continue to give practical expression to its interest in this subject to the point where national standards can be set up.

In reply to Mr. Moss's contribution to the discussion, the author would state that he considers it a major requirement that any piece of work should have all practical mechanical imperfections removed before there is any discussion with regard to its finish. The question of finish is raised in conjunction with durability of accuracy.

In other words, it has been repeatedly proved that extremely fine limits of accuracy, with their proportionately increased cost, are wasted on articles produced with a coarse finish, this being due to the high initial wear inseparable from a coarse finish.

The author readily agrees with Mr. Bouvier that the method which is put forward is not free from technical imperfections. However, as pointed out by Dr. Sawyer in his joint closure, the method of calibration has proved sufficiently accurate to enable a control to be set up which is generally commercially acceptable.

In line with Mr. Burlingame's comments, the development of the grinding wheel has been concurrent with that of the grinding machine. However, despite the high standard of attainment which has been reached by both elements, so far no definite standard of performance has been set up for either.

The author appreciates and acknowledges the enormous value of the pioneer work accomplished by Mr. Burlingame's company.

The author gratefully acknowledges the encouragement contained in these discussions, and in reply to the hopes that have been expressed that experiments will be continued, advises that

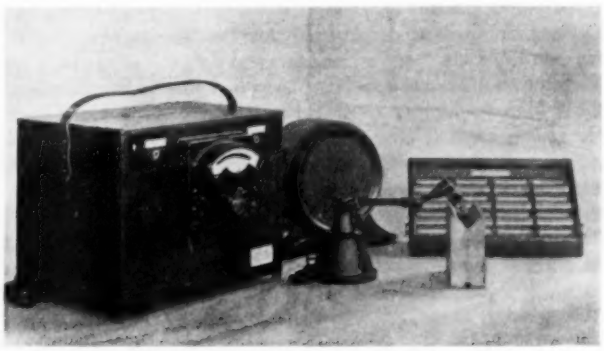


FIG. 17 NEW CALIBRATION INSTRUMENT

work on both the instrument itself and the technique of finish calibration has gone forward to the point where an instrument has been produced, as shown in the accompanying illustration, Fig. 17.

The instrument shown at the left of this illustration is portable and durable, and it is anticipated that it will shortly be made commercially available to all concerns having an interest in this subject.

MSP-53-13

Refinements in Finishing Cylindrical Bores

By J. W. HINDES[1] and JAMES G. YOUNG,[2] DETROIT, MICH.

THE limitations of boring, reaming, burnishing, and broaching as finishing operations on cylindrical bores have led to the general use of abrasives for such work. Improvements in manufactured abrasives and study of operating data, together with improved methods of application, have made grinding a precision process. The cost of operation and the difficulty of bearing maintenance of planetary-type grinders have brought about the use of the expanding grinder or so-called "honing" method. The latter method has the advantage of speed, improved accuracy and finish, and low cost. It also makes possible the precision grinding of long tubes, gun-barrel rifling, and bores with ports, keyways, and relieved portions.

THE problem of producing cylindrical bores held to ever closer limits of accuracy and finish has given industry no little concern. The bore produced must not only be round, straight, and smooth, but also of even texture with full metal surface, as nearly as possible identical with a run-in bore. A thorough discussion of the finishing of cylindrical bores by means of the expanding grinder or honing method must necessarily make reference to other practical methods of finishing and to the limitations of these methods.

Boring, reaming, burnishing, broaching, grinding, lapping, and the so-called honing method have been used with varying degrees of success.

Limitations of Boring, Reaming, Burnishing, and Broaching

Boring, although indispensable for the rapid removal of large amounts of stock, has been practically discarded as a finishing operation where close limits are required. Although the use of tungsten-carbide cutting tools has made it possible to produce a much better finish on many boring operations than was formerly possible, the nature of the operation itself prevents it from becoming a satisfactory finishing method.

The surface generated by boring is always a helical groove, varying in depth according to the overlap of the tool, the condition of the cutting edge, and the evenness of the metal. Such a surface has a series of ridges or crests, commonly termed "fuzz," which shear off or wear down quickly, causing excessive initial wear.

The accuracy of the bore as to straightness and roundness is also dependent upon the accuracy of the piloting, the evenness of the metal, the condition of the tool, and the skill of the operator.

Reaming is subject to a less degree to many of the limitations of boring. Out-of-roundness and taper are quite common, and considerable difficulty is found in producing cylinders of uniform size.

Burnishing or rolling has been used quite widely on metals which lend themselves to cold working, and on these it gives a smooth, hard surface.

When used on cast iron the fuzz is merely pressed into the wall and does not become an integral part of the surface. Bores are frequently out of round, especially when not having uniform external support over the entire length or circumference. Walls of varying density are necessarily produced as the metal is not displaced, the irregularities being pressed into the walls.

Broaching has found many applications where walls are heavy and uniformly supported. When properly fixtured, accuracy is not dependent upon the skill of the operator. As with other cutting tools such as in boring and reaming, the uniformity as to size and finish is influenced by the wear of the tool.

Broaching has not proved satisfactory for finishing bores with thin walls, or those having varying support such as cylinders with ports, cut-away sections, or flanges. The metal being pushed out under high pressure, rigidity of material is essential.

The limitations of the above methods of finishing cylindrical bores have led to the constantly increasing use of abrasives for that purpose.

Grinding Practice

Engineers and others who have followed manufacturing methods over the past 25 years will have noticed the ever-widening scope of grinding. Confined at first almost to the cutlery trades, and to the comparatively few parts which required precision, its machining value was not understood, nor its application to mass production appreciated. This of course was very largely due to the fact that in the early days of grinding, artificial abrasives were unknown, and quarried stone and emery so limited the choice that to obtain a really suitable abrading agent for every operation was impossible.

The wide range of carefully graded wheels and stones now available, and the data obtainable on speeds, feeds, etc., have changed practically all grinding operations from speculation to certainty. In addition to this the value of grinding as an agent for removing stock has been realized, and as the need for refinements and closer limits has been developed and demanded, machines have been evolved to meet the growth and perform an increasing range of operations.

Maximum speed for production and accurate working to limits is now possible with suitable grinding equipment, so that grinding is an essential factor in the manufacture of standardized duplicate parts. In the enormous field of mechanical industry it is being daily demonstrated that greater speed, closer accuracy, and reductions in operating costs can be obtained by introducing the most up-to-date grinding equipment, and this is possible and noticeable in every phase of manufacture.

The manufacture of abrasives and grinding wheels, and the several factors affecting their selection, would require a paper in themselves, but these subjects must be briefly dealt with in order to bring out the applications of grinding touched upon in the present paper.

Grinding wheels, distinct from natural grit stones, are usually manufactured from two types of abrasives, an aluminous abrasive, with bauxite as a base, and a silicon abrasive, with silica sand and coke as a base.

The bonding of abrasives is done by varying processes, and at least 75 per cent of the manufactured wheels and sticks are vitrified. It was found necessary to vitrify abrasives in order to remove metal quickly, and the amount of metal removed per cubic inch of wheel wear is very high due to porosity and strength

---

[1] Sales Manager, Hutto Engineering Company.
[2] General Manager, Hutto Engineering Company.

Contributed by the Machine Shop Practice Division and presented at the Semi-Annual Meeting, Detroit, Mich., June 9 to 12, 1931, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

of the bond. Vitrified wheels are not affected by water, acids, or changes of temperature.

Aluminous-abrasive grains are much tougher than silicon carbide grains, do not break apart easily, and withstand greater stress and are used on materials of high tensile strength.

In describing grinding wheels they are sometimes compared to milling cutters having a greater number of teeth or cutting points, and on a normal-size grinding wheel, millions of grains act each minute.

It is customary for wheels to cut freer after wearing down a little; this is explained by the reduction in the arc of contact, which results in the grains leaving the bond more quickly as the strain increases—for by maintaining the same surface feed, each grain has to do more work in a given time and that wears down more quickly.

Correct wheel selection is all-important, an error in one or two grades in density or bond having an enormous bearing on production, the latter being sometimes doubled by the difference of a grade or so in hardness of bond.

The advance of precision grinding machines in recent years is one of the important developments in present-day production methods, for in order to produce the greatest number of parts the exact limits for size and finish must be set, and modern production grinding has to conform to given rules and standards of precision.

The most common form of precision grinding is the grinding of cylindrical parts. In studying cylindrical grinding it is necessary to remember the close connection of previous operations. Often the grinding operation will correct imperfections, but careless use of this rule leads to excessive wheel costs, and where materials are hardened there is danger of surface cracks and local heating, resulting in inferior products and often heavy increases in scrap percentages.

There are a number of high-grade cylindrical grinding machines, of which several features worthy of mention to production engineers are: massive wheel spindles, ample speed changes, table and work speeds controlled independently, and rapid power traversing, operating in either direction. Some machines are fitted with an oscillating table motion that preserves the wheel face for form grinding. Other machines operate on a different principle, the wheels traversing the work, which latter rotates in a fixed position. As to the relative merits of traveling work tables or traveling wheel heads, there is room for discussion.

The introduction in quite recent years of hydraulic drives for grinding machines has also resulted in more convenient control mechanisms for all classes of grinding machines.

Some useful points to note for cylindrical grinding are: Be sure the speed is right; tap new wheels to be sure of soundness; never force wheels on to spindles, and where lead bushes are used, these should be a sliding fit on spindles, and thin rubber or thick paper washers should be fitted between the wheel faces and flanges; always watch the speed as the wheels wear down; and it is better economy to change to the right grit and grade of wheels, as the time required for changing is very soon made up on quantity production.

Automatic feeding insures economy in wheel wear, allowing regular advancement of the wheel and thereby permitting it to act under normal effective conditions. Highly skilled operators, however, exercise great care in applying cuts by hand, and are successful in preserving wheel wear.

The strengths of abrasive grains and bonds are effective only to the limit of their ability to withstand the shock forced upon them in grinding. By overloading or forcing, the grains become loose before actually serving their real purpose as a cutting medium, and correctness of the wheel face is lost.

Any wheel that is correct as regards its speed and that of the work piece being ground, will stand a given amount of feed, and can be automatically fed that amount with each stroke of the traverse.

It is fairly well agreed that production is at its highest point and quality when automatic feeding of abrasive wheels is adhered to. Excuses are made that automatic mechanisms get out of order; however, attention to the upkeep of these mechanisms reduces errors to the very minimum. Lack of lubrication is often the greatest cause of improper functioning of automatic feeds and mechanisms.

Truing up the faces of the wheels is a matter needing careful attention and study. A very light cut with a diamond is sufficient to true wheels that have been properly used, but by rough usage of wheels considerable abrasive is wasted; for example, the loss by removing, say, twenty thousandths from the face of a wheel 20 in. $\times$ 2 in. is $2^1/_2$ cu. in. Diamonds passed quickly over the faces of the wheels break up the surface for roughing, leaving the abrasive grains somewhat irregular, but for finishing, light cuts with the diamond over the wheel faces gradually wear the grains down to uniformity in height.

Copious flow of lubricant is essential for the precision grinding of cylindrical work; it serves to reduce the heat generated by the work and wheel, carries away the dust, and reduces the friction of the cutting particles of the wheel.

The correct use of steady rests is essential in the cylindrical grinding of many types of parts, and care should be taken that these are set right. The type of steady rest used varies with the work being done; often hardwood blocks suffice, but sometimes hardened steel shoes are necessary. Efficient steady rests insure high production, accuracy, good finish, and ease in operating. Incorrect supporting of the work sets up chatter in the wheels quite as much as do incorrect work speeds.

Attention should also be given to the driving units or dogs, as torsion in the driver when heavy cuts are taken will set up vibration, and when the chatter marks appear in spiral form the cause is usually traceable to vibration of the work spindle.

Chatter is also caused by glazed wheels and uneven thicknesses of driving belts, and wheels often glaze because of incorrect relation of the wheel and work speeds.

Care must also be taken with the tapers for work centers; sometimes larger center holes will overcome chatter, but it is important always to see that the tapers on the machine centers correspond absolutely with those in the work centers.

When chatter takes place after every precaution has been taken to insure perfect machine conditions and the wheels have been dressed and balanced, then inefficient supporting of the work is invariably the cause.

Centerless Grinding

Centerless grinding has made enormous strides in recent years. The principal elements are a grinding wheel, a feed wheel, and a work rest. The advantage of this method is the continuous passage of work pieces across the wheel, with the elimination of centering and driving units.

The work rotates at the speed of the feed wheel, this wheel being an abrasive bonded with shellac or rubber and having a coarse surface to prevent slipping.

In cutting, the grinding wheel keeps the work firmly against the feed wheel, and traversing is controlled by setting the feed wheel at a slight angle so that the centers of the feed and grinding wheels do not fall in the same plane; by slightly changing this angle and the speed of the feed wheel the traversing speed may be varied.

A great advantage of centerless grinding is that a smaller grinding allowance is possible as compared with the ordinary cylinder grinder, owing to the fact that the centerless grinder works to the

nearest full diameter, whereas the center-driven machine works from a radius only. Because of the rigid support given the work, heavy cuts are possible and uniformly accurate work assured. The increase in production on many parts now centerless-ground over the output of the ordinary cylindrical grinder is several hundred per cent.

Wheel Grinding and Its Limitations

Every one is more or less familiar with the operation and applications of the planetary-type grinder. It is widely used and has many advantages over boring, reaming, burnishing, or broaching. Bores can be held to size and free from appreciable taper, and a smooth finish obtained. Where the stock to be removed is not too great, it is frequently used to correct or produce the axis of the bore. Although it has been almost entirely replaced by honing in the finishing of internal-combustion engines, it is still the means used in finishing many sizes and types of cylinder bores which are not adapted to honing.

There are, however, objections to ground bores. The finish produced by an abrasive wheel rotating at high speed and fed slowly through the bore is a helical groove, the pitch of which is the amount of wheel advanced per revolution of the cylinder. The fuzz so generated is usually so minute that its shearing off on initial wear does not prove detrimental except in cases where a tight working fit is required. In such cases it is often followed by a honing or lapping operation.

The accuracy of this form of grinding depends considerably upon the skill of the operator and the condition of the machine. Owing to the difficulty of protecting spindle bearings from abrasive grit, it is extremely difficult to maintain them in condition to avoid play, resulting in out-of-round cylinders, where hard spots are encountered, and a tendency to chatter. Chatter marks may also be produced by glazed or unbalanced wheels.

A further objection lies in the operating cost, a fairly high degree of skill being required for operation of the machine. This type of grinding does not lend itself well to multiple operating as do boring, reaming, burnishing, or honing, so that work on multiple bore blocks is relatively slow and correspondingly costly.

Objections to Lapping

Lapping has been generally used to produce cylinder bores to the closest possible limits in accuracy and finish, but it is always slow and costly, and to a great degree dependent upon the skill of the operator. It has the additional disadvantages attendant upon the use of a free abrasive, and great care must be exercised in order to prevent cylinder walls from becoming enlarged with the grit. Its best application is in lapping two parts together. Honing, however, is the more satisfactory practice where parts of uniform size are to be produced independently of the piston or fitted part.

The Expanding Grinder or Hone

One of the most interesting developments in the finishing of cylindrical bores has been the introduction of the expanding floating grinder or so-called hone. The type of tool generally used is in the nature of an expanding segmental wheel, the grinding surface being composed of a series of abrasive stones placed longitudinally upon the circumference of a metal body with an adjusting means for expanding the stones to the walls of the cylinder being ground, and holding them under pressure against the wall, the tool itself being rotated and reciprocated simultaneously. A positive backing is given to the stones to prevent their contraction when meeting some resistance such as a high or hard spot in the bore, as spring-tension tools would ride over the high spots and dig into low or soft spots. Some idea of the extent of the application of this method can be gained by noting the type of work for which it is used.

At the present time the bores of nearly all internal-combustion engines are so finished, as are compressors of all sizes, in both household refrigerating units and large refrigerating plants, as well as commercial compressors of various types.

It is also widely used for finishing bushings, bearings, hydraulic cylinders, a variety of pumps, push-rod guides, connecting rods, gear hobs and bores, wristpin holes in pistons, long tubes, pneumatic drills and hammers, and air motors.

Although not yet the generally accepted method, it is finding a rapidly increasing use in finishing locomotive cylinders of numerous types, such as piston valves, air compressors, reverse-gear cylinders, fire-door closers, link-motion bushings, etc. It is not limited to cast iron, being used as well for grinding various steels, bronze, brass, aluminum, ebonite, stellite, and numerous alloys.

Advantages of Honing Methods

The chief advantage of the expanding-grinder or honing method is the ease with which close limits of accuracy are maintained, giving any desired finish at a high production rate and at a low cost. The accuracy of the work is not dependent upon the skill of the operator, and the method lends itself readily to multiple operation. It has a wide range of application as regards size of cylinders finished. It is being used for finishing the rifling of large guns as large as 16 in. in diameter and 65 ft. in length; seamless steel tubing 3 in. in diameter and 20 ft. in length, also push-rod guides $^1/_2$ in. in diameter, cornet valves, and slide trombones. It has proved especially applicable to the grinding of bearings having oil grooves, cylinders with keyways such as bores in gear hobs, cylinders with ports such as Diesel-engine liners, rotary valves, sleeves for sleeve-type motors, two-cycle engines, large air compressors, slotted push-rod guides, etc.

The successful finishing of such cylinders has been due to the type of construction employed, tools for such work having a minimum of six stones, so that while any one stone is passing over a keyway or port it is supported by the stones on either side of it, thus preventing its dropping into such depression. With this type of construction, tolerances of 0.0005 in. are easily maintained on most bores, and on many bores tolerances of 0.0002 in. are held.

Another type of work for which this type of tool has been found peculiarly suited is that of obtaining alignment between two bores such as in the finishing of piston bosses, machine-tool bearings, headstock bearings (plain bearings and seats for roller or ball bearings), piston-valve bushings, and printing-press bearings. This result is obtained by the use of a single grinder with a set of stones at either end, or by two independent grinders rigidly connected and used for grinding two cylinders of either the same or different sizes.

Compensation for Uneven Stone Wear

In order to obtain cylinders both straight and round some means must be provided for taking care of uneven stone wear, as it is evident that even though the abrasive be absolutely uniform and the stock to be removed be evenly distributed (either of which conditions seldom obtains), in the grinding of long bores the end of the stone in the direction of travel is worn down the more. This is compensated for by the use of floating cones which feed out the stone, keeping the grinding faces parallel. Accurate work with this type of tool has often been obtained when one end of each stone has been worn away $^1/_{16}$ in. more than the other.

In the operation of this type of tool it is necessary to reciprocate the head while rotating it, a ratio of reciprocating to rotating speed of roughly 1 to 3 having been found the most suitable for cast iron (65 to 70 actual surface feet per minute reciprocation

to 200 feet rotation). The purpose of combining these motions is to obtain greater cutting speed, the action of the stick being in a number of respects similar to that of a file in that by rotating the grinder only the stones will load with metal and glaze. The same would be more or less true of reciprocation only, as would be the case if a file were pushed straight across the work either sidewise or lengthwise. By combining these motions a shearing action takes place which produces faster and easier cutting, at the same time cleaning the face of the stone and preventing its glazing.

It has been quite consistently noted that if a stone is a little hard for the work and starts to glaze, it can be made to break down and cut freely by increasing the ratio of reciprocating to rotating speed. This of course, within certain limits, applies only to stones ordinarily suited for the particular type of work and does not make it possible to employ stones a number of degrees harder on the job.

The combined rotating and traversing motion not only cleans the face of the stone, but causes it to break down more evenly, giving a more uniform grinding surface and a closer average of errors in both material and abrasive.

The Coolant Used

In the grinding of cast iron it is practically universal practice to use kerosene as a coolant during the grinding operation; this also serves as a lubricant, causing the ready displacement of all broken-down abrasive particles on the face of the stone, and giving a much freer cutting surface. The kerosene also serves to clean the surface being ground, washing it clear of free particles of both metal and abrasive.

Kerosene is also quite generally used on other metals due to its low cost and the fact that it operates quite satisfactorily, although turpentine has been found to give a freer cutting action on bronze, brass, and some alloys. Mineral seal oil can be used on most operations but is more expensive.

In this type of grinding the use of water has been entirely unsatisfactory, not so much on account of its causing tools, machines, and work to rust, but for the reason that it causes the broken-down particles of abrasive and metal to adhere to the face of the stone forming a rather gummy mass.

Stones soaked with water before using with kerosene also show this tendency, as do also stones which have been immersed in motor or cutting oils.

Oil has not been found a satisfactory lubricant and coolant in itself, although recent tests indicate that a small percentage of spindle oil in the kerosene does not interfere with its action. Much experimentation has been made along the line of developing proper coolants, but so far kerosene has proved the most satisfactory.

The Abrasive to Use

For use on cast iron, 80-, 120-, and 150-grit abrasives have found widest use for roughing operations, and 120- and 180-grit stones for finishing, depending on the surface desired.

In all cylinders finished with 180-grit stones or coarser, criss-cross scratches are plainly visible, the rotating and reciprocating motion creating diamond-shaped figures throughout the bore. These scratches are not measurable in depth, being so slight that a few passes of a close-fitting piston remove at least 50 per cent of them.

Where turned rings are used, these scratches are essential to the proper seating of the rings, as is the case with most internal-combustion engines.

By the use of the finer abrasive stones (600 grit or finer) a mirror finish can be obtained, no scratches at all being visible to the naked eye. Such finishes can only be used to advantage in engines using ground rings or in hydraulic and compressor cylinders and the like where no seating of rings is required.

In the grinding of bronze, brass, aluminum, and aluminum alloys, fine-grit stones having a small percentage of bond have proved the most practical to use, as stones which are too porous or in which the grit is too closely held will have a tendency to tear and roll up the metal. In grinding such metals a copious flow of coolant is required, and where the metal is particularly soft, the application of paraffin wax to the face of the stone prevents loading and gives an improved finish.

One of the most interesting applications of this type of grinding has been in the finishing of seamless steel tubes, tubes as long as 60 ft. being finished on a production basis. Owing to the presence of scale on the walls, abrasive stones have not proved economical for the roughing operation on such tubes, and shaving tools known to the trade as "revocipers" have been employed; these are tool steel inserts which fit the grinder heads, replacing the abrasive sticks, and are operated by the same rotating and traversing motion as the grinder.

Each insert is made up of a series of cutting teeth with spaces between them for chip clearance, and so ground as to give a continuous cutting action in both directions of the stroke cycle. The interior finish of the tube therefore has the same characteristics as that of the ground tubes, i.e., criss-cross tool marks, these requiring from $1^1/_2$ to 2 thousandths (0.0015 to 0.002 in.) grinding to clean up. In actual production a stock removal of 0.007 in. per foot per minute has been maintained on 3-in. seamless steel tubing.

The value of this type of tool is not of course in the rate with which it removes stock, but rather in its adaptability to the finishing of tubes of lengths limited only by driving mechanism.

The success of this method of finishing can be accounted for in large measure by its adoption for gang grinding—4, 6, and 8 spindles being frequently used simultaneously—for use either on motor blocks or for the multiple grinding of sleeves or bushings held in fixtures.

In the automotive industry it is common practice to leave from 0.003 to 0.005 in. after reaming to be removed by grinding. The actual finishing time varies from 30 seconds to 2 minutes per block according to the nature of the block, quality of reaming operation, power of machine, and type of abrasive used.

In grinding airplane-engine and other thin-walled cylinders, generally nine or more stones are used in the grinder to prevent "breathing" of the cylinder or distortion of the thin walls through holding.

Another element entering into the success of the method is the fact that the accuracy of the work is not dependent upon the skill of the operator.

Conclusion

The advantages of the expanding grinder or honing method of finishing cylindrical bores are:

1 Perfection of finish both as to straightness and roundness.
2 Absence of fuzz, leaving a full metal surface as nearly as possible identical with a run-in bore.
3 Uniform density of surface maintained.
4 Accuracy not dependent upon skill of operator.
5 Adaptability to gang grinding with resulting high-speed production and low cost.

MSP-53-14

The Uses of 16-Mm. Movies in Industry

By MAYNARD L. SANDELL,[1] ROCHESTER, N. Y.

Industrial motion pictures may be readily divided into two distinct classes, namely, institutional and internal. Institutional pictures include productions employed for sales promotion and advertising purposes, or for any other use involving public exhibition. Such films should be made by the professional producer of motion pictures on standard-size 35-mm. film. Internal uses include instructing employees in safety, in machine-shop practice, in plant housekeeping, and in fire prevention, as well as making records of construction and research findings, and of testing, assembling, sorting, and packing operations.

This paper deals only with the latter class, describing briefly some of the many internal uses of industrial motion pictures, and showing how members of the plant engineering staff can make such movies on amateur standard 16-mm. film with inexpensive amateur equipment.



THE craftsmanship of cinematography may be readily divided into two phases; one relates itself to the amateur, the other to the professional. This paper deals primarily with the amateur phase of motion pictures in industry.

Amateur cinematography as practiced by the engineer has its limitations, and although he is furnished with portable, handy equipment with which he can easily obtain splendid results under those general conditions that circumscribe all amateur achievements, there is a point beyond which professional service will usually be needed.

For example, making movies for advertising and selling purposes does not, for reasons outlined later, fall within the category of the amateur's ability. Such productions certainly require professional technique and will not be discussed at length here.

Time-study and motion analyses by means of amateur movies have been purposely omitted, because a satisfactory exposition of such work demands more time than is available for this entire presentation.

The Economy of 16-Mm. Film

With the introduction in 1923 of 16-mm. film coated with a special, fine-grained, reversible emulsion, a practical, successful, and economical method of making motion pictures was made available for the amateur. The prime requisite was, of course, that they be relatively inexpensive; and, in addition, that the necessary equipment involved be portable and require no more than ordinary skill and photographic ability such as an amateur can easily and quickly acquire.

All of these requirements were met in the new film. Compared with standard 35-mm. film, the cost economy effected by the new 16-mm. width was well over 80 per cent. This great saving in cost, without sacrifice of either quality or length of picture, was made possible by the various economies involved, namely:

1 The new film was only 16 mm. in width as against 35 mm. for professional film.

2 Being less in width, each individual picture was proportionately less in height, so that 400 ft. of the amateur film was equivalent in projection time to 1000 ft. of standard commercial film.

3 With standard-sized professional motion pictures the negative had to be developed and a print made from it on positive film, and the positive then developed. By means of the newly devised photographic emulsion and a special chemical process, the 16-mm. film is first developed and then, without printing, reversed from a negative to a positive—virtually one operation as against three.

4 Before the advent of 16-mm. film, amateurs using the 35-mm. width had to send their films to laboratories whose facilities were organized for orders of 100 to 200 copies, and only in such quantities was the cost materially lessened, because of the time and expense involved in preparing the first master negative. The amateur seldom wanted more than one or possibly two prints, and it was for such accommodation that 16-mm. finishing stations were equipped, although as many duplicates as desired can be made at a nominal cost.

In addition, the film base used for making the new 16-mm. film was the slow-burning type made from cellulose acetate and generally known as "safety stock." While the cellulose nitrate film commonly used in motion pictures is safe when proper projection booths and other safety appliances are provided, its introduction into the home or industrial plant would be out of the question because of the fire risk naturally involved.

For this reason, the new film was made on the acetate base only and met the approval of the Underwriters' Laboratory standards for use where fireproof appliances were unavailable.

It was not long after its début that 16-mm. film attracted the attention of doctors, dentists, artists, and scientists, who turned to this simple and practical method of recording their technique for reference and teaching. The surgeon, for example, seeing his personal films on the screen, immediately visualizes the possibilities of 16-mm. movies applied to his professional technique.

Interesting work is being done in microcinematography with specially fitted 16-mm. equipment. The advent of this film has made the cine-photomicrography practical and economical for the research worker. The Bausch and Lomb Optical Company, of Rochester, New York, has perfected an apparatus using a synchronous motor in conjunction with the Cine-Kodak Model A, which, through a chain of gears, drives the camera at eight different speeds. The speeds range from 16 pictures a second to one picture every 4 minutes and 16 seconds. Such equipment permits focusing the microscope at any time without interfering with the photographic process. Studies involving

---

[1] Eastman Kodak Company. Mr. Sandell was graduated from Iowa State College with the degree of B.S. in Industrial Science. From 1925 to 1929 he was cinematographer in the visual instruction service at Iowa State College. In 1927 he was engaged in motion-picture work on barberry eradication for the Bureau of Plant Industry, U. S. Department of Agriculture.

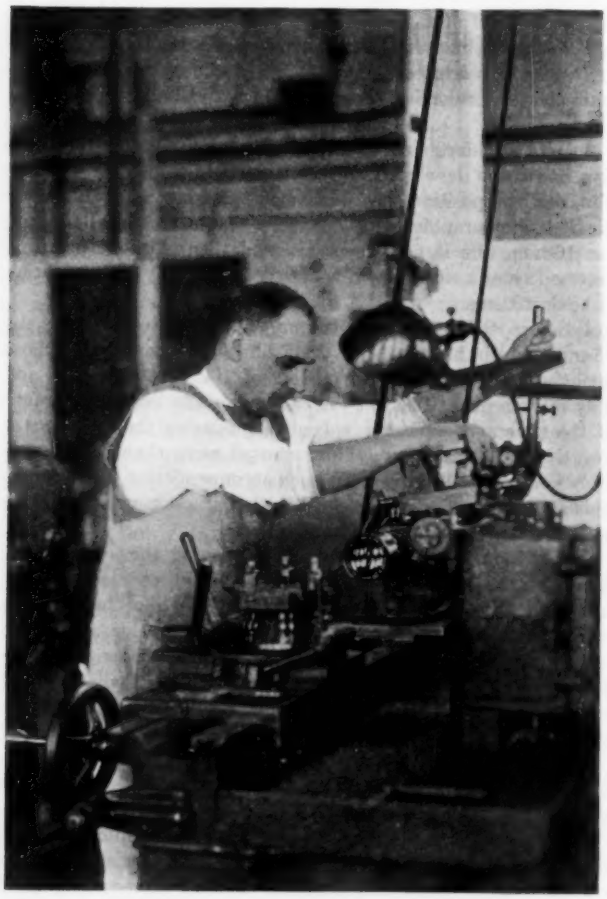
Contributed by the Machine Shop Practice Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

the formation and precipitation of colloidal suspensions, crystal growth and changes occurring in crystal structure during chemical reaction, and the study of Brownian movement in rubber latex, are examples of work being done. In the paint and varnish industry it is very desirable to study the action of the surface film of paint and varnish while drying. During this process certain phenomena occur which have a definite bearing on the quality and appearance of the dry surface. Many cine-photomicrographs have been made, and the information obtained has proved of value.

Uses of Industrial 16-Mm. Film Increasing

The adaptation of 16-mm. film to industry followed very shortly its adoption in other fields. The transition from its uses in pure science to the applied science of industry was,



Instructing Employees in Machine-Shop Practice Is Effectively Done by Means of 16-Mm. Movies

figuratively speaking, overnight. Experimentation by amateurs in and around industrial plants disclosed uses for the 16-mm. camera which had never been thought of previously.

An investigation of the industrial uses of photography was conducted in 1927 by Frank Seaman, Incorporated, of New York City, and the compiled data presented on December 23 of that year. In the course of this survey, 434 companies were called on in 62 separate industries. Actually, 43 cities in 12 states, representing the industrial centers of the United States, were visited. The report of this concern showed only 18 industries out of the 62 using 16-mm. movies, or about 29 per cent. In the interim of three years since the compilation of this report, the number of industries using 16-mm. film has increased to 32, or to nearly 52 per cent.

The 18 industrial users of 16-mm. in 1927 were classified as follows:

Commercial

- Automotive
- Explosives
- Food Products
- Hardware and Tools
- Textiles
- Machinery
- Publishing
- Radio
- Ship Building

Public Utilities

- Electrical Power
- Gas
- Telephone and Telegraph

Transportation

- Railroads
- Steamships

Businesses

- Financial
- Retail Stores
- Insurance

Education

- Universities



Performance Records of Machinery and Equipment Can Be Readily and Economically Made With the 16-Mm. Film Camera

By 1930 the following 14 industrial applications brought the number up to 32:

Commercial

- Airplane
- Cement
- Chemicals
- Construction
- Electrical Equipment
- Petroleum
- Iron and Steel
- Locomotive and Car Building
- Mines
- Paint
- Paper

Businesses

- Architects
- Commercial Photographers

Miscellaneous

- Dental Supplies

Sixteen-Millimeter Movies Used in Standard Practice and Employee Training

The compact, portable equipment for 16-mm. film permits recording the action of a machine or worker at close range without disrupting expensive manufacturing processes through distraction or interruption of the employee. For this reason, 16-mm. film lends itself admirably to work on standards of practice and process study.

Such films become an invaluable adjunct to process study when the acme of efficiency is sought. The minute details of a process cannot escape attention when a slow-motion pic-

ture is made of it. This is accomplished by a simple four-to-one attachment which gives, when projected, an action only one-fourth as rapid as the normal. Several models of the 16-mm. camera can be obtained with such high-speed attachments. In addition, the worker can study each element of a particular operation by running the machine by hand and projecting one frame at a time and studying each picture as a "still."

The Dunlop Tire and Rubber Corporation developed an efficient method for tire building. In attempting to describe this method for the benefit of the English Dunlop Company by the usual means of drawing, description, and time-study methods, it was found difficult to present the subject in such a manner that the English company could make use of it. In order to make the

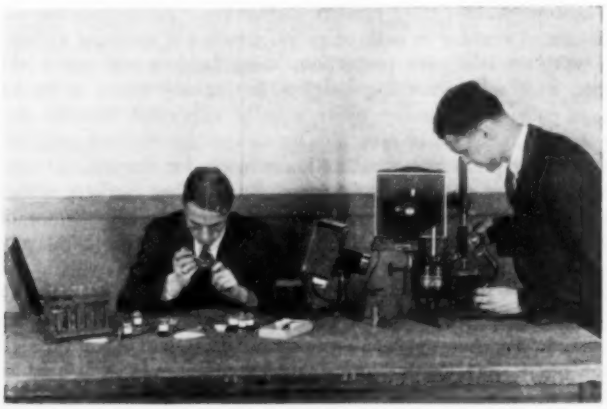


Scenes Like This May Be Staged in a Plant and Prove Very Effective in Safety Education

matter entirely clear, a 100-ft. 16-mm. film was made by an amateur and sent to the English company. The display of this film cleared up all points in doubt, and enabled them to make full use of the economical method. The demonstration was so pleasing to the English company that they in turn made a film dealing with an English process that was not clear to the American company, with like beneficial results.

There are many industrial operations such as assembling, sorting, and packing, involving action of such rapidity that the eye cannot follow them. Were it not for slow-motion pictures and single-frame projection, the exact nature of the operation would never be known except to the operator; and usually the movie explains better than the operator. An example of this nature was forcibly brought out a short time ago. An examination of a very efficient worker disclosed the fact that she was doing an operation entirely with her left hand, while her less efficient co-workers enlisted the aid of their right hands for the same operation. She explained her method many times and at great length, but still her explanations were unavailing. The rest of the workers had to use both hands. A slow-motion picture revealed an action of one of her fingers which was essential to her efficiency, but of which she was unaware. The other employees quickly acquired her method, and what was once considered the acme of speed and efficiency is now an ordinary rate.

The Western Union Telegraph Company have made some very interesting films for instructing their employees. One of these, a reel called "Gumming," is proving beneficial in



Photomicrography of Motion Is Made Practical and Economical With Apparatus Such as This
(Courtesy, Bausch & Lomb Optical Co.)



When Taking the Product to the Prospective Purchaser, Requiring a Well-Planned Scenario With Special Lighting and Professional Actors, Advice From a Professional Producer Should Be Sought

teaching girls how to apply the printed tape to the telegraph blank as it comes from the machine. Another reel, "Keyboard Errors," teaches the operators how to increase their speed and efficiency and lessen the number of mistakes by avoiding certain conditions or movements which are known to cause errors. The effects upon the resulting work of long finger nails, distraction, and other factors of this nature are forcibly brought out in these films.

Employees need safety education, and there is no better method of educating them than by the use of movies. Motion

pictures taken in the plant illustrate the particular points of danger, and the inevitable results of carelessness are driven home with an impression that is lasting. It is hard for a worker to believe that he is in danger or that he is not immune to injury. Consequently, though his very life may be in jeopardy, he becomes careless and some day meets with serious mishap. A film showing how the accident happened, and perhaps a close-up of the mangled and mutilated hand, though unpleasant, will prove a stimulus to caution in the future. Many little scenarios which could be staged around the plant for safety education immediately present themselves. Films showing the danger of running in halls or alleys, grinding or chipping without goggles or other eye protection, using ladders without safety feet, smoking or carrying matches, letting oily waste or wiping rags collect, and many other equally important hazards can easily be made on 16-mm. film by an amateur cinematographer, employing plant personnel as his subjects for the sake of convenience and to bring home forcefully to other workers within the plant the points of his pictures. Here is one "ounce of prevention" which pays large dividends on a comparatively negligible investment.

To give a new employee a broad knowledge of the company for which he works, and to make him a part of it in the least possible time, there is nothing so effective as a movie trip through the plant. He learns more in fifteen minutes by this method than he would in a month through the regular channels of absorption.

Employees can be taught how best to avail themselves of their leisure time through the judicious use of movies of sporting events and competitive games shown in their recreation rooms. Such procedure kindles the spirit of fair play and friendly competition in the workers' minds, and a rising standard of morale and esprit de corps results.

Adaptation of 16-Mm. Film to the Research and Testing Laboratory

The 16-mm. camera is a faithful ally of the industrial-research laboratory. It is an excellent observer and not subject to mistakes involving the human element, or "personal errors." The camera can be equipped in the laboratory so that the operator can take a single picture at any desired interval; or the highly paid technician can set it to "watch" a certain process while he devotes his valuable time to other work. Here again the microcinematographic apparatus may be used. By removing the device connecting the camera and the microscope and putting a photographic lens in its place, the camera will take a picture at any one of the eight intervals according to the gear ratio used.

Pen recorders are not available for each new branch of research, and it would be a needless expenditure to have them made when a movie camera can be used for many phases of the work. One 100-ft. roll of film will make 4000 recordings. It is advantageous for reference purposes to photograph all new developments while under test. The small movie camera will "see" the whole procedure from a vantage point too close for safety by any other method of observation. Tests are being recorded this way on experimental high-tension transformers; explosives; automobile and airplane motors running at excessive speeds; characteristics of new alloys while under compression, tension, shearing, and torsion tests; and the crushing strength of concrete piles, beams, and culverts. An interesting application that has come to the author's attention is that of an aviator, piloting a cabin-type plane, who uses a 16-mm. camera mounted on top of the wing to "see" the action of the elevators and rudder while under stress in flight. Undoubtedly there are many other uses than those mentioned.

Building contractors and structural engineers are finding 16-mm. film a valuable asset for making permanent, authentic, visual records of their construction work. By taking a few feet at regular intervals, the entire building process is recorded from turning the first bit of soil to driving the last rivet. A construction project, taking months to do, can in this way be shown on the screen in a few minutes; and yet at any phase of the work the projector may be stopped and the scene viewed as a still picture. By dated titles for scenes of particular phases of the construction, the rate of progress between periods may be noted. Such information proves a valuable aid for computing costs based on rate of progress when undertaking future contracts.

Industrial concerns having branches and stock houses find the use of 16-mm. movies a valuable aid for disseminating knowledge or information to the other houses. Especially is this effective in the case of foreign branches. The more information an employee has concerning the product which he helps to produce and sell, the greater an asset he is to his company. Realizing this fact, several far-seeing employers have made 16-mm.-width movies among their various plants for interplant exchange and the edification of their employees.

Necessary Equipment for Making 16-Mm. Movies

Except where there is an abundance of window light, motion pictures within the factory or laboratory will require some artificial light. For this purpose, three to five portable flood lights of 500 watts each will be sufficient. The author has obtained very excellent results using "Kodalites." One of these lamps, when placed six feet from the subject, will illuminate, sufficiently brightly, an area of four square feet. For an ordinary factory close-up, more than three of these flood lights are unnecessary.

The camera should be equipped with an f.1.9 lens, which permits pictures with a minimum of artificial light, and may be either hand-cranked or spring-motor driven. An f.1.9 is a very fast lens of extremely wide aperture and has a speed equivalent to nearly 20 times that of a commonly used Kodak. A lens of lesser aperture should not be used because of the additional number of flood lights which would be required. The spring-motor-driven camera is usually held in the hands and takes, for each winding, about 15 ft. of film, which is equivalent to about 40 seconds on the screen. The hand-cranked model is used on a tripod and is capable of taking 100 ft. or a little over four minutes of projection time. Any dealer in amateur cine equipment will give what simple instructions are necessary for using lighting equipment, reflectors, backgrounds, etc. As a usual thing a few tests will solve the particular problem at hand.

The film used for making industrial 16-mm. movies is available in two emulsions, regular and panchromatic. The regular is suitable for all general purposes in and around the plant, and the cost is only about 20 per cent as much as the corresponding standard film for reasons explained earlier in the paper. Whenever the nature of the subject necessitates a correct rendition of color in monochrome, panchromatic film should be used. Regular film is mainly sensitive to violet and blue light only, very slightly sensitive to yellow and green, and practically not at all sensitive to red. Thus, when a picture is taken with ordinary film, violet and blue are the only reflected colored lights that make appreciable impressions on the film. Other colors to which the film is not sensitive make no impressions, and therefore show as black on the screen. When panchromatic film is used, correct color values are rendered in varying shades of gray, because the film is not only sensitive to violet and blue, but red, green, orange, yellow, and all other colors of light are correctly recorded. Brass fittings on a steam engine, for example, will not give the desired contrast with the darker steel parts when ordinary film is used, because of the yellowish light re-

flected by brass. Panchromatic film will increase this contrast to the normal which the eye sees because of its relative yellow sensitivity. In addition, panchromatic film, when used with Mazda lamps, requires less light than the regular film. The spectrum from a tungsten-filament light source has a predominance of red and yellow, the effect of which is lost on regular film because of its "blindness" to light of this color, as explained above. By using panchromatic film, the extra light is utilized, and good pictures can be taken where the light would be inadequate for regular film.

It is impossible within the narrow limits of this paper to explain all the uses and potentialities of 16-mm. motion pictures in industry. The very fact that the whole procedure is extremely simple and economical, requiring no special technique for making permanent, visual records, is assurance enough that its possibilities will increase as each new industrial use is found. And, in addition, the fact that duplicate prints may be made at a very nominal cost also enhances the value of 16-mm. industrial film. These "dupes" are printed from the original film and put through the regular reversal process at a cost of less than 40 per cent that of professional film. About sixty of these duplicates can be made from an original film without appreciable loss in quality.

Since only the merits of 16-mm. originals have thus far been discussed, it may be well at this point to say something regarding the use of 16-mm. movies which have been reduced from professional 35-mm. stock. These films have the advantage of the professional touch. There are, necessarily, many limitations imposed upon the use by the amateur of 16-mm. film in the factory; and when one or more of these contingencies arise, the professional cinematographer should be called in. When taking the product to the prospect requires a well-planned industrial scenario; when animated drawings may be needed to explain visually the ultra-technical details of the business or product; or when many interiors presuppose carefully regulated artificial lighting equipment, then, indeed, should professional help be sought. No amount of gaily colored sales and advertising material or cleverly phrased oratory can ever efface the stigma in the mind of the prospective customer of a poorly produced movie. For this reason it is advisable to have institutional movies made by a professional on standard film in order that the manufacturer may put his best foot forward in presenting his product to the customer, or in carrying the story of his plant or product to the public.

Again, to lower costs on duplicate prints and facilitate showing the finished picture, 16-mm. film comes to the rescue with the reduction print. These films are made by reduction printing from the original standard negative, and the salesman or demonstrator is thus enabled to carry 16-mm. prints of professional quality to show to his prospect by means of a light, compact, portable projector.

Conclusion

By way of summary, it may be said that the use of motion pictures in industry is readily divided into two distinct classes, namely, institutional and internal. The institutional use involves those productions employed for sales promotion and advertising where the film carries the story of the product or plant to the customer or public. Such films should be made by the professional producer of industrial motion pictures on standard 35-mm. film. The internal use is that made within the plant for instruction of employees in safety, shop housekeeping, and fire prevention; standard practice and employee training; layout and process engineering; recording research and testing operations; records of construction and building progress; and many other purposes in and around the plant. Uses such as these in which the film is not to be shown outside of the organization readily lend themselves to amateur technique and 16-mm. film and equipment. The convenience, simplicity, and low cost involved easily justify such applications.

Selected Bibliography

1 "Motion Picture Photography for the Amateur," by C. E. K. Mees. *Journal of the Franklin Institute*, vol. 196, 1923, pp. 227–245.

2 "Future Developments in the Sixteen-Millimeter Field," by H. C. McKay. Transactions of the Society of Motion Picture Engineers, vol. xii, no. 36, 1928, pp. 1115–1120.

3 "A Microcinematographic Apparatus for the Owner of a 16-Mm. Motion Picture Camera," by Heinz Rosenberger. *Science*, vol. lxxxi, no. 1836, March 7, 1930, pp. 266–267.

4 "Moving Pictures Help Move Tractors," by Walter Gardner. *Advertising and Selling*, vol. xiv, no. 7, January 22, 1930, p. 30.

5 "An Apparatus for Motion Photomicrography of the Growth of Bacteria," by Stanhope Bayne-Jones and Clifton Tuttle. *Journal of Bacteriology*, vol. xiv, no. 3, Sept., 1927, p. 157.

6 "Motion Photomicrography With the Cine-Kodak," Transactions of the Society of Motion Picture Engineers, vol. 11, no. 30, p. 213.

7 "Motion Pictures of Micro Organisms," Bausch & Lomb Optical Company, Rochester, N. Y.

8 "Kinematographic Studies in Aerodynamics," by Alexander Klemin. Mechanical Engineering, March, 1928, p. 217.

9 "Dramatizing Industrials," by Paul D. Hugon. *Movie Makers*, vol. iv, no. 12, December, 1929, p. 792.

10 "Tabloid Industrial Scenarios," by Paul D. Hugon. *Movie Makers*, vol. v, no. 9, September, 1930, p. 540.

11 *International Review of Educational Cinematography*, March, 1930, p. 271.

12 "Photography," New International Encyclopedia.

Discussion

Allen H. Mogensen.[2] The principles of motion economy are so tied in with machine and tool design that it is wondered why it is not more effectively used. One manufacturer of milling machines uses the film to good advantage in order to make for more efficient operation of the machine and the tool equipment. He is also able to show the advantages gained by proper design to prospective customers with the films. Many of the present machine tools require that the operator be an acrobat in addition to his other necessary qualifications. Machine design and tool design must incorporate far more than mere ease of fabrication and low cost of production. In addition to considering fits and finishes, processing, and assembly, the designer and product engineer must also think of the man who is to use the tool or operate the machine, and it is here that a thorough knowledge of the principles of good motion practice will prove invaluable.

At the plant of the Cadillac Motor Car Company a motion-study laboratory has been established. This is used for instruction, not for theoretical solution of production problems. Groups of foremen, supervisors, time-study men, tool designers and makers, set-up men, and product engineers here learn the fundamentals of motion economy. It is here that the film has definitely proved its value, not as a means of making detailed micro-motion studies, but in teaching these principles to those who can directly and effectively lighten the work of the man in the shop.

A double die can be used in place of a single one, thus often at least doubling the productive capacity of machine and operator. The use of both hands may enable the operator to keep the machine running a greater portion of the time. Drop deliveries may not only greatly increase production, but enable a rhythm of operation more in keeping with the machine capacity. Redesign of the control mechanisms of the machine, bringing the control within the normal working area, may be a solution to what was before a serious problem in fatigue. There are many actual examples where the consideration of these simple principles

[2] Assistant Editor, *Factory and Industrial Management.*

during design or in redesign resulted in inconceivable increases in production, with consequent reduction in cost, and often solved a distressing labor problem.

One more important point. We are all concerned with the safety of the man who must use the tool or operate the machine. Often the application of these principles of motion economy may solve a problem that seemed insurmountable. Instead of slowing down production or otherwise adding to the cost of the ultimate product, thinking in these terms may give a surprisingly simple solution. We may so engage both hands in a useful portion of the cycle at the correct time as to eliminate the hazard. Thinking along this line will open up many new possibilities.

Back of all these principles of good motion practice stands the motion-picture camera, one of the most valuable production tools of today. Without detracting from its value in the field of micro-motion study, it can be stated that as a means of making a whole organization "motion-minded" the camera is supreme.

DAVID B. PORTER.[3] The writer wishes that the author had said more about the very important and far-reaching use of the 16-mm. movie in motion-study work. It would seem as though a treatment of this application would be of primary interest.

The degree of refinement and specialization to which machine design has been carried has tended to reduce the amount of machine time in proportion to the amount of manual time in the productive cycle of a unit of output. This places greater emphasis on the operator's part in the productive cycle. It therefore becomes increasingly imperative to improve his methods in order to secure the maximum of useful work with a minimum of motion and effort. Since the motions which a machine operator makes are partly predetermined for him by the machine, the designer should have a knowledge of the principles of motion and should incorporate these in his design. Also, this same knowledge should be applied to the design of jigs and fixtures so that they may be used with the greatest motion economy. The foreman, time-study man, instruction boss, and operator need to understand these principles so that all factors surrounding the operator's work may be analyzed and reduced to "the one best way."

Several companies have had considerable success in conducting classes in motion study for the purpose of teaching their people the principles of motion and of training them to think of work in terms of motion. For this they have found the motion-picture film most convenient for bringing into the laboratory or class room the material which the instructors may use in teaching. The motion-picture camera will continue to be used by the expert in resolving jobs that are important enough to warrant such detailed analysis, but such jobs must be relatively few in comparison to the countless number where time, let alone expense, would not permit of such refined treatment. It is to these jobs that a personnel trained to think in terms of motion will bring an attitude of mind and a method of analysis that will be not only most far-reaching in effecting economies, but in making the jobs more interesting to the workers. It is for such training of a selected personnel that the motion-picture equipment is indispensable and is likely to find its greatest usefulness in the motion-study field.

JOHN M. LESSELLS.[4] In recent years there has been an increased application of the motion-picture camera to research problems. For instance, there have been the applications by Drs. Prandtl and Tietjens to the study of flow of fluids around different bodies and the application by Messrs. Baud and Peterson[5] to the study of gear-tooth stresses. At the recent Congress for Applied Mechanics in Stockholm, a very interesting paper was also presented showing the action of stress on models using polarized light. All these activities show that this type of camera is valuable as a research tool, and in this respect it would be of interest to hear from the author just what is the maximum number of pictures which can be taken per second by the most modern 35-mm. equipment.

R. FAWN MITCHELL[6] AND M. W. LARUE.[6] Progressive factory managers are awake to the vital necessity of watching every operation and effecting every economy that will cut down overhead and improve the output. After all is said and done, a factory is the essence of motion, so that it is not surprising that a motion-picture camera has found such favor in assisting factory managers to solve many of their problems—mechanical, training, and research.

Pictures of intimate phases of manufacturing operations are used by salesmen to convince the prospect of the care used in making the equipment and provide a very effective selling talk. However, the principal interest in this paper will be confined to those uses of motion pictures that more directly touch upon the factory manager's peculiar problems.

A close community interest among employees is an invaluable asset to any business, and a great number of manufacturers are using motion pictures within their organization to promote this interest.

In conjunction with clubs of various kinds, activities outside of working hours, and in self-government projects, motion pictures grasp the interest and the imagination of the employee, giving him a sense of actuality of the thing before him.

For general entertainment, employees frequently enact movie plays themselves, exhibiting their screen prowess to fellow employees at regular meetings. Many subjects can be leased or rented from the many libraries of motion-picture film located in practically every town of any importance throughout the United States.

Now to consider the practical adaptation of movies to production problems in the shop, there are two natural questions that will be asked:

1 How can motion pictures be used to solve my particular problem?

2 What is the most economical method of securing results, what is the cost, etc?

The writers do not wish to create the impression that they are authorities on the subject. However, at our factory in Chicago we have, to some extent, studied the application of the motion-picture camera to various phases of factory production, and time and motion studies have engaged the attention of a number of our engineers.

Motion-study men tell us that motion study is not a speeding-up process. On the contrary, it seeks to find the one best way of doing a job, which is usually the easiest way. Time study reveals that one man makes four motions to do the job, while another uses six. Obviously, the latter should be shown how he, too, can increase output and improve the work, in order that he may increase earnings and decrease labor.

A competent engineer with a stop-watch can make time studies and motion studies and return to his office with figures which, when analyzed, can be used to definite advantage. If a motion-picture camera is added to the stop-watch he can then make a micro-motion study of the operation to be analyzed. The engineer not only has his stop-watch readings, but a visual record

[3] Assistant Professor, Industrial Engineering, New York University. Mem. A.S.M.E.

[4] Engineer, Research Department, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

[5] "Load and Stress Cycles in Gear Teeth," by R. V. Baud and R. E. Peterson, *Mechanical Engineering*, September, 1929, p. 653.

[6] Bell & Howell Company, Chicago, Ill.

of the operation synchronized with the time element. This can be reviewed time and time again, with the assurance that the personal element, in the use of the stop-watch, is also eliminated.

To make a micro-motion study, the operation to be analyzed is photographed, including in the field a microchronometer or stop-watch if desired. There is a type of microchronometer now on the market which operates by a synchronous motor which is ideal for the purpose. By recording photographically the movements of the operation and the movement of the clock, time may be recorded to within 0.002 of a minute.

With the recent improvements in camera design for the 16-mm. amateur film, motion-study work is made particularly easy. Among the most recent improvements in camera design of tremendous value to the analyst are the turret head and the critical focusing device. The use of lenses of varying focal length permits the placing of the camera in such a way as not to interfere with the routine performance of the operator or operation and enables the picture to be made right in the shop.

Of particular importance is the fact that such cameras are obtainable with a range of various speeds that permit photographing an operation in normal speed and then turning around and photographing the same in semi-slow motion. The speed of the camera can be adjusted to the speed of the operation so that the action can be slowed down just enough to facilitate analyzing the action without slowing it down so much that the essential sequence or rhythm of the operation as a whole is lost sight of.

For simpler operations it is possible to do away with the clock by running the film through the camera at a constant speed; thus if the film passes through the gate at 1000 frames per minute, the elapsed time between any two frames is 0.001 minute. An element of motion occupying 2 ft. of film, or 80 frames, must have taken 0.08 minute to perform. Thus the camera alone can be made to fulfil both functions. The highest class of spring-driven cameras now available can be considered accurate to within 2 per cent. This is satisfactory for quite a large number of operations.

Probably the most satisfactory method of doing this type of work is to include the clock in the picture area, as suggested previously. This permits individual analysis of each frame so that the exact time elapsed for each portion of an operation can be determined accurately. At the end of the paper will be given references to articles describing different methods of doing this work for those interested in following it up.

There are some special types of machines now in common use, such as weaving machines, folding machines, and other automatic equipment for handling complicated operations. Every one knows how easy it is for these machines to get out of order, and also every one knows how hard it is to place one's finger on the exact cause of the difficulty. Here the super-speed camera proves its value. It is possible to take slow motion pictures of extended length with professional cameras. However, there is available a compact, spring-driven camera operating at the same speed as the professional camera, namely, 128 pictures per second, giving a picture length of about 5 or 6 seconds. Ordinarily this is sufficient to analyze the usual run of such operations so that one can readily follow on the screen happenings which are too fast for the human eye to correctly analyze. The action is slowed down eight times by taking pictures at this speed. By slowing down the projector this can be slowed down still further, though naturally this is not recommended unless the circumstances render it imperative.

Satisfactory movies taken of intricate operations tie up very nicely with ordinary time and motion study. By their aid employees can be shown the importance of different phases of an assembly very quickly and very vividly. It helps materially in enabling them to appreciate the importance of their particular part of the job in the whole. If the average employee is able to appreciate just how much his fellow worker has to rely on his workmanship, it is not a hard idea to sell the employee the importance of exercising sufficient care to have the pride of workmanship necessary to turn out a good job.

An interesting example of the use of pictures for this type of work is to be found in the telephone companies. These companies train switchboard operators not only in the actual manipulation of plugs, but also in the understanding of the circuit and the complicated wire traffic which they never see.

Automobile manufacturers train dealers in correct servicing methods, how to perform the various types of service in the most approved factory manner.

Airplane companies, motor companies, and practically every type of manufacturer can use motion pictures in analyzing and for training. There is no operation, prowess, or routine of any nature but that can be taught better by movies.

The General Electric Company has developed an interesting side line to pictures originally designed for the purposes of internal instruction. These films are sent to other factories and help more than anything else to enable branch managers in different countries to correlate their productions with that at the main factory. It was soon found that there were numerous requests for permission to show these films to high schools, universities, etc. Naturally, films shown under these conditions to budding engineers, etc., had a material sales value that, if anything, was even more effective because they were not designed to act as a selling film.

In conclusion, we would like to mention a few thoughts on the possibilities of motion pictures for research work. Possibly the average factory manager is not primarily interested in pure research, but there are many things which it is desirable for them to find out for certain. For instance, in order to check the qualities of their lubricating oil, motion pictures taken through the microscope have been found very valuable. These pictures can also be taken in color as the color of the oil often acts as a guide to its other qualities, and in this manner a permanent record is kept, which is valuable for future reference. All engineers are familiar with the value of still photographs taken through the microscope showing steel structure. Motion pictures taken of similar subjects, say under strain, etc., are providing information of great value as to the action of different steels under different types of stresses. The same thought can obviously be applied to other metals, alloys, etc.

The 16-mm. amateur size film is comparatively cheap so that if the results are not satisfactory the first time, they can be shot over without inconvenience or more than the loss of a couple of dollars' worth of film. Therefore, there is every inducement for the factory manager to investigate the possibilities of this new tool. Sufficient work has been done along this line so that it is possible to state without equivocation that motion pictures provide a most effective means for a factory manager to improve his production, to train his employees, and to do research work where records, in motion, are available for comparison at a later date.

The following articles have appeared in *Factory and Industrial Management:*

"The Movie Camera, an Aid in the Search for the 'One Best' Method," by Allan Mogensen, June, 1930.

"Operation Analysis With the Motion Picture Camera," by M. A. Dittmar, Ph.D., Assistant General Manager, Lehn & Fink, July, 1930.

"Training Time Study Men," by Allan Mogensen, Sept., 1930.

"Micro-Motion Study Applied to the Manufacture of Small Parts," by R. M. Blakelock, General Electric Company, October, 1930.

Author's Closure

Replying to Professor Porter, the author would say that, due to the variety of methods proposed for applying the motion picture to time and motion analyses, it is considered inadvisable at this time to include more than a mere mention of any specific method. The following, however, are the three most commonly used methods:

The Gilbreth system uses a microchronometer or fast-moving clock which is photographed simultaneously with the worker. The advantage of this system is that the microchronometer record is independent of film speed—the camera may be cranked at various speeds. These clocks were expensive, and required frequent winding and calibration, and in addition it is difficult to keep the instrument in the field of view, in focus, and out of the operator's way.

The second method uses a stop-watch mounted at a fixed distance from the camera with a small electric bulb and special lens. It is Joseph A. Dubray's method, and insures correct exposure and focus at all times, but it often interferes with the field of view, just as does the Gilbreth.

The third system uses a constant-speed camera wherein the actual time is integral with the film. If the speed of the camera is made constant and changed from its present standard of 960 to 1000 frames a minute, the time between succeeding frames will be 0.001 minute, and the time of a period may be obtained by counting frames. Knowing that 16-mm. film has 40 frames to the foot, a ruler becomes the measuring device for the time element.

By attaching a small synchronous motor to a hand-cranked camera, very accurate results may be obtained throughout the entire roll of 100 ft., which is equivalent to 4 minutes. The accuracy is dependent, of course, upon how closely the frequency is regulated at the power station, but ordinarily this is within one-half cycle; so the error is considerably less than plus or minus 1 per cent. Such a method would have none of the inherent disadvantages of the systems now in use.

Replying to Mr. Lessells' question, undoubtedly the ordinary motion-picture camera is limited to speeds not exceeding 256 pictures a second, and in many cases 128 is the maximum due to the strain imposed upon the delicate mechanism by driving it at such an excessive rate. This speed retards action eight times, which is sufficient for studying the way in which a turning leaves the bar on a lathe or in which a milling tool or twist drill works. Micro-motion studies seldom require rates as high as this.

C. Francis Jenkins built a special camera to take 3200 pictures a second. Heape and Grylls designed a camera for the British Admiralty with which, by photographing 10,000 pictures a second, they were able to show the action of shells penetrating hardened-steel targets.

By using a film wound on a cylinder and feeding the cylinder ahead by a spiral while rapidly turning, Baron Shiba of the Aeronautical Research Laboratory, Tokyo Imperial University, was able to make 20,000 exposures a second.

The device used a system of polished mirrors, turning synchronously with the cylinder, which made the intermittent mechanism unnecessary. Later Toyotara Suhara made 40,000 exposures a second by the same system.

These pictures were slightly smaller than the standard frame, but were enlarged and printed on standard film by a special machine.

MSP-53-15

Cemented Tungsten Carbide for Cutting Tools

By L. J. ST. CLAIR,[1] PHILADELPHIA, PA.

In this paper the author tells of the origin of cemented tungsten carbide and the most satisfactory method for making it. Because of its hardness its use for wear-resisting tools is valuable, and particular emphasis is placed on its use for cutting tools. The physical properties of cemented tungsten carbide and its economic value as a cutting medium are discussed. Design and care of the tools are essential factors in successful application. Proper support for the cemented-tungsten-carbide tip and allowance for proper clearance angles are essentials of design.

The tool set-up should be as rigid as possible, as chatter or vibration is detrimental to successful life. The critical speed at which the tool should run for a particular job should be determined, and the machine be operated below this. Grinding of the tool is important, and the author tells of proper grinding methods. Use of cemented tungsten carbide for cutting tools has resulted in increased machine speed, longer life per grind, reduced tool maintenance, less waste of material in setting up, and greater accuracy throughout a long run.



THE COMPARATIVELY recent advent of cemented tungsten carbide into the field of cutting mediums is apt to convey the impression that this material has been known only a short time. However, such is not the case, as tungsten carbide has been known for more than 40 years, but due to certain inherent physical weaknesses and the difficulty of using its extreme hardness to good advantage, little general interest in this material has developed.

The economic pressures brought to bear on Germany with the coming of the World War made it necessary to obtain or find an adequate substitute for diamonds in drawing tungsten wire for lamp filaments. As a result, Messrs. Baumhauer and Schroter, metallurgists at the Osram Lamp Works, Berlin (an affiliation of the General Electric Company), carried on certain experiments with tungsten carbide and soon discovered that the essential properties of strength and toughness must be gained if this material was to become of commercial value.

Further experimentation disclosed the fact that metallic cobalt could be utilized for this purpose. The combination of these two materials resulted in what is now spoken of as cemented tungsten carbide.

Messrs. Schroter and Baumhauer first applied this material, cemented tungsten carbide, to wire-drawing dies as a substitute for the diamonds they had found so difficult to obtain. The success of their experiments was proved by the fact that they got equal, or better, results. Their next step was the replacement of diamond tools for the machining of various Bakelite compositions, which, as all know, present certain machining difficulties in their extreme abrasive action on the edge of a cutting tool.

Their success on this second application was the signal for considerable interest in this new material, and its possibilities were quickly recognized by interested companies who secured the rights to exploit further and apply the material. The physical properties of cemented tungsten carbide are interesting and give indications as to which of its characteristics may be taken advantage of or compensated for in iis application to cutting tools for various materials.

1 Its extreme hardness is perhaps the best known quality. Measured on the Rockwell "A" scale, its hardness lies between 87 and 92. These figures are more interesting by remembering that infinite hardness on this scale is designated 100.

2 Its coefficient of expansion is only half of that of Invar steel, which has the lowest coefficient of expansion of any known metallic substance.

3 Its electrical conductivity is low.

4 Its compressive strength is greater than that of any known material.

5 Its tensile strength determined by a transverse rupture test is somewhat more than half of high-speed or tool steel, although this value depends on the cobalt content of the material, as well as the treatment which the individual compositions receive throughout their preparation and manufacture.

6 As expressed by Young's modulus of elasticity, its bending modulus is almost three times that of steel, and is the highest modulus of elasticity of any known material, coming well above tungsten.

As might be expected, the metallurgy of combining two powdered constituents to produce this metallic substance differs considerably from the metallurgy of common practice and requires a new and different technique of procedure.

Cemented tungsten carbide is made from powdered cobalt and powdered tungsten carbide. The two powders are mixed by ball-milling, and various tests have indicated that during this milling operation a coating of cobalt is formed around the extremely hard tungsten-carbide particles. It is for this reason that the grain size and milling time become such important factors in producing cemented tungsten carbide with the required properties of hardness and strength.

This new combination of powders is then pressed into shape on hydraulic presses, after which this pressed bar is fired to increase its cohesive strength to a point where the mass is workable and may be easily cut, shaped, or turned. The final step in the production of a cemented-tungsten-carbide blank is the sintering, which is simply a firing process. The blank is placed in a hydrogen atmosphere, brought to a high temperature, which is maintained for the required length of time, and then removed. It is then ready to be attached to a tool shank.

A cemented-tungsten-carbide blank is attached to a tool shank for several reasons, among which are: The economy of such a practice, as it is only necessary to obtain a cutting edge, and secondly, as has been already pointed out, the physical characteristics of this material have indicated that this is the logical way to utilize this material to its best advantage.

Due to the tendency of the cobalt to combine with oxygen,

[1] Manager, Philadelphia District, Carboloy Co. Mr. St. Clair graduated from the University of Maine in 1923, and for several years was connected with the General Electric Co., investigating and installing wage-payment and planing systems. He joined the Carboloy Company in 1929.

Contributed by the Research Committee on Cutting Metals and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

considerable experimentation had to be carried on before the present method of brazing the cemented-tungsten-carbide blank to the tool shank with copper in a hydrogen atmosphere was perfected. Tests were made of attaching the cemented-tungsten-carbide tip mechanically and by welding, but these were discarded in favor of the copper braze, which is the practice generally followed today.

The value of cemented tungsten carbide as a cutting tool is undoubtedly the most interesting phase. What will it do? That can be best answered by pointing out the advantages that have been obtained in its use, throughout the industrial world, on many different types of applications. Because of its inherent hardness, its application to gages and other tools that must of necessity be wear resisting, has proved it a decidedly valuable material in maintaining the close tolerances which are an everyday part of quantity-production life.

It is true that up to the present time the greatest emphasis has been placed on its application to cutting tools for ferrous and non-ferrous metals, and its value on applications of this type is more tangible and easily distinguished. Briefly, they are as follows: Because of its great ability to resist abrasion, longer tool life naturally follows, as cemented tungsten carbide will hold its keen cutting edge under severe conditions, much better than any other material commercially available. This longer tool life means a saving in man power and machine power through the elimination of the lost time now expended in setting up and grinding tools. This is in effect a double saving, as it converts that time into productive effort. Again, it has been found possible to run cemented-tungsten-carbide tools at much higher machine speeds than has heretofore been the practice with other tool materials. In many cases, the increase in machine speed has been limited by the machine tool itself. It is obviously an advantage to decrease the machining time per piece, because this means a lower production cost and a greater return on the capital invested in machine equipment and man power.

In addition to this, these wear-resisting qualities make it possible to obtain longer runs without any sacrifice of accuracy. The quality of finished parts is greatly increased, due to the fact that this material is not limited in its ability to machine parts of various hardnesses successfully. That is simply another way of saying that material which heretofore had to be scrapped, due to inaccuracies, no longer goes to increase the size of the industrial world's scrap heap.

Another extremely important value of cemented tungsten carbide is seen in the releasing of certain alloys which could not be commercially machined until the introduction of cemented tungsten carbide as a cutting medium.

Knowing something of the physical properties of cemented tungsten carbide and having some appreciation of its economic value as a cutting medium, the next query is: What are the factors that lead to its successful application? These are: (1) Design of a cemented-tungsten-carbide tool, (2) the machine tool, (3) care of the tool in operation, and (4) grinding.

Tool design is fundamentally a question of getting sufficient support for the cemented-tungsten-carbide tip, and at the same time allowing for the proper clearance angles. Generally speaking, the tip should be sufficiently large to withstand any strain placed on it, and the shank should be designed to have the greatest top-to-bottom dimension possible, so as adequately to support the cemented-tungsten-carbide tip. Because of the vast difference in the physical properties of cemented-tungsten-carbide, as compared to high-speed and other tool steels, the maintenance of minimum clearance angles is very important. In general practice a front angle of 6 deg. or less, depending on the diameter of the work, has been found to work to the best advantage.

As has been pointed out, one of the characteristics of cemented tungsten carbide, that tended to retard its commercial usefulness, is the fact that it will not bend or deflect without breaking. In applying a cemented-tungsten-carbide tool on any machine, this characteristic must be kept well in mind and the general condition of the machine studied critically before the tool is run. First of all, the tool set-up should be as rigid as possible. The machine should be in good mechanical condition and any looseness that might produce vibration or chatter be eliminated. This is important, as any chatter or vibration will be found particularly detrimental in the successful life of a cemented-tungsten-carbide tool.

Cemented tungsten carbide can be used on a large percentage of the machine equipment now in use provided these machines are in, or put in, the best possible mechanical condition. Any expense incurred in doing so will find justification in the good performance of cemented tungsten carbide. This necessity for rigidity in tool set-up and machine, when using cemented tungsten carbide, quickly indicated the need for improved machine design. Today, much of the machine equipment sold has been designed with a great deal of thought given to this subject, so that cemented tungsten carbide may be used more efficiently.

When the operating conditions surrounding a cemented-tungsten-carbide application are ideal, or as nearly so as it is possible to make them, there are certain other factors which must be considered, if the greatest tool efficiency is to be obtained.

First of all it is well not to attempt to run a machine at the highest speed possible, despite the fact that one of the primary advantages when using cemented tungsten carbide is increased machine speed right from the start. It is better to start at a lower machine speed and accelerate it gradually as, in this way, it will be easier to discover the critical speed at which the cemented-tungsten-carbide tool should be run on that particular application. Following such a routine will eliminate a lot of grief.

A second thing that should be well considered is the actual tool set-up. It should be as rigid as possible and the overhang of the tool kept at a minimum. Excessive overhang tends to develop chatter and vibration, two conditions which almost invariably defeat any advantages which may be expected when the tool is working under normal conditions.

Like everything else, cemented tungsten carbide will respond to attention and will repay the cost of such time and trouble in longer life between successive grinds. Experience in many plants has proved the value of not running a tool to death, but instead, stopping it while it is still in fair shape and honing it up. By so doing the time charge of each successive grind is decreased, and the initial cost of a cemented-tungsten-carbide tool is more easily proved out in the greater number of units machined during the complete life of the tool.

Failure to comply with this necessity of rigidity is one of the greatest causes for tool failure with cemented-tungsten-carbide tools. It is bromidic to point out that tool failures are the cause of much expense, both in actual dollars and wasted time.

Grinding a cemented-tungsten-carbide tool is a simple matter that enters very largely into the efficiency of tools tipped with this material. The grinding machine used must be free from vibration, and both hand grinding and machine grinding can be used with satisfactory results, though machine grinding is preferable since the proper angles can be held more accurately. When hand grinding, the following should be considered: (1) The use of rest and angle fixtures on pedestal-type grinders is strongly recommended, and the tool should be constantly in motion back and forth. This minimizes glazing and keeps the wheel true.

When grinding a cemented-tungsten-carbide tool in an auto-

matic machine, such as a surface grinder, the down feed should not exceed 0.003 in. in any case. The proper type of wheel to be used depends on the grade of the cemented tungsten carbide, the stock to be removed, and the finish required. Best wheel performance will probably be obtained at about 5000 ft. per min. surface speed.

The tools should not be forced against the wheel, but light pressures used, and the tool moved transversely across the cutting face of the wheel. The wheel should be dressed frequently to prevent glazing. A glazed wheel causes local overheating and checks the cemented tungsten carbide. It can be ground wet or dry. If a coolant is used it should be a heavy flow.

The adaptation and use of cemented tungsten carbide by many production plants, both large and small, throughout the country since its introduction as a cutting medium, has shown that there are very decided economies to be effected with this material. These include: (1) increased machine speed, (2) longer life per grind, (3) reduced tool maintenance costs, (4) less material wasted in setting-up, and (5) greater accuracy throughout a long run. The results obtained have also indicated from what sources these savings come.

Discussion

Coleman Sellers, 3d.[2] The author of this paper failed to mention one advantage of the tungsten-carbide tool which is sometimes valuable. That is the ability to cut hard cast iron and bronze. Material that for one reason or other comes from the foundry with hard spots, hard scale, or sand burnt into the surface may be machined where high-speed steel tools will fail. The importance of grinding tungsten-carbide tools for successful application should be emphasized. In the first place it is necessary to produce the correct angles with the clearance at a minimum for the particular material and work in hand. It is also important to produce a keen cutting edge free from nicks or grooves. The grinding should not be left to the individual operators. This practice is not conducive to good results and is very wasteful. One man should be trained to handle the grinding of the material and all tools brought to him.

Machine grinding is preferable for the following reasons: (1) It is faster as the tool is held more firmly, (2) it is more accurate as to angles of rake and clearance, (3) standards are established and maintained, (4) better work may be produced, (5) water may be used in sufficient quantity to speed up grinding without injury to the tool, (6) less skill is required on the part of the operator, (7) life of tool is lengthened by eliminating waste of tungsten carbide, and (8) labor costs of grinding are minimized.

Of the machines for grinding tools, those grinding on the periphery are preferable as a line contact cuts more freely. It is also less apt to glaze the wheel or overheat the tool so as to injure the tungsten carbide.

A further advantage results in the ability to place a stream of water at the line of contact where with a cup wheel with surface contact the water may only be placed around the edge of the contact, and the center will tend to overheat.

Lapping is not essential with machine-ground tools, at least on cast iron, as has been proved by a number of recent tests on both rough and finish cuts, using ground and lapped tools versus machine-ground tools.

Last spring the Sub-Committee on Tungsten Carbide of the Special Research Committee on the Art of Cutting Metals of the A.S.M.E. sent out a number of questionnaires to possible users of this new cutting material. A report on the answers received was made at the Detroit meeting in June. The writer has analyzed these in somewhat further detail in regard to grinding, and the answers are interesting in the light of being actual experiences of leading industrials in this country.

Many of the firms replying to the questionnaire had so little experience with this material that their answers were too brief to be conclusive, and in the summary given herewith the answers are not included where their experience was limited to one or two tools.

Of 38 successful shops, 13 used machine-ground tools, 16 used hand-ground tools, and 9 did not mention the method, but 3 of them had the tools ground outside. Eight shops reported failure or very doubtful success and all used hand-ground tools. Two others reporting failure did not mention the method of grinding.

The clearance angles varied anywhere from 2 to 29 deg. Six degrees was the most popular clearance angle, probably because it is the standard of many people for high-speed steel. Those who used clearance angles above 7 deg. were successful only on non-ferrous metals such as brass, monel metal, and aluminum alloys. Two reported satisfactory results with 7 deg. on cast iron. All the remaining were 6 deg. or under. Some failures could be traced directly to incorrect clearance angles. For instance one company reported failure of a cutting-off tool. It stated this tool was ground to 25 deg. clearance angle and used on cast iron.

Results on lapping did not indicate that it is essential. Nine companies reported that they lapped tools; three of them on finishing tools only. Of these nine companies, six reported success, two failure, and one doubtful success. Thirty-two companies reported they did no lapping, 26 of these were successful, six reported failure. In addition two companies reported success with and without lapping. Thirteen companies used honing after grinding, reporting success. Three of these reported on the grain and grade of stone which varied considerably. These were 180-L, 200-K, and 280-N.

There seems to be a great variation in the time required for grinding tools. The answers ranged from 3 min. to 3 hr. Four reported an average time 5 min. per tool; four reported about double the time for high-speed steel. Seven reported between 10 and 30 min. Only one reported time for grinding a milling cutter. The figure given was 4 to 5 hr. for a 10-in. milling cutter.

The variety of answers to the questions on the grinding, as outlined, indicate a lack of general knowledge of the subject. The importance of the grinding of tungsten-carbide tools has evidently been overlooked in many cases. Several shops reported they did not note what clearance angles they were using, showing they paid little attention to this feature.

It is obvious that a better understanding of the grinding of tungsten carbide would aid in its more generally successful use.

Roger D. Prosser.[3] The remarks of the author regarding the history and use of cemented tungsten carbide are very interesting and instructive. The important facts in the process of manufacturing cemented tungsten carbide should be understood by users of the material, as the great difference between the methods of manufacture of cemented tungsten carbide and high-speed steel accounts for the difference in their characteristics, and the consequent necessity of different methods of application.

The production of cemented tungsten carbide by the method briefly explained by the author is not as simple as the short discussion of the process might lead one to believe. The utmost care must be exercised throughout the manufacture, and the

[2] Supt. Engineering Dept., William Sellers and Co., Inc., Philadelphia, Pa. Mem. A.S.M.E.

[3] Thomas Prosser & Son, American representatives of Fried. Krupp A. G. of Essen, Germany, and importers of Krupp Widia cemented tungsten carbide, New York, N. Y. Mem. A.S.M.E.

refining of the tungsten, the state of pulverization, ball milling of the ingredients, temperatures and time of sintering, and other factors greatly affect the final product. Through careful regulation of the various factors involved, great improvements are constantly being made in the quality and usefulness of cemented tungsten carbide.

Measurement of the hardness of cemented tungsten carbide on a Rockwell machine does not give an accurate indication of the resistance to abrasion or cutting properties of the material. The resistance of the material to indentation, as shown by the Rockwell test, indicates partly the hardness of the tungsten carbide and partly the character of the binder.

The Rockwell test does give some indication of the relative hardness of various pieces, but at this time the only accurate method of determining the performance is by an actual cutting test. It should also be remembered that cemented tungsten carbides produced by different methods of manufacture and having the same Rockwell hardness reading have shown quite different results in performance.

Extended experimentation has been conducted to determine the best method of attaching the tips to the shanks, and the copper braze has been adopted by Krupp as standard from the time that cemented tungsten carbide was originally placed on the market about four years ago. If the braze is properly made, it is seldom that any difficulty is experienced with the tips coming away from the shank, even when taking exceedingly heavy cuts, provided that a tip sufficiently large to carry away the heat is used on a shank of suitable size to withstand the pressure. As the author stated, cemented tungsten carbide will not bend, and any bending of the shank almost inevitably results in breakage of the tungsten-carbide tip.

The paper emphasizes the desirability of maintaining a smaller clearance angle to support the tip. It might be mentioned that a reduction of the clearance angle from 6 deg. or 8 deg. down to 3 deg. or 4 deg. has in some cases resulted in doubling the length of time between grinds.

Automatic grinding in a rigid holder results in considerable pressure on the work, with consequent danger of overheating and resultant grinding cracks. It is doubtful whether a down feed of over 0.005 in. can be used with safety, especially if one of the harder grades of tungsten carbide is being ground. Light down feeds and less rigid holders should be used wherever possible.

Most of the troubles which were experienced with cemented-tungsten-carbide tools at first have been eliminated. Overenthusiastic salesmen as well as users have learned that these tools are not a cure-all. They have become accustomed to the proper method of application, and grinding practice has greatly improved. The workmen understand that the tools must be properly handled. The tools are employed not only in the most difficult places, but also are being used more and more on the production jobs where real savings can be effected. The history of high-speed steel seems to be repeating itself, and cemented tungsten carbide is finding its place in industry.

FRANK W. CURTIS.[4] Perhaps less has been accomplished with tungsten carbide as applied to milling than in any other form of metal removal where cutting tools are used. Even in Europe, where tungsten carbide was first developed, very little has been done with milling cutters. There are several milling applications in this country, of course, but these are few and far between as compared to turning and boring tools.

The present high cost of milling cutters equipped with tungsten carbide has a tendency to hinder progress. There is no question that satisfactory returns will more than offset the initial cost involved, but then again there seems to be no assurance or guarantee of satisfactory performance.

Experience with tungsten-carbide milling has shown that there is a decided advantage in having milling machines basically designed for this new metal. The machine must have ample rigidity and strength to enable metal removal at faster rates, including of course high spindle speeds suited to tungsten-carbide speeds. Next, the cutter must embody a design that affords the correct application for tungsten carbide. Here, again, it is a case of rigidity and strength with the addition of proper cutting angles.

There are two types of return that can be gained with tungsten-carbide milling cutters. One offers a small profit, the other a more generous profit. Existing machines can only be expected to offer smaller returns, whereas the newly designed machines, built for tungsten-carbide application, will enable much larger profits. There are many milling machines in use today that are not suited for tungsten carbide and prospective users should know where to draw the line.

When experiments with tungsten carbide were first started the first cutter, a 12-in. face mill with 32 blades, did not prove successful due to brazing failure of the tungsten-carbide tips. Several of the tips broke loose after very little use before any comparative results were obtained. Unfortunately, cutters having many teeth are likely to be ruined when one tip breaks, but the same is true with other cutting metals.

It was realized that every effort was being made to improve brazing and that one failure would not justify withdrawal of further tests, so an 8-in. cutter of similar design was made using 20 tungsten-carbide blades. Results were bettered considerably, and several experiments were conducted that taught the limitations of present cutter design when applied for use with tungsten carbide.

In the first place, when this cutter was stopped in the cut—that is, the spindle was stopped with the feed engaged—all the blades that were engaged in the cut invariably broke, if the cut was of medium or heavy nature. The tungsten-carbide tips seemed to fracture or break in two, but the braze did not let go. This offered another problem and led to further experiments with cutter-body design. What was evidently needed was greater support and more rigidity.

The latest cutters were made by inserting the blades almost flush with the face of the body and cutting chip clearances in the body proper. This design has been most successful and is probably the nearest approach to a foolproof cutter. Several cutters made this way have undergone severe tests and no failures have been experienced. The blades received proper support directly in back of their cutting edges, and there was no chance for them to spring under pressure of the cut, since there was practically no overhang.

As to the cutting angles, the writer is not ready at this time to announce the extent of experiments, although progress has been made and it is believed that blades set at a slight negative angle with the axis of the cutter body will serve for general-purpose work with an assurance of a satisfactory finish.

In line with experiments on tungsten carbide, a milling machine with spindle speeds up to 1000 r.p.m. has been developed and built, basically designed for high-production milling, using tungsten-carbide milling cutters. Many recent tests have been conducted on this machine and the results have shown that increased machine stiffness and strength are highly essential if tungsten carbide is to operate at its maximum efficiency.

So far most of the tests have been with cast iron and aluminum, although a cutter is being made that will be used for milling steel. On cast iron the average peripheral speed suited for general-purpose work is 275 to 300 ft. per min. The feed per tooth per

[4] Research Engineer, Kearney & Trecker Corp., Milwaukee, Wis.

revolution will vary from 0.007 to 0.012 in., with 0.010 to be preferred from a standpoint of power consumption and assurance of a free-cutting action. On aluminum it is possible to operate tungsten-carbide cutters around 1500 ft. per min. with as much feed as the machine will stand. At these higher speeds the feed per tooth per revolution will be less than specified for cast iron, although the feed per minute will be greatly increased. Hence, it is more a matter of machine power. Cuts have been taken in aluminum $^1/_{16}$ to $^1/_8$ in. deep with a 5-in. inserted-blade tungsten-carbide cutter operated at 1300 ft. per min., using 240-in. per min. feed to prove the cutting ability of tungsten carbide. Such cuts are not impractical nor impossible, although the machine must be designed primarily for high-speed work of this kind.

Malleable iron will machine similar to cast iron at about the same speeds and feeds. Bronze and copper are also machinable with tungsten carbide, although not many experiments have been conducted with these metals. With the cuts taken, regardless of the material, a very fine, smooth finish has been possible, which leads to the belief that finishing cuts can be taken with tungsten-carbide cutters, which is somewhat contrary to earlier beliefs.

Nothing has been done with solid-type cutters because it is the experience that inserted teeth have many advantages not offered in solid-type cutters. When a tip of a solid cutter breaks, the repair requires complete rebuilding and regrinding. With inserted-blade cutters, however, single teeth can be replaced without difficulty and without necessarily having to take out the remaining teeth.

It has been found that it is much better to regrind tungsten-carbide cutters before they become too dull. There are two reasons for this: First, dull cutters not only tear the metal they cut, but there is a greater chance of breaking the tungsten-carbide inserts if the edges do not cut freely. Second, there is more waste in grinding cutters that have become extremely dull, and likewise the cost of regrinding is somewhat higher.

Above all, it is essential to grind tungsten-carbide cutters with the greatest of care. Before application, the teeth of a cutter should be checked in a fixture to determine their concentricity and squareness. For face mills, it is found that the teeth should be concentric within 0.001 in. for finishing purposes and 0.002 in. for roughing cuts. The faces should be square within 0.0005 in. for finishing and 0.001 in. for roughing. These limits can be easily maintained if an amplifying indicator is used while grinding.

There is need for cutter grinding equipment suited to tungsten-carbide cutters because the present machines do not seem to offer enough stiffness to overcome deflection when the wheel passes over the tungsten-carbide tips. Many minor improvements are also needed to enable grinding with ease and at less time than is required at present.

Summing up this experience, there is no doubt that tungsten carbide has a very valuable and definite use in connection with milling, although the outstanding drawback has been the absence of machines embodying features so necessary to assure satisfactory results, which to some extent has retarded broader progress. It is believed that most failures have resulted from an attempt to follow high-speed steel and Stellite cutter design rather than to develop cutters to suit tungsten carbide. With a rigid machine and a well-built cutter, the possibilities of tungsten carbide for milling are unusually broad.

James B. Giern.[5] A cylinder-department foreman once remarked: "I would like to see the contrivance which could machine a cylinder, round, straight, and finished." Such a contrivance is here. It is the tungsten-carbide tool, provided it is used in the proper combination. Its combination is a rigid and expanding boring-bar.

Boring-bars in general may easily be said to have been somewhat "step-children" to modern production, yet they have found quite a market on account of their rather cheap upkeep and their faculty for boring round. However, a special situation has been created, by which the boring-bar will rank in the first class, by the advent of tungsten carbide. For with that as a material for cutters, the other two conditions may easily be met, and this completes the picture as to efficiency, such as is demanded.

In cast-iron boring, the boring naturally is in the two main stages of roughing and finishing. By the use of fixed centers such as bushings, preferably rotating bushings, and shallow feed, say 0.015 in. per revolution, and a total of 250 ft. per min. surface speed, or thereabouts, the roughing operation may be carried on and the result is a round hole, created under no side-thrust and no heat worth mentioning; consequently, a good preparatory operation.

That is one of the cardinal points in machining—viz., the better the preparatory operations, the better the finish. As to the finishing operation, semi-finishing may occasionally be omitted altogether by observing the foregoing. Fixed centers are important and the foregoing operations should not be depended on to create a center, as it were. Finishing is quite successful at a surface speed ranging from 250 to 300 ft. per min.

Grinding. Regarding the grinding of tungsten-carbide tools, much has been written and said concerning it. As a rule it may be stated that about half the cutting angle usually in vogue may be employed with success. The reason is very simple, for the ordinary tools soon wear off the extreme cutting edge, or perhaps crumble, the result being that it crushes its chips, rather than cuts them, whereas tungsten-carbide tools retain their keenness, so that it is unnecessary to allow for this condition. Also, this is the reason for their cool cut. Radial back-off should prove very successful and is recommended for experimentation along this line.

In the practical use of tungsten-carbide tools, not enough emphasis can be laid on the importance of rigidity. As far as possible jig-tools should be self-contained, so as to avoid any machine inaccuracy, or wear, influencing the boring.

Data and Statistics. Of statistics, there are not many. Yet these few are impressive:

Shock-absorbers: Have been bored at the rate of 2 every 22 sec.
Cast-iron pistons: Cross-bore 40,000 between grinds
Thin cast-iron carburetor parts: 120,000 on one grinding
Bronze bushings: With limits of 0.00025 in. 140,000 between grinds
Cast-iron brake drums: 17 in. diameter, 5 in. deep, 5 min. 300 pr. grind, one operation only.

Predictions are but predictions, but the possibilities point toward elimination of thousands of operations; toward unprecedented economy and the use of hitherto unheard of materials in future machine building.

C. J. Schipplock.[6] The writer wishes to call attention to the fact that much must be done before cemented tungsten carbide as a tool material can be profitably employed in a more general sense than is now possible, taking into consideration, of course, the fact that up to this time its use is confined to certain classifications of tools, and that it has not been successfully adapted to the making of threading tools and units of like character.

Taking the respective items in the order in which they are

[5] Western Tool and Manufacturing Co., Detroit, Mich.

[6] Engineer of Tests, Crane Co., Chicago, Ill.

mentioned in the paper, attention should be brought to the fact that not only can cemented tungsten carbide, when possible of correct application, be expected to equal the performance of diamond tools, but in many instances it will surpass them in economy and efficiency. One instance of this is a turning operation on brass castings that required an appreciable amount of stock to be removed in one roughing cut, with a return cut of lesser feed and depth. A first-class finish must be provided as well as a fairly high degree of accuracy; a definite taper being involved, and the cut being of intermittent nature. On this operation, with an increase in cutting speed to compensate for reduced feed in comparison with carbon tool-steel operation, to more closely suit the requirements of tungsten carbide, each tool of course being operated on a basis best suited to permit of maximum performance, comparison between carbon tool steel, diamond insert, costing $85, and a tungsten-carbide tipped tool with 2.5-gram tip summed up as follows:

Carbon tool steel—75 pieces per grind or an average life of 40 min.

Diamond insert—700 pieces per grind or approximately 6-hr. operation

Tungsten-carbide tip—Count discontinued as five weeks' operation was maintained before it was deemed necessary to grind.

It must be remembered in this instance that the machine equipment involved was specially designed to perform the particular operation and was, with slight change and expense, adapted particularly to the requirements of the tungsten-carbide tool in so far as feeds and speeds were concerned, and consequently permitted the tool to provide maximum performance. It might be further stated that in the hands of a careful operator this same tool continued to give service on the above basis from January to the middle of August, cutting red brass every working day of that period, the working limit per grind, however, being reduced to one week, when a very light dressing-up put the tool back into condition for the next week's performance.

Before going to another phase of this matter, and in order to continue along the lines of discussing each item in the respective order, mention is made by the author that other methods of attaching tips have been generally discarded in favor of the copper braze, and although he does not specifically include the use of brass brazes, in the opinion of the writer this latter material is superior to copper, as experience with several hundred of each type has resulted in loosening of an appreciable percentage where copper was depended upon and as yet the first occurrence is to be experienced where brass was used. This not only covers tools tipped under the supervision of the writer, but in the greatest number of instances where tools purchased from outside sources were involved. The only reason that can be given offhand, no particular research having been attempted as yet along these lines, is that the brass better resists softening where high frictional temperatures are developed in the tip, and must be radiated through the brazing medium.

Returning to the application and operation of tungsten-carbide tools and particularly to the running of these tools at higher speeds, it is needless to say that this is an incontrovertible requirement, as not only does the nature of the material demand operation along the lines of light feeds and consequent high speeds to compensate, but continuous production, especially under present-day machine-tool conditions makes it hazardous, so far as tool survival is concerned, to contemplate feeds greater than 0.030 in. under any circumstances, even though shank sizes up to $1^1/_2$ in. square are possible and front clearances approximating 4 deg. are accurately maintained. Feeds of 0.015 in. to 0.020 in. provide best practice with cuts in proportion to tool design and machine abilities. Exceptions to these figures, which are purposely rather definitely given, can be made, and the writer is frank to state, have been successfully practiced in both directions on experimental work but, as already mentioned, where continuous production is concerned with operators or machine hands of average ability, and machine equipment in good condition and of the character to be found in the average well-equipped shop today, if costly losses are not desirable, or as is more often the case decidedly not permissible, great care must be exercised in keeping feeds down to or somewhat below the above range.

Now if in the past it has been found possible and in fact good practice to utilize feeds of 0.063 in. and upward on ferrous materials with tensiles up to 115,000 lb. per sq. in., utilizing cutting speeds of 35 ft. per min. and upward, say with cobalt high-speed cutting tools, it is evident that spindle and table speeds of lathe and mill-tool equipment will have to be increased from three to five times to permit an even break in cubic removal within a given time where tungsten-carbide tools are used, and the above feed limitations are necessary. Further, if on top of this "even break," advantage is to be taken of the superior speed possibilities of cemented tungsten carbide, these r.p.m. requirements can be multiplied by such factors as will be required to satisfy the demand imposed.

As there apparently is no such effect possible as "burning" tungsten-carbide tools through frictional heat (except for loosening the braze), as is experienced in drawing the temper of steel-cutting tools, the only limits in speed that must be observed are the probability of softening the braze; the speed, rigidity, and horsepower capabilities of machines driving the tools; the ability of the structure being machined to resist distortion through pressure and heat generated; and the possibility of exerting sufficient chucking or clamping power to hold the work without distortion. These are no idle suppositions or mere conjecture, but will all be encountered in some phase or other when utilizing higher speeds, and more particularly where varying or intermittent cuts are concerned. The magnitude of the speeds that may be expected and regularly encountered can be judged from the advertisements of a machine-tool manufacturer, who without doubt has found extended research and experiment necessary before sponsoring spindle speeds up to 4400 r.p.m. in order that available advantages of this type tool can be realized economically.

The advent of cemented tungsten carbide as a tool material throws industry today, as far as tooling practices are concerned, into an even greater period of transition in machine-tool development than was experienced at the development and adoption of high-speed steel in place of carbon tool steels, and unless some means is discovered or invented to increase the strength and ductility of cemented tungsten carbide this transition will be confined entirely to one line of endeavor, and that will be on the part of the machine-tool builder to attempt furnishing these surface speeds, with ever-increasing difficulty in likewise furnishing proportionately increasing rigidity.

Attempts along this line of solution will also demand greater accuracy in forgings, castings, and such other types of articles as generally require machining, this accuracy having to do more particularly with the elimination of lumps, hard spots, or other causes of uneven cuts, as it is needless to state that where such high speeds are incidental to operation, and defects of the nature mentioned become even the exception, the successful prolonging of the life of a tool that has a low impact value becomes a difficulty, and at present prices, an exceedingly costly gamble in which high stakes are played for, and unless properly guarded, can be easily lost.

The present factor of cost combined with that of fragility in comparison with other tool materials will tend to delay the general use of cemented tungsten carbide for tool purposes

more than the necessity for making appreciable expenditures for new or rebuilt machine equipment to use it properly. The introduction of tantalum carbide, however, promises some relief along these lines as it has been within the experience of the writer to witness corresponding if not somewhat better performances on the part of tantalum carbide than those furnished by cemented tungsten carbide, and at prices considerably below those demanded for the latter.

All that has been said in the foregoing in connection with the machining of ferrous materials applies in a corresponding degree to non-ferrous, with the possible exception that harder grades of tool materials can be utilized with prolonged resistance to abrasive wear or, if preferred, the softer, tougher grades best utilized on high-tensile materials can be applied with a corresponding lessening of breakage.

Nothing that is set forth in the foregoing is intended to disparage the qualities of cemented tungsten carbide as a tool material nor in any way convey the impression that its use should be discouraged; on the contrary, in order that prospective users may appreciate some of the procedures that are vital to the successful culmination of its adoption, and that they do not inadvertently suffer costly losses that might at least temporarily or possible entirely change their original desire to utilize its advantages, endeavor is made to bring these important facts to attention in the belief that ultimately this will speed correct understanding of its proper place in the particular field to which it is adapted rather than delay it through improper use. One thing is certain, the manufacturers of this material are doing their best to educate, and cooperate with, purchasers of these tools but, as in every other human endeavor, a great deal must be learned through direct experience of the individual and, as stated before, due to the great cost of cemented-tungsten carbide this experience is liable to prove extremely expensive education unless, through benefiting by the experience of others, breakage and other failures are kept to the almost irreducible minimum.

The human element in the adaptation of this material is particularly vital of consideration, especially where the operators or machine hands on a piece-work basis have been trained for years in existing practices, and must first be untrained both in the grinding and use of the tool, and re-educated on a new basis. Again must it be stated that this is important where present-day machines are also concerned, for in spite of cajoling, threatening, and disciplining, when an operator feels that he is being limited by light feeds in a certain set-up, and he is using the maximum speeds available, it is generally found to be, at least until a costly tool or two has been damaged, and he learns by his own experience that it just cannot be done, a great temptation to "hook it up a notch or two" on the feed set to approximate more closely the old feed practice, and thereby help out the pay envelope at the temporary expense at least of the firm footing the cost of the tool.

To offset some of the present disadvantages under which cemented tungsten carbide is laboring, there are many other advantageous possibilities opened up through its use, not the least of which is mentioned by the author when he says that certain alloys are released for practical use which could not be commercially machined heretofore. Not only is this a fact, but the technique of cutting is also improved in many ways, one instance of which is the fact that unbelievably highly burnished and accurately finished surfaces can be made in one cutting by reducing the front clearance and maintaining the proper position of the tool in relation to center so that the front of the tool immediately under the cutting edge rubs or burnishes the cut, the highest speed possible being utilized. This procedure is limited of course to certain types of metals or materials that lend themselves by nature to such treatment (brass, for instance) and the tool likewise must be properly prepared by careful polishing or lapping. Should this be attempted with ordinary tools the temperature generated would soon destroy both cutting edge and burnishing surface.

Proper support of cutting edges by correct design and maintenance of this by correct grinding are so important as to warrant special provisions in connection with the latter. Free-hand grinding of tungsten-carbide tools without some adjustable rest, which will provide and indicate correct angles, is nothing more or less than rank extravagance, and should not under any circumstances be permitted. Wheels should also be of the cup type to eliminate "undercutting," especially where small diameters are utilized in dressing large tools, and the proper grade and grain for the particular type of grinding to be done should be made a subject of close study. A 100-H silicon-carbide wheel has been found particularly suited to all-around use on both hard and soft grades of tungsten-carbide tools by the writer. Where any appreciable number of such tools are used in a shop, grinding equipment with suitable fixtures and wheels can be provided for less than $500 per unit; and without fear of contradiction, it may be said that this cost can be saved inside of a reasonably short period of time, especially when a man trained in this work is put in charge of the tools and grinding equipment.

In this connection it must also be stated that men using tools should not attempt to get that last piece off of the tool before grinding, but determine on any given operation by close observation at first what a fair number of pieces per grind should be, and even then set a limit a good 10 per cent to 20 per cent under to allow for a safety margin, as again it is in order to remind that those last few pieces may prove expensive through crushing off the nose of a tool at the rate of $1 per gram.

As can be readily judged from these remarks, this discussion covers mainly the use of the rugged class of turning and cutting tools. The more costly types of profile and form-cutting tools, with which some experience has been gained, where the generating of accurate outlines increases the cost due to labor in the making of the tool, must be carefully approached from an engineering standpoint and equally as carefully used. Where this has been found possible of accomplishment unbelievable results as regards performances have been obtained, and to date special machinery, in which ordinary tooling materials will not survive long enough to warrant setting up, has been designed and put into successful production.

W. S. Huson.[7] There still seems to be some misapprehension as to the structure and treatment of cemented tungsten carbide. There is a feeling that it is an alloy similar to tool steel and partakes of features pertaining to steel. There is not, however, the coalescence of the elements typical of steels; it is just what its name implies—a cemented material in which cobalt is the bond, holding the grains of hard tungsten carbide together. The success obtained in cutting tools and the continuing laboratory work and investigations lead to the hope that out of it all there will come a fused alloy lacking the present brittleness, and having the good characteristics of high-speed steel, but much harder and better adapted for formed tools and the like. As to the structure and proportion of component parts there is an excellent paper by Wyman and Kelley of Schenectady, N. Y., given at the Chicago meeting of the A.I.M.E., September, 1930, which goes deeply into the laboratory side of the material and shows by micrographs how varying proportions affect the final structure.

It is in some ways unfortunate that the grinding and sharpening of cemented-tungsten-carbide tools come in for so much comment; the idea that they are very susceptible to fracture from heat or undue pressure seems to have gained a strong hold. This

[7] Technical Industrial Engineer, Brooklyn, N. Y. Mem. A.S.M.E.

is not so for, though the material is brittle and can be ruined by improper attempts at grinding, the fact remains that the temper cannot be modified by grinding when properly done. This is the same as for steel tools, except that in the tool steel care must always be taken not to draw the temper of the cutting edge. These tools lend themselves very well to dry grinding as the edge can be watched and the policy of sharpening tools before they are too dull is a good one to follow. They should be held firmly yet gently against a sharp wheel and, while a somewhat longer time is needed than for steel tools, if the same care is taken, the result will be satisfactory. Wet grinding is entirely feasible, and possibly the best sharpening is done on a grinder where the tool can be clamped, and the clearance angles preserved. However, with minor fixtures at hand, this can be done by hand sharpening dry or wet, and the plan of having one man do all the grinding of these tools has much to commend it. Good wheels of suitable grit run at proper speed in a rigidly set-up grinder and reasonable care in not trying to force grinding because the material is hard will make the work as commonplace as for any tools. The use of fine-grit wheels for giving a keener edge and honing for still smoother edges all help, but due to the peculiar structure of the material honing is not quite as effective as with steel tools.

Cemented tungsten carbide is in the process of further development. Wherever it can be applied at present within its limitations it is a profitable investment, and the outstanding hope is that as experience broadens, the peculiar hardness of the tungsten carbide will be adapted to much greater efficiency not only in edged tools, but in dies and the like, where extreme hardness is desirable.

Author's Closure

The author is thoroughly in sympathy with the discussion as brought forth by Coleman Sellers, 3d. Much of his discussion had to do with grinding tungsten-carbide tools. Proper grinding is the greatest single factor in their success.

In regards to the discussion by Roger D. Prosser, he emphasizes the desirability of maintaining smaller clearance angles to support the tip. He states that reducing the clearance angle from 6 to 8 deg. down to 3 or 4 deg. has resulted in doubling the length of life between grinds. This statement is particularly valuable in that it greatly diminishes the cost of using tungsten carbide.

With reference to the discussion of F. W. Curtis, we have a great number of milling cutters in actual production to date, and in many cases we have increased the production by 50 per cent and have increased tool life in the ratio of from 10 to 1 to 30 to 1 over high-speed steel and Stellite cutters. It is believed he emphasizes too strongly the need of new machine-tool equipment. The mentioned results are being obtained on machines 5, 10, and 15 years old. Of course these machines often have to be put in good repair.

With reference to the discussion of C. J. Shipplock, the author wishes to take exception to this discussion in regards to the feeds, in which he claims tungsten carbide is limited to actual operation. We consider a feed of 0.060 in. per revolution a medium feed for tungsten-carbide tools, providing the shank is in proportion. The majority of tungsten-carbide tools in use with shanks $1^1/_2$ inch square can readily use a feed of 0.060 in. per revolution. With shanks 2 by 2 and 2 by $2^1/_2$ and 2 by 3, it is not at all unusual to take feeds ranging from $^1/_8$ to $^1/_{16}$ and with depths of cut as high as 1 in. We have changed our ideas somewhat in regards to using the light feeds and high speeds. We prefer in most cases to use normal feed and increase the speed normally. This discussion has put undue emphasis on the subject of using fine feeds and high speeds, and in a great many cases this practice is causing much more injury to tungsten-carbide tools than by using heavier feeds and less speed. Naturally in approaching the foregoing condition we feel that thought should be given to the design of tool overhang and machine rigidity. The improved tool design has been successful in reducing such breakage as has happened in the past. The most severe applications can be successful with the proper tool design

OGP-53-1

The Lorenzen Gas Turbine and Supercharger for Gasoline and Diesel Engines

Particulars Regarding the Design and Construction of a Gas Turbine Having a Rotor With Hollow Blades Which Are Cooled by Air Circulated Through Them Under Pressure, Together With a Discussion of Present and Possible Applications

By CHRISTIAN LORENZEN,[1] BERLIN, GERMANY



THE gas turbine works upon the same principle as the internal-combustion piston engine: after precompression of the working mixture (in the gas turbine the air and fuel are compressed separately) ignition takes place, and in the expansion that follows a part of the heat is converted into mechanical work. In the piston engine, however, work is done as the expansion progresses to the end of a stroke, whereas in the turbine the expansion of the gases and the resulting work are continuous.

The hopes held out by the gas turbine are based upon the possibility it affords of complete expansion of the gases, which is impossible in the piston engine owing to the limitations of cylinder dimensions; further, the performance capacity of a piston engine is much restricted by reason of the difficulties encountered with increased dimensions.

Just as in the piston engine, with regard to the heat supply we have to distinguish between—

a Heat supply under constant volume. The combustion takes place as an explosion under increasing pressure. The Holzwarth gas turbine works according to this method

b Heat supply under constant pressure. The combustion takes place without an increase in pressure, as the gases can expand accordingly. The Lorenzen turbine is an example of this method.

Method *a* as compared with method *b* has two drawbacks: First, it is necessary to employ regulating or timing devices in the hot combustion chamber. Second, there is a considerably higher shock and eddy loss at the regulating valves and at the entrance edge of the turbine blades owing to the great variation of velocity in the gases. On the other hand, the thermodynamic efficiency of the explosion turbine is superior to that of the constant-pressure turbine.

In the piston engine and in general also in the gas turbine the working gases are generated inside the machine itself—in the combustion chamber in the case of the gas turbine. The turbine, however, can also work in combination with a piston engine, utilizing the energy in the exhaust gases of the latter. In recent years various turbines have been operated according to this method, notable among them being the Lorenzen exhaust-gas turbine described later in this paper.

Compared with a steam turbine, which generally has many stages, the gas turbine has but one or two working stages, owing to difficulties encountered in the materials from which it is constructed. It then operates on the constant-pressure principle, similar to the original De Laval turbine, and it is obvious that for this reason the gas turbine is at a disadvantage compared with the steam turbine. However, this difficulty will probably be overcome in time, but in comparing the two types of turbine it may be said that the exhaust from the gas turbine does not carry away as much waste heat as does that of the steam turbine, in which latter the loss in one pound of steam is 990 B.t.u.

That it has taken such a long time to develop a reliable gas turbine in spite of the many years' experience which we have had in building steam turbines, is to be ascribed to various reasons. Principally, however, as we shall see in a later portion of this paper, it is owing to the enormous stresses, both mechanical and thermal, to which the material is subjected, and to the low efficiency of the turbo-compressor—the only type of compressor that can be considered in connection with a gas turbine.

The vital question in building a gas turbine is to prevent the blades from attaining a red heat, principally at their enter-

[1] Engineer and Managing Director, Lorenzen G.m.b.H., Berlin. Mr. Lorenzen was active in early automobile developments in Germany, and went to England in 1905 for the exploitation of German designs there. When aviation started he began building airplane propellers in England. Before the World War he returned to Germany and engaged in airplane-propeller manufacture there. He participated in the earliest supercharger and adjustable-pitch aircraft-propeller developments. About the time of the Armistice he proposed a combination of adjustable-pitch propeller and exhaust-gas supercharger with automatic controls so that an airplane engine would always be charged at one atmosphere absolute pressure and its propeller pitch always adjusted to the air density where the plane was flying. The Treaty of Versailles forbade Germany to build airplanes, and so Mr. Lorenzen turned to equipping other engines with superchargers and to the development of the gas turbine. Recently he has resumed work with aircraft power plants.

Presented at the National Oil and Gas Power Meeting of the A.S.M.E. Oil and Gas Power Division, State College, Pa., June 12–14, 1930.

ing edges. Before expansion takes place the gases have a temperature of from 1000 to 1200 deg. cent. (1832–2192 deg. fahr.) and a pressure of a few atmospheres in a stationary container. To build the latter sufficiently strong to resist the pressures and temperatures involved is not a matter of great difficulty. After expansion the temperature of the gases drops about 150 to 350 deg. cent. (270–630 deg. fahr.) according to the pressure ratio employed. With a temperature drop of 150 deg. cent. a velocity of 600 meters (1968 ft.) per sec. is obtained, and with one of 350 deg. cent. (662 deg. fahr.) a velocity of about 850 meters (2790 ft.) per sec. The jet of gases now passes through and impels the blades of the fast-running turbine rotor. The peripheral speed of a single-stage rotor should, if possible, be half of the jet velocity. Fortunately, the efficiency curve of a turbine is reasonably flat near its maximum, so that with the highest possible peripheral speed a fairly good efficiency is obtainable. The centrifugal force encountered at such speeds is high even for normal temperatures, and were the turbine blades to be heated to the same temperature as the jet of gases the tensile strength of the blade material would scarcely be able to resist it.

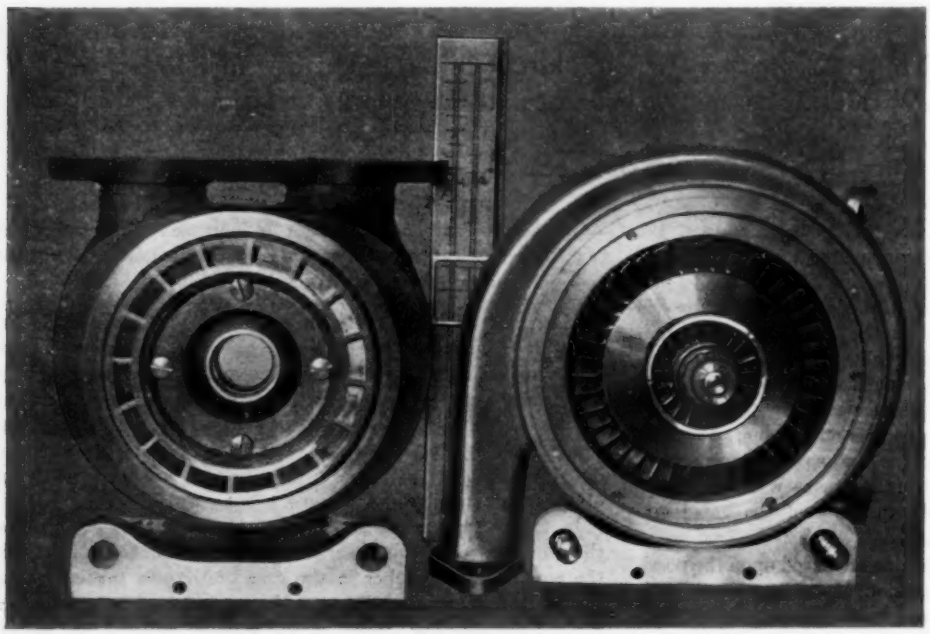


FIG. 1 SHOWING INTERIOR OF THE LORENZEN TURBINE

Water injection has been used effectively to reduce temperatures, but it has the disadvantage that most of the heat is used up in generating steam, leaving but a small amount available for work. Cooling by means of water jackets has the same disadvantage, as may be seen from the working area of the *p-v* diagram. To lower the temperature by further expansion means the application of an exhauster to create a vacuum beyond the turbine, which could only be done at the cost of efficiency. A better way, and the one which has been followed in building the Lorenzen turbine, is to overcome the difficulty by suitable construction.

THE ROTOR OF THE LORENZEN TURBINE

The construction of the rotor of the Lorenzen turbine is the same in a pure or "live" gas turbine—i.e., a turbine with its own combustion chamber and producing its own gas—as in an exhaust-gas turbine driven by the waste gases from a piston engine. Its special characteristics are as follows: [see Figs. 1 and 2]:

1 It is a single-stage impulse turbine acted upon in the axial direction by the working gases
2 The turbine rotor acts at the same time as a compressor wheel
3 The turbine blades are hollow, with very thin walls, and form the continuation of the compressor wheel in such a way that
4 The compressed air serves at the same time to cool the hollow blades internally by flowing in a radial direction from the inner to the outer ends.

The hollow blade which specially characterizes this turbine is the result of long years of search for a good and reliable material and the most suitable shape and form in which to dispose it. The tensile strength is about 100 kg. per sq. mm. (142,000 lb. per sq. in.) and the thickness of the wall about 0.25 mm. or 0.01 in.

That this method of cooling the turbine blades is effective may clearly be seen from the following considerations. The gases flow between the blades with a velocity of about 350 to 450 meters (1150–1475 ft.) per sec. and at a temperature of perhaps 700 deg. cent. (1292 deg. fahr.). Using accepted formulas for calculation, this gives a heat transfer of 750 kg-cal. per sq. m. per hr. per deg. cent. (154 B.t.u. per sq. ft. per hr. per deg. fahr.). To prevent the blades from becoming red hot it is also necessary that the air flowing inside the hollow blade shall have sufficient velocity so that it will carry away the heat as fast as it comes through the blade walls from the gas side. This can be effected by converting the static pressure created in the compressor wheel into velocity. For example, if the peripheral speed of the blades is about 320 meters (1050 ft.) per sec., then an air velocity of 250 meters (820 ft.) per sec. is obtainable which will cause a heat transfer of 572 kg-cal. per sq. m. per hr. per deg. cent. (117 B.t.u. per sq. ft. per hr. per deg. fahr.), and the heat flow through the wall, according to the formula

$$K = \frac{1}{\frac{1}{\alpha_G} + \frac{1}{\alpha_L}}$$

will be 324 kg-cal. per sq. m. per hr. per deg. cent. (66 B.t.u. per sq. ft. per hr. per deg. fahr.). The temperature of the hollow-blade wall will be

$$t_W = t_G - \frac{K}{\alpha_G}(t_G - t_L) = 428 \text{ deg. cent. (802 deg. fahr.)}$$

This temperature is allowable provided suitable material is used, particularly as at the most stressed part of the blade, namely, the root, the wall temperature is considerably lower owing to the reduced heating and to the better cooling obtained with the still cool air at the beginning of expansion.

The Lorenzen Turbine as an Exhaust-Gas-Turbine Supercharger

In an exhaust-gas turbine the impelling gases come from a piston engine and flow to the turbine nozzles. The cooling air leaving the hollow turbine blades is collected in a diffuser or volute chamber surrounding the rotor, where its velocity is converted into pressure, and it can then be used as compressed air for scavenging or supercharging the engine. The gain by this compressed air through utilization of the exhaust gases which otherwise would be wasted increases the economy and makes it possible either to increase the output of the engine or to obtain the same output with smaller cylinders.

The kinetic energy remaining in the exhaust gases leaving the piston engine may easily be determined from the p-v diagram by planimetering the hatched area $ABDEFA$ in Fig. 4, assuming the expansion line AB to be an adiabatic. A is the point at which the exhaust valve opens, and EF indicates the back pressure.

For a Diesel engine with a cylinder bore D of 500 mm. (19.68 in.) and a stroke S of 700 mm. (27.56 in.) running at 155 r.p.m. the indicated horsepower N_i (see Fig. 5) will be 188 hp., the brake horsepower N_e, 150 hp., and the theoretical energy in

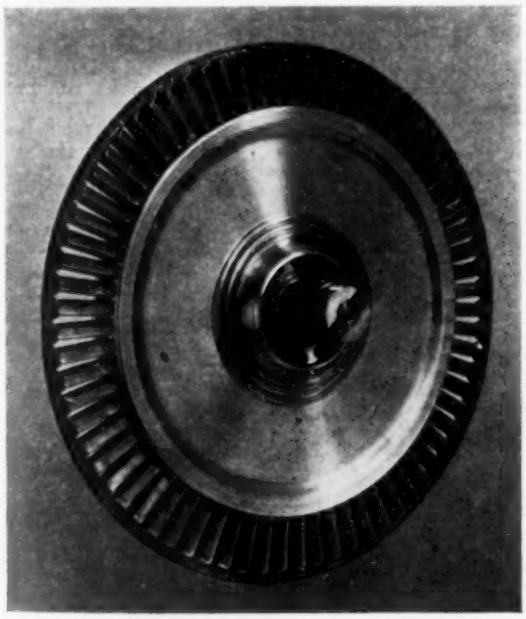


Fig. 2 Rotor of Lorenzen Turbine

the exhaust gases sufficient to generate 32.2 hp. Using this energy to drive the turbine supercharger, the working diagram of the engine will be that shown at II, and the performance of the exhaust gases will be accordingly increased. Diagram II*a* corresponds to a supercharge of $0.9 \times 1.25 = 1.125$ atmos. (16 lb. per sq. in.). In the factor 0.9 the losses of flow from the turbine to the cylinders have been taken into consideration, whereas diagram II*b* corresponds to an overloading of $1.50 \times 0.9 = 1.35$ atmos. (19.2 lb. per sq. in.). The theoretical performance of the exhaust gases gives for II*a*, 54.8 hp. and for II*b*, 77.1 hp. The energy required for the compression, if assumed to be adiabatic, is 4.9 hp. in the case of II*a*, and for II*b*, 10.9 hp., so that the performance capacity of the exhaust gases for II*a* is 11.2 times and for II*b* 7.0 times the energy required for supercharging the engine.

To obtain good performance of exhaust-gas turbine superchargers in connection with multi-cylinder Diesel engines, it is proposed to attach a small turbine to each cylinder, or in smaller-sized Diesel engines one turbine to two cylinders. In the second case the inlet casing of the turbine will be divided into two halves by a wall. This will expose the nozzles to the gases of one cylinder only. The gas jet of high velocity imparts a considerable amount of energy to the turbine rotor, causing a reverse action as soon as the pressure in the cylinders is released. The turbine wheel passing the nozzles rapidly keeps up this flow and ejects the remaining gases from the cylinder which it is scavenging. If several turbines are employed, then the compressed air from the various exhaust-gas superchargers

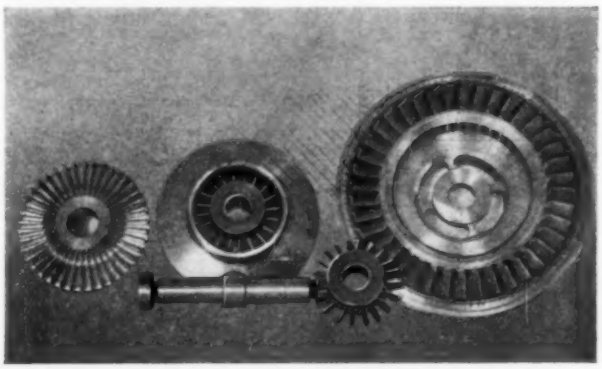


Fig. 3 Component Parts of the Rotor of a Small Exhaust-Gas Turbine

should be led into one container. This increases the reliability of scavenging to a considerable degree, because one turbine getting out of order will not stop the delivery of air: the other turbines will simply deliver more air with a little less pressure.

One great advantage of the turbine supercharger lies in the fact that it can be attached to any existing engine without making any constructional alterations, whereas a mechanically driven supercharger requires a driving shaft with the necessary gears, clutch, etc. In applying the turbine supercharger it is only necessary to alter the run of the piping.

In practice it has been found that the application of a turbine supercharger makes the operation of an engine very flexible. The reason is that with increasing performance of the engine the output of the turbine also increases automatically, although the engine may keep on running at the same speed. This of course is impossible with a mechanically driven compressor unless a speed changer is employed.

One of these turbines was built to run at 30,000 r.p.m. During the tests it was often brought up to 40,000 r.p.m., and at the conclusion of the trials it was run up to the bursting point. At 48,000 r.p.m. measurements could still be made. When it burst it was running at about 50,000 r.p.m., this very nearly agreeing with the theoretical calculations made previously.

In this and also in later tests with larger turbines, temperatures up to 835 deg. cent. (1535 deg. fahr.) were measured ahead of the turbine and 875 deg. cent. (1607 deg. fahr.) back of it. The increase in temperature was due to incomplete combustion in the engine, and through the escape of cooling air this caused after-burning in the turbine. By taking the turbine to pieces after such tests it was found possible to determine by the color of the rotor disks and the roots of the blades that there had been a temperature of about 280 to 300 deg. cent. (536–572 deg. fahr.).

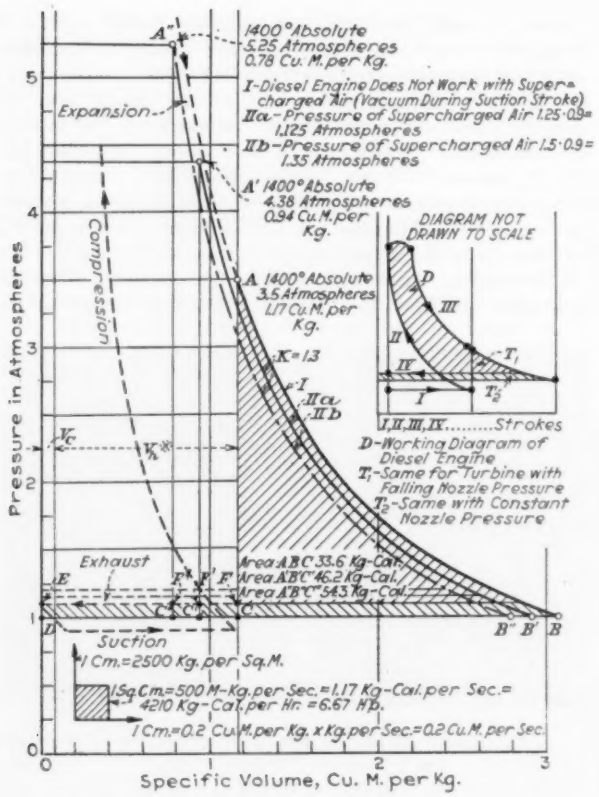


FIG. 4 *p-v* DIAGRAM OF TURBINE UTILIZING EXHAUST FROM A DIESEL ENGINE

THE LORENZEN TURBINE AS A PURE GAS TURBINE

The excellent results obtained with the Lorenzen rotor in exhaust-gas turbines led to its employment in a pure or "live" gas-turbine plant on a much larger scale. It was decided to build a machine according to the constant-pressure principle ("constant pressure" meaning a continuous generation of the working fluid), although the Lorenzen rotor can also be used in the explosion type of turbine. This was done to avoid the difficulties encountered by timing and regulating devices inside a hot combustion chamber. The arrangements employed are shown in Figs. 6 and 7, namely,

(*a*) Turbine and compressor combined into one rotor (Fig. 6), and

(*b*) Turbine and compressor with separate rotors (Fig. 7).

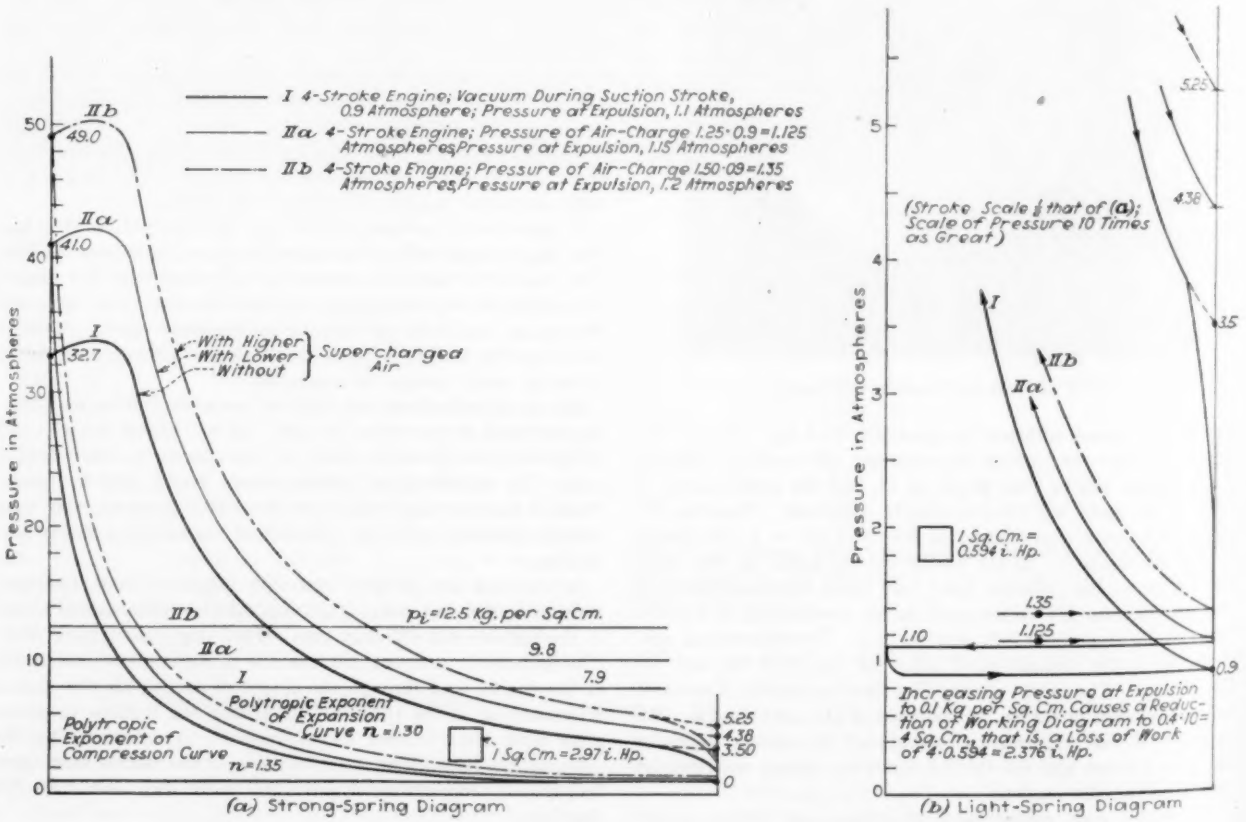
The first arrangement is only applicable for low pressures which can be obtained in one stage, whereas the second must be employed when higher pressures and larger quantities of working fluid are required.

In the first arrangement (Fig. 6) the air enters the rotor

FIG. 5 (*Below*) THEORETICAL INDICATOR DIAGRAMS OF LORENZEN TURBINE UTILIZING EXHAUST FROM A DIESEL ENGINE

(Cylinder bore, 500 mm. (19.68 in.); stroke, 700 mm. (27.56 in.); r.p.m., 155; mean piston speed, 3.62 m. per sec.; area of piston, 1964 sq. cm. (304.4 sq. in.); indicated hp. per cylinder N_i, 188 (case I); mean effective pressure p_i, 7.9 kg. per sq. cm. (112.3 lb. per sq. in.).)

	Case I	Case IIa	Case IIb
Supercharged air, atmos.	..	1.125	1.35
Ratio of air charge	~ 100	125	150
Ratio of oil burned	100	125	150
Ratio of effective output	100	125	150
Mean effective pressure, kg. per sq. cm.	7.2	9.8	12.5
Indicated hp. of one cylinder	188	233	298
Friction loss, hp.	38	45	73
Brake hp. N_b	150	188	225
Mechanical efficiency	0.80	~0.80	~0.76



at *a*, flows through the compressor and the hollow blades, its high velocity then being converted into pressure in a volute chamber or diffuser surrounding the rotor. This compressed air is then led into a heat exchanger of special construction (Fig. 8) which has been in use for about two years in the Berlin municipal gas works and operates exceptionally well because of the continuous internal rotation of the air and gases flowing. After flowing through the heat exchanger the air is led into the combustion chamber, where sufficient fuel is added to obtain the desired temperature. As the combustion chamber is open toward the turbine nozzles, the same constant pressure exists inside it as in the compressor—less, of course, the loss in the heat exchanger. The gases now expand in the nozzles of the turbine and impel the hollow blades of the turbine rotor, after which they pass into the heat exchanger where they give up their remaining heat to the compressed air.

The second arrangement (Fig. 7) differs from the first, a special compressor being employed. The cooling air may then

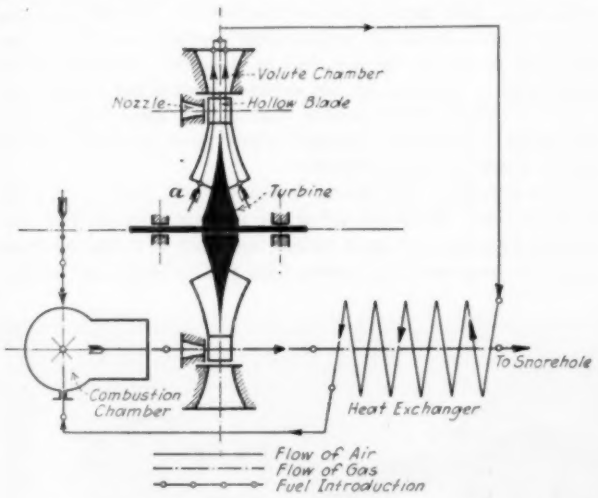


FIG. 6 COMBINED TURBINE AND COMPRESSOR ROTOR

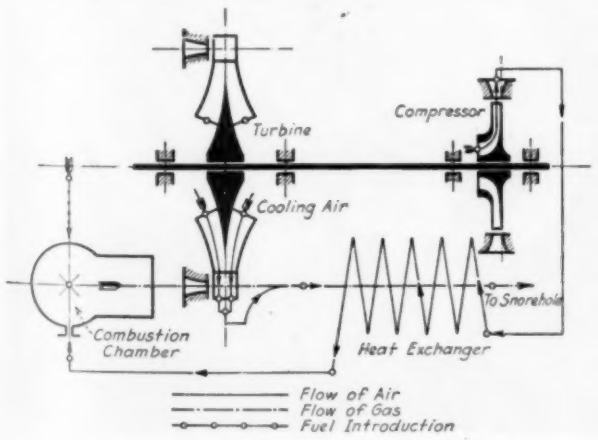


FIG. 7 TURBINE AND COMPRESSOR ROTORS SEPARATE

be used for heating purposes or led under a boiler or the like. The highest efficiency is obtained when the cooling air is applied as described in the first arrangement in a special combustion chamber with lower pressure and its own sectional portion of nozzles.

That it is necessary to utilize the heat remaining in the exhaust when a high efficiency is to be obtained, can easily be seen from the entropy diagram in Fig. 9, which has been drawn to scale for a special example. In this drawing *ABCDA* represents the quantity of heat added through fuel supply, and *AEFGHKA* the turbine performance after deducting the work required for the compressor and the cooling air. The heat re-

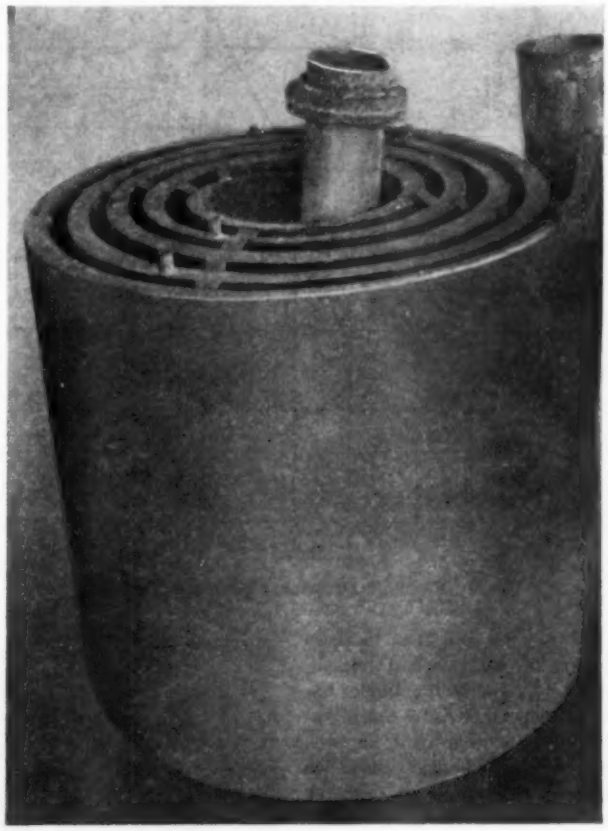


FIG. 8 SPECIALLY DESIGNED HEAT EXCHANGER

maining in the exhaust is shown by the area *LMNOL*. The relation of the area *AEFGHKA* to *ABCDA* gives the thermal efficiency when the heat remaining in the exhaust is not utilized. In order to regain the greatest possible amount of heat from the exhaust for use in the turbine plant, a heat exchanger may be considered. How far this is feasible depends only on the size and the efficiency of the heat exchanger. If, for example, we allow a difference of 100 deg. cent. (180 deg. fahr.) between gas and air and the turbo-compressed air comes from the compressor at 125 deg. cent., we can then easily raise its temperature to 600 deg. cent. (1112 deg. fahr.) in the heat exchanger. In the combustion chamber it is desired to bring this temperature up to 1000 deg. cent. (1832 deg. fahr.), so it is necessary to add fuel according to the area *APQDA*, making the thermal efficiency the ratio of the area *AEFGHKA* to *APQDA*, or $\eta_{th} = \sim 29$ per cent for the example in question.

The curves of Fig. 11 show the relationship between amount of fuel, size of heat exchanger, and thermal efficiency at various pressures and temperatures at the nozzles in a gas-turbine plant of 7500 kw., assuming that the gases impelling the hollow blades have a temperature of 750 deg. cent. (1382 deg. fahr.) and a constant peripheral speed of 320 meters (1050 ft.) per sec. The isothermal efficiency of the turbo-compressor is $\eta_{is} = 72$ per cent, as the loss of pressure in the heat exchanger is assumed to be 0.15 atmos. (2.13 lb. per sq. in.). The plant then shows an

efficiency of about $\eta_{th\,max} = \sim 31$ per cent when the working fluid is at a pressure of $p = 3.5$ atmos. (45.8 lb. per sq. in.). The effect of a rise in temperature is very great—for example, an increase of 100 deg. cent. increases the total efficiency about 5 per cent.

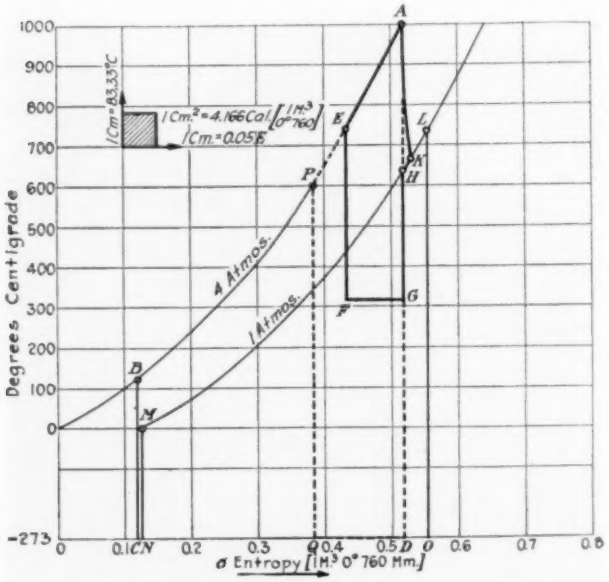


FIG. 9 ENTROPY DIAGRAM FOR A SPECIAL CASE

It may be said that in these curves the utilization of the cooling air by the hollow blades has not been taken into consideration; on the other hand, the area of the heat exchanger, which has been calculated as an ordinary straight-run exchanger, will be considerably reduced by using the spiral form shown in Fig. 8.

Fig. 10 shows a pure (or "live") gas turbine in the course of assembly. At the center is the rotor, its diameter being 990 mm. (38 in.); behind it at the right the inlet casing and the nozzles, both entirely built out from the special Krupp steel NCT3; and at the left the outlet casing, which is welded up from V2A Krupp steel. In addition both casings are enclosed in a sheet jacket, the space between casing and jacket (3 to 4 in. wide) being filled with firebrick ground to about pea size and put under pressure through a pipe connecting with the compressor. This arrangement protects the outer jacket from the combustion heat and the inner casing from pressure.

Fig. 12 shows the entire plant. Regulating devices are necessary only for the intake to the compressor and for the fuel supply, and these are of the same types as those used with other compressors or steam boilers.

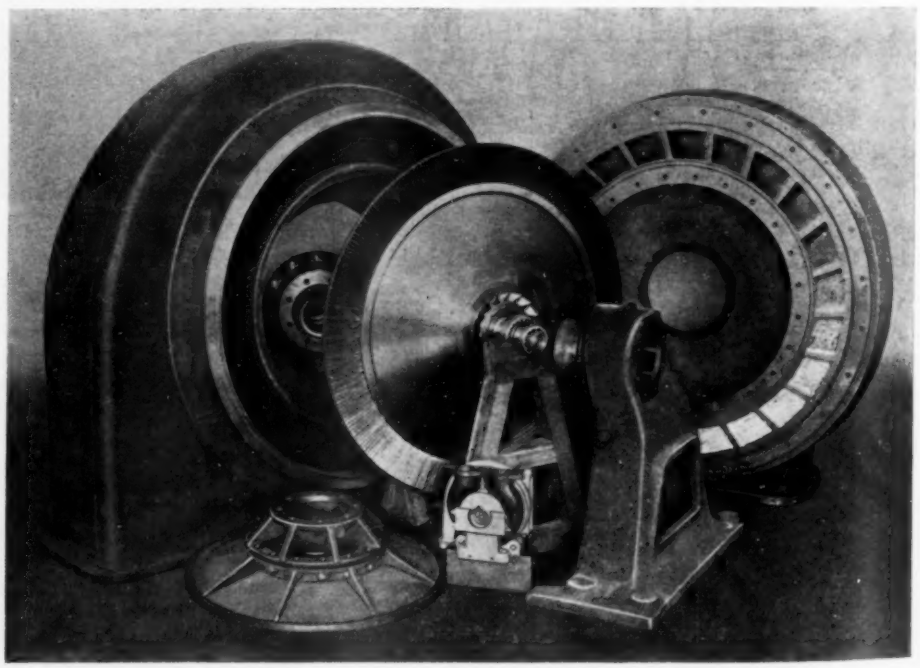


FIG. 10 ASSEMBLY OF PURE GAS TURBINE WITH 38-IN. ROTOR

FUTURE POSSIBILITIES

Another method of utilizing the heat in the exhaust and one which shows the highest efficiency is the following: When the compressed air is not preheated by the exhaust, its temperature must be raised from 125 deg. cent. up to about 1000 deg. cent. (257 to 1832 deg. fahr.) by the fuel supply. Owing to the limitation of the temperature to about 1000 deg. cent. (1832 deg. fahr.) the combustion takes place with about three times as much air as is necessary and the exhaust from the turbine with a temperature of about 700 deg. cent. (1292 deg. fahr.) contains sufficient oxygen to bring its temperature up to 1400 to 1500 deg. cent. (2552–2732 deg. fahr.) by adding fuel. This is high enough to be used under a steam boiler or in melting metals in foundries or the like. If the heat remaining in the exhaust is fully utilized in this way, then the turbine plant will show an efficiency of more than 85 per cent.

The combination of a gas turbine with a steam-turbine plant may be of special interest in municipal electric plants or in ship propulsion, and particularly in battleships, because the gas-turbine plant is ready at any moment to start, and would have sufficient power to set the ship in motion. A steam-turbine plant able to utilize fully the exhaust from a gas turbine will have an output about double that of the gas-turbine plant. The space required for the gas-turbine plant in such a combination would be inconsiderable.

In gas works, coke plants, foundries, etc., there is a great waste of heat. With the aid of the Lorenzen turbine this heat can also be saved to a great extent, inasmuch as it may be transferred to compressed air coming from the turbine, and then employed therein to do mechanical work. The temperature of this waste heat largely ranges from 400 to 600 deg. cent., 750–1100 deg. fahr.). A constant-pressure-turbine plant, however, first begins to work when the temperature of the working fluid exceeds 500 deg. cent. (932 deg. fahr.), and at first glance it would seem that little could be done. But remembering that

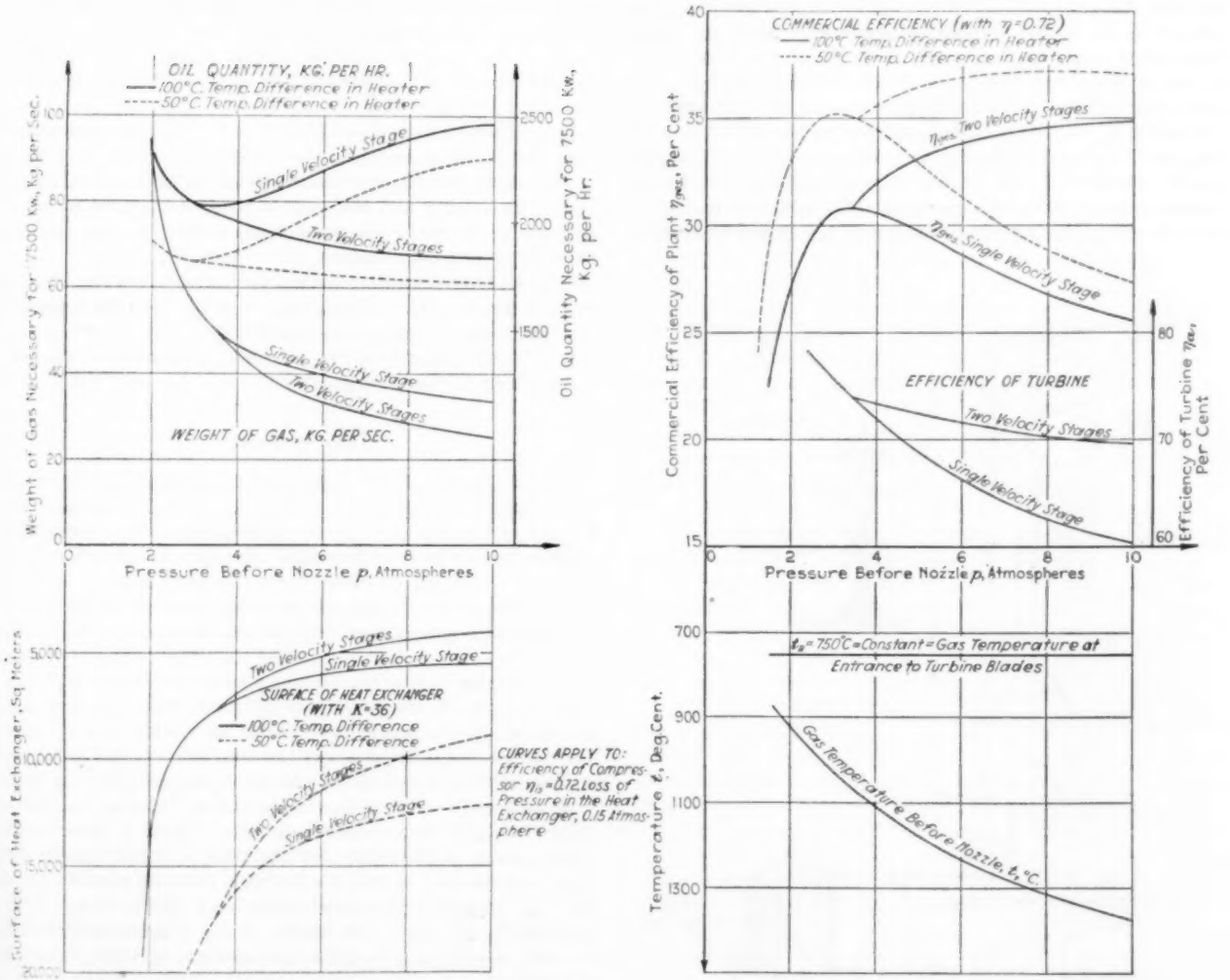


FIG. 11 EFFICIENCY, FUEL REQUIREMENTS, HEAT-EXCHANGER SURFACE, ETC., OF A 7500-KW. GAS-TURBINE PLANT FOR VARIOUS PRESSURES AND TEMPERATURES AT THE NOZZLES

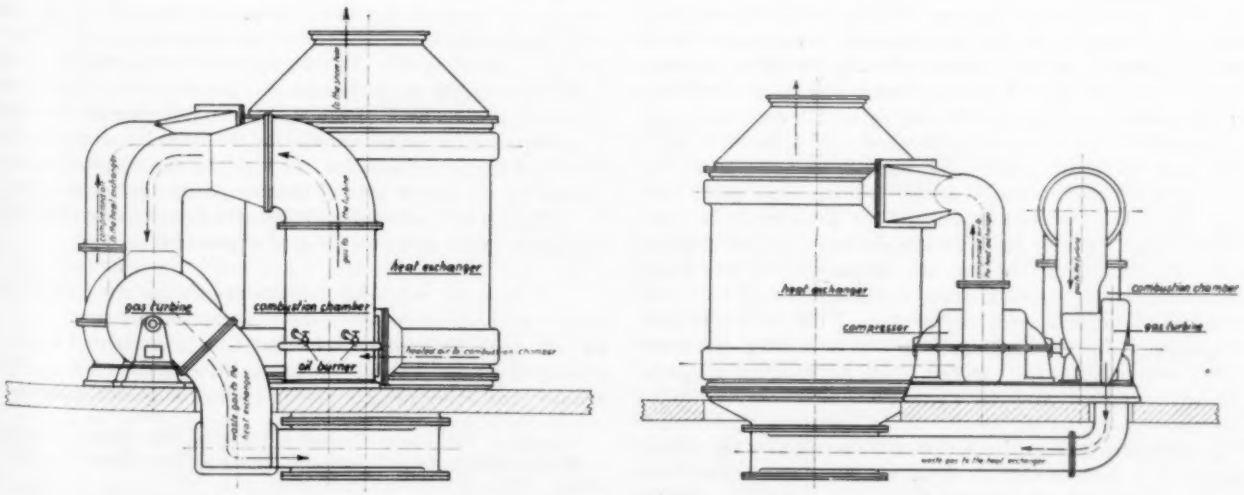


FIG. 12 ARRANGEMENT OF PLANT IN WHICH TURBINE OF FIG. 10 IS INSTALLED

the heat from the exhaust of the turbine is still available, the method shown in Fig. 13 can be employed. Most gas ovens make use of a recuperator in which the secondary air is preheated to about 600 to 800 deg. cent. (1112–1472 deg. fahr.). This preheating is done by the waste gases coming from the retorts. When leaving the retorts these gases are at about 900 to 1000 deg. cent. (1652–1832 deg. fahr.) and are cooled down to 400 to 600 deg. cent. (752–1112 deg. fahr.) in passing through the recuperator. If now these gases are led into a heat exchanger through which compressed air is driven in the opposite direction,

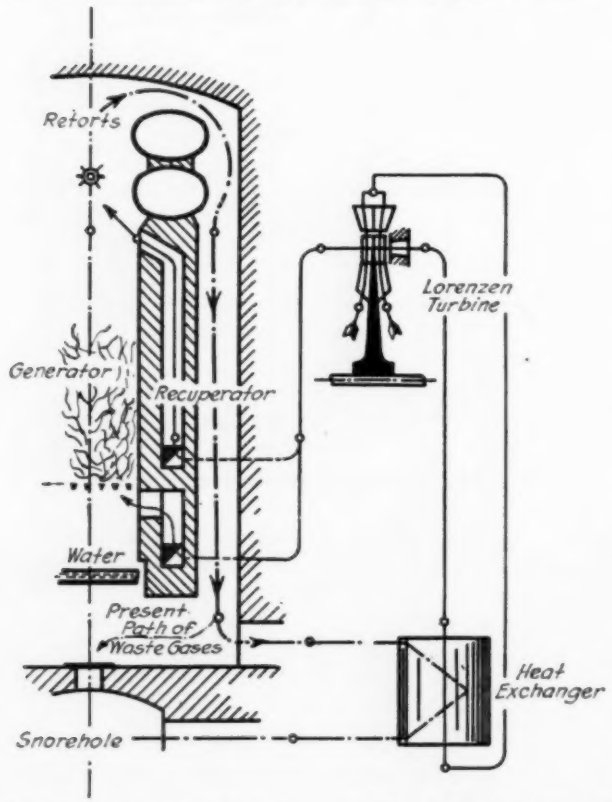


FIG. 13 ARRANGEMENT FOR UTILIZING WASTE GASES FROM GAS PLANTS

then the temperature of the latter will be raised to 300 to 500 deg. cent. (572–932 deg. fahr.) on their way to the nozzles of the turbine. Owing to the low pressure and low temperature of the air, the drop in temperature will only be slight; however, the exhaust air with its remaining heat is now led to the intake of the secondary air, say, at 300 deg. cent. (572 deg. fahr.), and as it is already partly heated it therefore cannot absorb so much heat from the waste gases. The result is that the latter will now enter the heat exchanger at 600–700 deg. cent. (1112–1292 deg. fahr.) and the compressed air will accordingly be more highly heated and the exhaust air will also have a higher temperature. By circulating the heat the temperature of the waste gases when leaving the recuperator is gradually raised to a point at which mechanical work can be done. This is the method of operating when the turbine is added to existing gas ovens which have recuperators. In new ovens the recuperator can be omitted altogether, and the waste gases after leaving the retort chamber can be led directly to the special heat exchanger. The waste gases having more than double the weight of the secondary air, it is obvious that an equal weight of compressed fresh air may be heated; the other half can then be led as primary air into the gas generator, or may be used under a boiler to generate steam. In a battery of ovens consuming 35 tons of fuel per day, the energy developed is in the first case about 200 hp., beside which there is a saving of fuel; and in the second case when a steam plant is installed, it is about 500 hp. continuously.

From these few examples, which of course could be considerably added to, it will be seen that the Lorenzen turbine is applicable over a range extending from the motor car, flying machine, Diesel engine, locomotive, and ships, up to the largest power stations, and can be operated as an exhaust-gas turbine, a pure gas turbine, or as a hot-air turbine.

In conclusion the author desires to express his sincere thanks to Prof. Fr. Schäff, of Nuremberg, who has been his scientific assistant during the whole development for about fourteen years, and to his son, Max Lorenzen, who has constructed all the turbines shown in the photographs.

Discussion

JAMES A. MOYER.[2] Dr. Lorenzen has presented some very interesting data in regard to a unique design of gas turbine. He has left the beaten path by not attempting to obtain the best possible results from a theoretical standpoint; meaning that, instead of adopting the usual high-efficiency explosion system, he has been content to experiment with the more unusual constant-pressure combustion. By this method he has, however, gained the advantages of eliminating the impact and eddy losses at the entrance edge of the turbine blades which result from the great variation in the velocity of the gasses when the explosion system of combustion is used. It is interesting to note that Dr. Lorenzen has applied with considerable success and in a unique way the idea of velocity stages in his gas-turbine designs. Unquestionably the ingenious blade system which he has introduced marks an important step in the progress of gas-turbine development. Deterioration of the turbine blades due to the high temperature involved has probably been the greatest drawback to successful accomplishments with gas turbines. This method should make the blades of gas turbines sufficiently durable to withstand the high temperatures to which they are exposed.

In order to be ideally successful, however, a gas turbine must combine the high thermal efficiency of the internal-combustion engine with the mechanical simplicity of the modern steam turbine. The designers of gas turbines have been proverbially between "the two horns of dilemma." When they have tried to utilize the velocity of the gases due to a single-stage expansion, the blades and nozzles have been found to deteriorate very much too rapidly for any practical service. On the other hand, when designers have cooled the gasses of combustion by the injection of water or excess air, or even by the method of water-jacketing to make the temperature of the gases suitable for the materials available for blade and nozzle construction, the high thermal efficiency as expressed by the simple laws of thermodynamics is unattainable. By taking a middle course in this case, Dr. Lorenzen has probably brought us much nearer to the goal of practical success.

A. L. BERGER[3] AND OPIE CHENOWETH.[3] The Air Corps has carried on the development of the turbo centrifugal supercharger aircraft since 1917 and has in that time accumulated a vast amount of service experience. From this experience it was soon learned that superchargers having high compression ratios re-

[2] Director, University Extension Division, The Commonwealth of Massachusetts, Department of Education, State House, Boston, Mass. Mem. A.S.M.E.

[3] War Department, Air Corps, Matériel Division, Wright Field, Dayton, Ohio.

quired intercoolers to reduce the temperature of the air entering the carburetor to a figure that would permit satisfactory engine performance. It was also learned that much better turbine bucket life could be obtained by not only removing the housing surrounding the turbine wheel, but by permitting the propeller slipstream to blow over the wheel, thereby cooling the buckets. In view of these experiences, it does not seem reasonable for aircraft to have to remove the heat imparted to the turbine buckets by the air being compressed and then to cool this air by intercooling. This would require a much larger intercooler which would, in turn, increase the resistance of the airplane.

(*a*) When used as a supercharger for aircraft, what would be the estimated weight of a unit suitable for 2 to 1 supercharger compression ratio for a 600 horsepower, four-stroke cycle engine operating at 3000 r.p.m.?

(*b*) When used as a supercharger for an engine of this type, what efficiency could be expected from the unit as a whole and from the turbine?

(*c*) Since the air compressed is used for cooling the turbine blades, what would be the estimated size of an intercooler necessary to reduce the air temperature to 60 or 80 deg. for the engine referred to in question (*a*)?

(*d*) When operating with an exhaust-gas pressure of about 5 lb. per sq. in. at the nozzle box, would there not be a leak of exhaust gas into the air being compressed to such an extent that the output of the engine would be reduced by this exhaust contamination?

CHARLES E. LUCKE.[4] The commercial possibilities of the gas turbine will depend upon two factors, neither of which is mentioned in the paper. It may be assumed that the efficiency of the gas turbine cannot be as high as the efficiency of the internal-combustion piston engine. It is possible that a gas turbine may be built with an efficiency somewhere between that of the internal-combustion piston engine and that of a steam engine or turbine of the same size. This is the first factor—the efficiency obtainable in different sizes of machines so that engineers may compare these values with those now known for steam and internal-combustion piston engines.

The second factor is concerned with cost. If a gas turbine with what might be called an intermediate efficiency can be built, then its commercial place will depend upon its cost; and if this is also intermediate, then there is a possibility of its finding some place, and its possibility is greater if the gas-turbine plant's complete cost should happen to be less than steam of the same size, which, however, is very doubtful.

It does not appear to be really worth while to spend time analyzing machines for which there is no economic need, and therefore for which there is no market, even if it can be built. With reference to the gas turbine, this possible market, which is a measure of the effort justified in developing the machine, will depend jointly on its cost and on its efficiency. The author has entirely ignored both of these questions, and therefore his paper is one of purely academic interest.

LEE SCHNEITTER.[5] What is the method of starting the live gas internal-combustion turbine?

W. G. LUNDQUIST.[6] The writer is not sufficiently familiar with the subject of gas turbines in general to discuss Dr. Lorenzen's turbine as such, but in connection with its use as a turbo-compressor for supercharging aircraft engines he would like to present a few items which occur to him.

The exhaust-driven supercharger admittedly has many advantages not obtained with a gear-driven supercharger, particularly for high-altitude work. However, the salient feature of Dr. Lorenzen's turbo-compressor—namely, the cooling of the turbine blades by the air which is being compressed—would, the writer fears, be detrimental in this case, since the added heat taken from the turbine blades would increase the trouble caused by detonation unless this heat were removed by proper intercoolers, which would complicate the particular installation considerably. Even with gear-driven superchargers rotating at comparatively moderate rates (23,000 to 25,000 r.p.m.) the temperature rise through the supercharger is proving to be a problem, and any additional heat such as introduced by the Lorenzen turbine would be most unwelcome, particularly when supercharging for increased power at sea level or low altitudes. If the turbine were to be used in connection with a separate supercharger impeller, then the installation would resemble the customary turbo-supercharger set up, and the only advantage to be claimed would be improved cooling of the turbine blades. It would be interesting to have a comparison of blade temperatures of such a turbine installation compared to the blade temperatures of a turbo-supercharger installation in which a conventional turbine wheel were operated in the open and directly in the slipstream of the propeller. For comparatively large amounts of supercharging, where it is necessary to employ some method of cooling the air after it comes from the supercharger regardless of the type of drive, the writer believes that the Lorenzen turbo-supercharger could be used advantageously, as it could also be used for supercharging Diesel engines where detonation is not a problem.

It may be true that it is easier to attach and operate successfully a turbo-supercharger to an already built engine that it is to attach and operate a gear-drive supercharger (radial engines excepted), but a comparison of a turbo-supercharger installation with a properly designed gear-driver supercharger will convince any one that the gear-driven compressor offers a cleaner and more compact design than will probably even be attained with a turbo-compressor. Furthermore, a properly designed gear-driven supercharger is more rugged and dependable, since it has no temperamental turbine wheel to cause trouble. Gear-driven superchargers operating at 30,000 r.p.m. and over are not usual.

The writer would appreciate an explanation of the statement in the paper which says that "a steam-turbine plant able to utilize fully the exhaust from a gas turbine will have an output about double that of the gas-turbine plant." A previous statement says that the gas-turbine exhaust does not carry away as much heat as does the exhaust of a steam turbine.

HAROLD CAMINEZ.[7] Exhaust-gas turbines for supercharger drives have been used with limited success in this country. As pointed out by Dr. Lorenzen, the high temperatures and high speeds under which these turbines operate present many difficult mechanical problems. Dr. Lorenzen's method of cooling the supercharger turbine blades may probably solve some of the difficulties experienced with aircraft exhaust-gas turbines.

As pointed out by Dr. Lorenzen, a very high peripheral speed is necessary to obtain good efficiency in a single-stage gas turbine. With a given peripheral speed, the centrifugal effect on the impeller blades is inversely proportional to the diameter, so that the blades on the smaller diameter rotors will be more highly stressed. It is probable that with proper turbine-blade cooling no difficulty would be experienced from this in a stationary machine, but in a supercharger installed in an airplane, which

[4] Professor of Mechanical Engineering, Columbia University, New York, N. Y. Mem. A.S.M.E.

[5] Plant Betterment Engineer, Electric Bond and Share Company, New York, N. Y. Assoc-Mem. A.S.M.E.

[6] Engineering Department, Wright Aeronautical Corporation, Paterson, N. J. Jun. A.S.M.E.

[7] Allison Engineering Company, Indianapolis, Ind.

is subjected to vibration and rapid maneuvers, rotors that are satisfactory for stationary work do give trouble.

The bearings for the rotors are also a source of trouble in an airplane supercharger turbine. In normal oprations these bearings have practically no load, but due to the effect of vibration and maneuvering in aircraft, the gyroscopic action of the high-speed turbine rotor puts a severe load on the rotor bearings.

With Otto-cycle-engine operation, the cooling air from the Lorenzen hollow turbine blades could probably not be employed to advantage unless the air was run through an air cooler before being fed to the engine cylinders. It would be necessary to study whether the weight-head resistance and complications caused by this additional air cooler would make the use of this air from the turbine advisable. However, in a compression-ignition or Diesel-cycle engine, the use of this hot compressed air from the supercharger turbine blades should be desirable, as it would permit the use of lower compression ratios in the engine.

The writer believes that the problems connected with the Lorenzen gas turbine for aircraft supercharger drives are possible of solution and that such a machine would be highly desirable for aircraft Diesel engines, particularly when fitted to engines used in aircraft that are not subjected to violent maneuvers.

CHARLES A. MORSS.[8] Dr. Lorenzen's gas turbine, when used as an exhaust-gas-driven turbo-supercharger for internal-combustion engines appears to be subject to some of the same criticisms which have been directed at other previous designs of this equipment. These criticisms are, mainly, as follows:

(1) Heating of air used for supercharging the piston engine.

(2) The great difficulty of properly supporting in bearings and maintaining free from vibration a wheel traveling at approximately 30,000 r.p.m., and of maintaining proper clearance between the wheel and its casing.

(3) Danger of destruction of the turbine from the use of unclean air for supercharging.

(4) An increase of exhaust back pressure of the piston engine.

(5) A variable velocity of the exhaust gases impinging upon a turbine wheel of constant peripheral speed.

The first three of these criticisms are applicable to Dr. Lorenzen's turbine to a greater degree than to other designs of exhaust-gas turbo-superchargers in which the turbine and supercharging wheels are separate. For example, heating of the air used for supercharging the piston engine is inherent in Dr. Lorenzen's turbine to a marked degree, since the air is used for directly cooling the turbine blades. This characteristic is of course at variance with the requirements of the piston engine, which should receive intake air at the lowest possible temperature in order to get good volumetric efficiency.

Similarly, the need for close and positively maintained clearances between the turbine wheel and its casing is particularly important in Dr. Lorenzen's turbine because he makes use of a labyrinth packing seal between the turbine chamber and the supercharger diffuser.

Finally, the danger of destruction of the turbine due to dirt particles and other foreign matter in the intake air, which is the danger of all high-speed centrifugal superchargers for internal-combustion engines, is particularly great in Dr. Lorenzen's turbine because the air passes at its highest speed through turbine blades of very thin wall section and of temperature approaching red heat.

The last two criticisms mentioned—namely, an increased back pressure on the piston engine and a variable velocity of the exhaust gases—result in lower efficiency, the former of the piston engine and the latter of the turbine. However, these last two objections can be obviated, in whole or in part, by converting the pressure energy of the exhaust gases into velocity energy and storing this velocity energy in what might be called a velocity-energy receiver, thereby imparting a constant velocity to the exhaust gases.

The writer would like to point out, in conclusion on Dr. Lorenzen's exhaust-gas turbine, that in its application as a turbo-supercharger for modern aeronautical gasoline engines it comes into direct competition with mechanically driven centrifugal superchargers, which in their present development are capable of giving almost complete reliability at speeds of 20,000 to 30,000 r.p.m., with a considerable advantage in weight and complexity over the exhaust-gas turbine. The other side of the picture is of course that its application to aeronautical gasoline engines is far from being the only one, or even the most important one, of its possible uses.

F. GLENN SHOEMAKER.[9] About what percentage of the air that passes through the turbine leaks past the rotating elements?

EDGAR J. KATES.[10] The running clearances between the rotor and casing being apparently quite small and the operating temperatures quite high, is there not some danger of distortion and binding when a cold turbine is being started? If so, what precautions must be taken to avoid trouble?

W. M. KAUFFMAN.[11] The analogy of the development of the prime mover described to a similar phase of pioneer development of the steam turbine during a steam-engine age elicits considerable interest in those advantages of the Lorenzen turbine which have justified this enormous work. The advantage of the principle of rotating elements is appreciated. It appears also that a wider range of fuels may be utilized and various thermal cycles employed.

From the standpoint of fuel, we find patent proposals for the use of gas, hot air, exhaust gas, oil, and powdered fuel. This opens at once a field of application for a successful turbine in which these fuels may be used directly. The savings effected over a similar steam-turbine unit with gas- or oil-fired boilers are evident. Plants utilizing gas from coke ovens, gas producers, and blast furnaces would represent possible applications.

However, if one considers the gas turbine relative to the gas engine or Diesel engine in a similar manner as in the steam-turbine analogy, he will find the problems more difficult of solution and the advantages over the piston engine appreciably smaller. It is handicapped in the degree of compression of the working charge. In the piston engine both compression and expansion take place with high mechanical efficiency. This efficiency is approximately 20 per cent better than in the turbo-compressor. A decided advantage in the case of the piston engine lies in the amount of negative work necessary to bring the compressed-air charge into the working cylinder. At present, the temperature necessary in a gas turbine to secure efficiencies exceeding those of the reciprocating engine are such as to be destructive of the available materials.

As regards the field of competition, the basis of merit is not so pronounced. With regard to utilization of a larger range of fuels, a distinct advantage is found.

The writer would ask the question, What types of fuel have been used in the Lorenzen turbine, and what would be the relative advantages or disadvantages of burning oil and powdered fuel,

[8] Formerly Assistant Service Manager, Wright Aeronautical Corporation, Paterson, N. J. Present address, West Chop, Martha's Vineyard, Mass. Assoc-Mem. A.S.M.E.

[9] Assistant Head, Power Plant Section, General Motors Research Laboratories, General Motors Corporation, Detroit, Mich.

[10] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

[11] I. P. Morris & De La Vergne, Inc., Philadelphia, Pa.

as against gaseous fuel, such as coke oven or producer gas?

J. P. Stewart.[12] In this paper there seemed to be some suggestion that the gas turbine might be used to power aircraft. Is it not true that the accessories, such as the combustion chamber and heat interchanger, must be so large as to preclude its possibility, not only for aircraft, but for any mobile vehicle?

When using the turbine as a prime mover, what is the ratio of cooling air to the actual combustion air required by the turbine? Is the former all of the air that the turbine requires, or is additional cold air bled into the combustion chamber?

E. C Magdeburger.[13] It appears that the heat exchanger which apparently contributes so materially to the overall efficiency of the gas-turbine installation, as described by the author, would be a very bulky thing and would bear in size about the same relation to the gas turbine as the steam boiler does to the steam turbine. Will the author please elaborate on the volumetric space occupied by a gas turbine and the corresponding heat exchanger in any installation which he has built so far or purposes to build. The writer also would like to know how large a gas turbine has been actually built and operated according to the designs of the author.

Oliver Field Allen.[14] The purification and increase of the charge in four-cycle engines by having the openings of the exhaust and admission valves overlap while creating a slight suction in the exhaust line was accomplished experimentally some 30 years ago, and various schemes for supplying excess air to two-cycle engines were tried 15 years ago. Supercharging in the modern sense was initiated during the World War to facilitate high-altitude flying by maintaining sea-level atmospheric pressure in the engine cylinders. This development brought forth both exhaust-gas turbine and high-speed, gear-driven turbo-blowers now used as standard equipment on many types of airplanes in America and Europe.

About five years ago the application of exhaust-gas turbine-driven blowers for supercharging land and marine oil engines for sea-level operation was started. Correction for low atmospheric pressure in high-altitude power-plant installations and provision for peak-load operation by supercharging began to be seriously studied, and installations have been succesfully made using electric-motor-driven turbo-blowers and belt-driven positive-pressure blowers, as well as exhaust-gas turbine outfits.

Until quite recently, except for air service, some type of supercharger has been added to existing engine models designed for operation without supercharging. It is only within the last year or so that land and marine engines have begun to be designed to be used regularly as supercharged machines, with the supercharger considered as an essential part, as it is in aircraft motors.

Both the advantages and the limitations of supercharging are better understood now, and Dr. Lorenzen's exhaust-gas turbo-blower comes at a time when the industry is prepared to utilize it.

The gas turbine is a development long dreamed of, and this paper is especially valuable in telling of concrete accomplishments, while giving much to think about as to further developments.

AUTHOR'S CLOSURE

The numerous inquiries made by discussers from the scientific as well as the industrial standpoint indicate the great interest taken in the Lorenzen gas turbine. As several of the questions are propounded more than once by different individuals, the author will deal with all of them in general without referring to specific discussors.

All of the questions propounded may be divided into two classes:

1 Those on the exhaust-gas turbine as a supercharger, and
2 Those on the live-gas turbine as a prime mover.

With regard to the former, many years of practical test results are at the author's disposal to back up the following statement referring thereto. But while the live-gas turbine described in the paper has been built, lack of means has thus far prevented carrying out thorough and complete tests.

The Exhaust-Gas Turbine as a Supercharger

Nearly all the inquiries of this class express fear of excessive heat when using the cooling air from the turbine to supercharge the piston engine. To be quite candid, the same fear was shared by the author as well as by the officials of the German research department for aviation during the preliminary *stationary* test. However, in order to get actual test results it was decided to build a turbine small enough to be suitable for a 40-hp. engine and then fit it into a motor car. The temperature of the cooling air measured during the stationary test ranging from 100 to 125 deg. cent. (= 212 to 260 deg. fahr.), it was thought advisable to employ an intercooler. The tests were started in January, 1928, when it was very cold, and the cooling effect was so great that a bypass was introduced in order to regulate the temperature by uncooled air. It was soon found that the engine ran its best when no intercooling took place. The intercooler was left in the car as it was thought it would be needed in summer time, but even then it was found that there was no need for the intercooling. The arrangement was such that by simply turning a three-way valve the air could either be cooled or led to the engine at the temperature with which it left the supercharger, and hardly any difference could be noticed in the actual running of the car. The reason for this result the author ascribes to the following facts. Owing to the great velocity of the car, from 80 to 100 ft. per sec., and the comparatively large difference in temperature, a considerable portion of the heat is carried away through the casing of the diffuser and the piping leading to the carburetor, both exposed to the outside air, and another portion is used to evaporate the fuel at the jet in the carburetor. This produces a combustion gas instead of an air and gasoline mixture, therefore the explosion is more rapid and better use is made of the fuel in the engine. The latter had a ratio of compression of 1:5.35. The foregoing results are fortified by tests made in Europe and covering more than $2^1/_2$ years, during which a motor car was run more than 50,000 miles. Further tests in America during the hot months of June and July of this year gave a similar result. The engine withstood well the increased temperature of the charge without "knocking," as all combustion engines run better on a hot gas, provided the charge is under pressure to assure that the required weight will be injected. The cooling effect just mentioned could not of course bring down the temperature of the fresh charge in a rapidly moving car or airplane to that of the surrounding atmosphere. However, there is no need for this in the Lorenzen supercharger, as compared with a gear-driven compressor, or when turbine and compressor are two special machines and the former is running without housing and is exposed to the propeller slipstream for cooling purposes. In the latter two cases a considerable proportion of the engine's power is required to drive the supercharger. For example: to compress the air for a 600-hp. engine at a 2:1 ratio, about 100 hp. is needed with the turbo-compressors available today. If, therefore, the output of the engine is to be kept constant, then the latter must

[12] Vacuum Oil Company.
[13] Aide on Diesel Engines, U. S. Navy Department, Washington, D. C.
[14] New York, N. Y. Mem. A.S.M.E.

receive a charge sufficient for an output of about 700 hp., thus overloading the engine. For a gear-driven compressor the energy required is taken from the motor shaft; in a direct-acting turbine and special compressor the pressure in the nozzle box must be increased, which causes back pressure on the engine. It is obvious that a turbine running in a closed casing, being acted upon by all of the motive fluid and having only the ventilation work of one machine to overcome, will run at a higher speed with a given pressure at the nozzles, than when the conditions mentioned by Mr. Berger exist. Owing to the favorable running conditions the pressure in the nozzle box in the Lorenzen supercharger for aviation is considerably less than 5 lb. per sq. in. at sea level, and from 1600 ft. upward is below atmospheric pressure.

With regard to the inquiry about leakage of exhaust gases into the cooling air, it can be stated that there is none, several analyses with Orsat apparatus showing no traces whatever. On the contrary, there is a small leakage of fresh air into the exhaust, which amounts to about 25 per cent of the air drawn in by the compressor. These measurements, however, were made several years ago, since which time improvements have been made by labyrinth sealing, and it will be less today, but no further measurements have been made to enable the author to give exact data.

As regards the life of the bucket, it can be said that after two and a half years of daily use such as that to which a motor car is subjected, there is no sign of wear whatsoever. The hollow blades become black-brown in color, and their surface covered with a thin coating of soot from the exhaust.[15]

The efficiency of the supercharging unit as a whole is about 25 per cent, and the efficiency of a turbine of so small a size, about 65 per cent.

The size of intercoolers in aircraft will not have to be increased as compared with that of other superchargers. In fact, the author is of the opinion that they may be omitted altogether (see test results obtained with motor car).

According to theoretical calculations the temperature of the buckets is thought to be 450 deg. cent. (842 deg. fahr.), but in many actual test runs where the casing of the turbine was at a red heat, when later examined the rotor and the labyrinth packing seal at the top of the blades distinctly showed the well-known blue color which metal exhibits at a temperature of about 300 deg. cent. (572 deg. fahr.).

The required high circumferential speed can safely be obtained owing to the extreme lightness of the blades. In small turbines their weight is about one-sixth of an ounce, and the centrifugal stress when running at full speed is about 1200 to 1400 lb.

No trouble whatsoever due to rapid maneuvers has been experienced during the tests in the car, which often has been run over bad and winding roads. During the two and a half years' tests there have been three cases of a ball bearing's being smashed because of a broken ball. The gyroscopic action in a supercharger suitable for a 600-hp. aviation engine is, at the highest speed of the rotor, 750 cm.-kg. (65,000 in-lb.) at an angular velocity of the airplane as such of 0.5 deg. per sec., and this sets up a thrust of 56 kg. (125 lb.) in the roller bearing which it will easily stand for so short a period.

To avoid distortion through different degrees of expansion of the rotor and casing, the labyrinth joint is located in the diffuser, which has practically the same temperature as the rotor with its buckets; whereas the nozzle box and the outlet housing being fixed in their central portion, can expand radially without interfering with the rotor.

The total weight of a supercharger suitable for a 600-hp. engine is 35 kg. (77 lb.).

[15] Such a "jacket" reduces the heat transmission considerably, and is properly chargeable to the original cooling of the blades.

The Live-Gas Turbine as a Prime Mover

Although no actual test results are available as regards the live-gas turbine, the author does not think he is far from the facts in the following statements. The live-gas turbine has been developed from the exhaust-gas turbine and is of exactly the same construction, only it works under more favorable conditions, inasmuch as there is no pressure container, the working pressure being set up by the air compressor driven by the turbine itself. Therefore the impelling gases are of uniform velocity. There are no regulating, stop or timing devices in the hot zone. The combustion chamber is an enlarged passage of cylindrical shape with fireproof lining in order to insure ignition, and once ignition takes place, combustion is continuous. The output of the turbine is easily regulated by varying the fuel supply and air intake in the compressor, which latter can both be made to work in conjunction so as to assure a constant temperature, although for overloading purposes a special fuel supply is provided for.

With regard to the efficiency curves given in the paper, it should be pointed out that these only refer to a self-contained pure-gas-turbine installation. But there is nothing to prevent—on the contrary, much to warrant—building the Lorenzen turbine in conjunction with a steam-turbine plant or with smelting works or the like where heat is required, because we then obtain an efficiency in the turbine plant of 86 to 90 per cent, certainly a result never imagined. However, this will readily be understood from the following example which refers to a single-stage constant-pressure turbine, and the efficiency will be increased with higher pressure and a multiplicity of velocity stages in the gas turbine's further developments.

Example. The temperature in the Lorenzen gas turbine is limited by the heat resistance of the turbine casing only, and not by the rotor or bucket as has previously been the case. To get a performance of 3500 hp. out of the turbine illustrated in the paper, the working fluid must conform to the following:

Weight	24.4 kg. per sec. (53.7 lb. per sec.)
Pressure	3.1 atmos. (44.1 lb. per sq. in.)
Temperature	1000 deg. cent. (1832 deg. fahr.)
Weight of fuel	0.605 kg. per sec. with 9500 kg-cal. including radiation and combustion losses, or 20,200,000 kg-cal. per hr. or 80,100,000 B.t.u. per hr.

The performance of the turbine is then $75 \times 300 \times 3600/427 =$ 2,200,000 cal. per hr. or 8,734,000 B.t.u. per hr. The exhaust (at 700 deg. cent.) leaves with $178 \times 24.4 \times 3600 =$ 15,650,000 cal. per hr. or 62,140,500 B.t.u. per hr. available for steam plant, or for both together, 70,874,500 B.t.u. per hr. The efficiency is then $= \frac{70,874,500}{80,100,000} = 0.88$ or 88 per cent.

This extraordinary high efficiency is due solely to the fact that the entire exhaust from the turbine can be utilized outside the turbine, owing to its content of oxygen, a special characteristic of the constant-pressure gas turbine only. Of course the output of the turbine working in conjunction with other prime movers is limited by the amount of exhaust which can be used outside the turbine. For the combustion in the turbine plant 25 lb. of air are used, and the exhaust therefore still contains 53.7 — (25 + 0.60) = about 27 lb. unconsumed air. This is sufficient for the further combustion of 0.669 kg. (1.47 lb.) oil under the boiler, which brings up the temperature to about 1500 deg. cent. (2730 deg. fahr.)—a temperature amply high for steam boilers—as compared with a maximum temperature of about 1700 deg. cent. (3100 deg. fahr.) when no work is taken out of the gases.

A comparison with other motors based on the relation of the net output to the heat value of the fuel supplied to the combustion chamber or boiler, gives the following efficiencies:

	Per cent
Lorenzen gas turbine making use of the exhaust	88
Diesel engine making no use of the exhaust	33
Lorenzen gas turbine self-contained (single-stage expansion only)	31
Modern steam-turbine plant	23
Motor-car engine	21

The cost of the Lorenzen gas-turbine plant depends largely upon its application. If it is installed to work in conjunction with other plants or works as described above, then its cost and its space requirements will be considerably less than for any other motor actuated by heat.

The turbine itself is smaller than a steam turbine with the same output. Most of the space is taken up by the air compressor and the combustion chamber, as may be seen from Fig. 12 of the paper, which is drawn to scale. Further, it will be seen that the bulkiest part is the heat exchanger, which, however, is employed only when a "self-contained" gas-turbine plant is in question; otherwise the steam boiler or smelting furnace takes over the function of the heat exchanger, which of course reduces the cost of the plant considerably. In a "self-contained" plant the size of the heat exchanger again depends upon the efficiency required. The scale drawing shows an exchanger giving a total efficiency of 31 per cent. To obtain an efficiency of 35 per cent the size of the exchanger would have to be nearly double, but even then the space required and the weight would be less than in the case of a steam-turbine plant with the same output.

Starting has to be done by auxiliary means of one sort or other. If the turbine is to be used to drive a dynamo then it will be advantageous to build the latter to work as motor-generator so that the turbine can be started electrically. If, however, the turbine is used to drive a ship propeller or the like, the turbine must be disconnected from the driving shaft and air blown into the combustion chamber directly or through the compressor by a blower similar to those used for forced draft under steam boilers. The single-stage-turbine rotor, being free, will start to rotate at a very low pressure after fuel has been added and the temperature of the air raised.

The ratio of cooling air to working fluid is about 1:10. If working in "conjunction" this air is led into the exhaust gases, the heat carried away remaining in the process; if working as a "self-contained" unit the cooling air may be led into a diffuser where its velocity is converted into pressure, and thence through a special combustion chamber when the pressure of the main working fluid is different—a design which has not been considered in estimating the 31 per cent tabulated above.

Whether the turbine can be used as a prime mover in aircraft depends on its further development, and upon improvements in the turbo compressor now in hand.

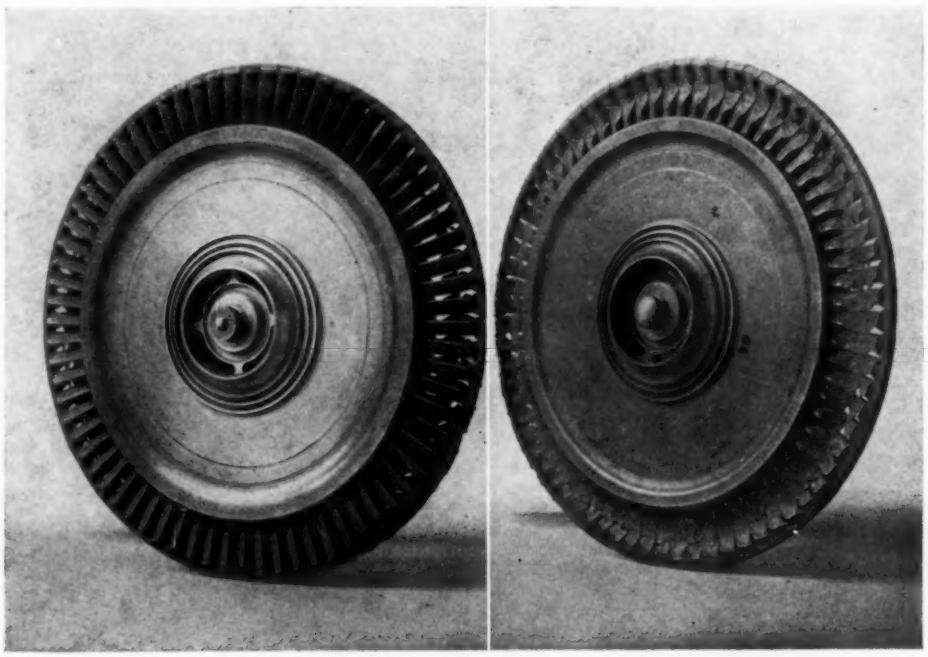


FIG. 14 SHOWING THE HOLLOW TURBINE BLADES AFTER CONCLUSION OF TESTS

The kind of fuel employed is quite immaterial, whether high-B.t.u. oil or gas from coke ovens or blast furnaces. Whether powdered fuel can be used is a matter to be determined by experience; however, the author does not think that such fuels as peat, pulverized straw, or the like in a fine grain will affect the buckets very seriously. The pulverization of peat is obtained by a simple drying process, after which it only needs to be carefully sifted.

The best proof of the reliability of the hollow turbine blades may be seen from Fig. 14. During the stationary tests on a 600–800-hp. aviation engine a connecting rod broke, which smashed the piston, a portion of it being carried by the exhaust into the turbine without being noticed by the attendant. Four months later after the engine had been repaired the tests were continued, but the turbine did not run as well as it had done before. It was therefore taken apart for examination and found in the condition as shown in the photographs. During these stationary tests a temperature from 835 to 885 deg. cent. (1535 to 1625 deg. fahr.) was measured in the turbine housing. In the left-hand illustration the outer ring of the turbine disk is somewhat darker, due to the "blue" color given it by the heat to which it was exposed.

In what has preceded, the author believes he has dealt with all the inquiries put before him, so far as these can be answered on the basis of test results and theoretical calculations. No doubt further developments will introduce difficulties which cannot be foreseen today, but the theoretical calculations with regard to the

live-gas turbine have been made with a certain reserve and great care, particularly with respect to compressor improvements, which will have a dominating effect upon the total efficiency, and in this regard the future may have some surprises in store. It certainly can be said that the working conditions for the live-gas turbine are more favorable with its constant flow and temperature of the working gases than those under which the exhaust-gas turbine works satisfactorily; and if the necessary means are made available, then the further work upon the former will be carried on in the belief that "Where there is a will, there is a way."

OGP-53-2

A Simple Method for the Calculation of Natural Frequencies of Torsional Vibration

By FREDERIC P. PORTER,[1] BELOIT, WIS.

The calculation of the torsional-vibration characteristics of the shafting of reciprocating-engine installations is generally considered a laborious undertaking. This paper presents a method for calculating the natural frequencies of the shafting that is relatively simple in its application and yet retains the accuracy of the longer methods. This simplification is obtained by the proper choice of the equivalent system for mathematical treatment and by the use of tables and charts. The accuracy of the results has been compared with that of the results of other investigators.



AT THE present time it is well known that critical speeds of torsional vibration often cause serious interferences in the operation of reciprocating-engine installations. These interferences may occur in the sinister form of the failure of shafts, the breakage of flywheel or alternator spokes, or the general failure of the engine or driven machinery; or other effects may be observed such as the excessive wear of parts, noise in the engine, and even flickering lights in electrical installations.

It is often suggested that torsional-vibration troubles may be overcome by making the shaft diameters sufficiently large. This is very true for slow-speed engines or engines with a small number of cylinders. As the shaft speeds and the number of cylinders become greater, however, this remedy leads to enormous shaft diameters.

As an alternative it is usually necessary to design the installation so as to place the larger vibrations at certain speeds where they will be the least objectionable, and then avoid them in continuous operation.

The first part of the problem from a design standpoint is the calculation of the natural frequencies of torsional vibration for the shaft arrangements. Various methods have been devised to accomplish this result. There is at least one characteristic which is common to all of these methods, namely, the reduction of the actual installation to an irregular shaft line-up which has approximately equivalent torsional-vibration characteristics.

One type of equivalent shaft arrangement is obtained by considering each cylinder, flywheel, and alternator, and other large compact parts as concentrated masses connected by elastic shafts having no mass. This equivalent arrangement will be called the "concentrated-mass system." Carried out in the proper detail this method gives reliable results. For a six-cylinder engine with flywheel and alternator this would give an equivalent system of eight concentrated masses connected by seven elastic shafts. The solution is then carried out by either an analytical or graphical procedure. For examples of the solution for this type of equivalent system, reference may be made to the published material by Holzer,[2] Geiger,[3] Wydler,[4] and others.

Another type of equivalent shaft arrangement is obtained by averaging the various masses distributed along the shafting of the installation into a number of connected uniform shafts having both mass and elasticity. This equivalent arrangement will be called the "uniform-shaft system." In the general application of this method certain of the larger masses, such as heavy flywheels, may be considered concentrated at certain points of the uniform shafts. This method was given by Prof. Frank M. Lewis,[5] who proposed a graphical procedure for the solution. An analytical solution for this type of equivalent system and for any combination with the former one was given by the author in a previous paper.[6]

While these various methods, if carried out with sufficient care, do give reliable results, their application often takes considerable time. For general application by designing engineers there is therefore an increasing need for simpler methods of calculation that are both rapid and reliable. It is the purpose of this paper to present such a method for calculating the natural frequencies of torsional vibrations in the shafting of reciprocating-engine installations.

The equivalent shaft arrangement used with the simpler method of calculation for multi-cylinder engines is a combination shaft system. For example, the inertia of the engine parts is combined to form a uniform shaft with mass by spreading the inertia at each crank over half of a cylinder distance on each side of its center line. Then if the flywheel is attached to one end of the crankshaft and is not far from the adjacent cylinder, sufficient mass is deducted from the flywheel to make the uniform shaft

---

[1] Fairbanks, Morse & Co. Mr. Porter was born in Massachusetts. After graduating from Webb Institute of Naval Architecture, he was employed by the New York Navy Yard, where he later became assistant mechanical engineer. In 1928 he went with the Fairbanks, Morse company as design calculator. Previous papers on the subject of torsional vibration by Mr. Porter are: "The Range and Severity of Torsional Vibration in Diesel Engines," Paper No. APM-50-8, 1928, also APM-50-14, 1928, and "Practical Determination of Torsional Vibration in an Engine Installation Which May Be Simplified to a Two-Mass System," Paper No. APM-51-22.

Contributed by the Oil and Gas Power Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] Holzer, "Die Berechnung der Drehschwingungen," Verlag von Julius Springer, 1921.

[3] Geiger, "Mechanische Schwingungen und ihre Messung," Verlag von Julius Springer, 1927.

[4] Wydler, "Drehschwingungen in Kolbenmaschinenanlagen," Verlag von Julius Springer, 1922.

[5] Lewis, "Torsional Vibration of Irregular Shafts," Journal of The American Society of Naval Engineers, Nov., 1919, p. 857; also "Torsional Vibration in the Diesel Engine," Society of Naval Architects and Marine Engineers, Nov., 1925.

[6] "The Range and Severity of Torsional Vibration in Diesel Engines," Trans. A.S.M.E., Paper No. APM-50-8, 1928.

over the cylinders extend to the flywheel. The flywheel and other isolated heavy masses are considered concentrated and connected by elastic, massless shafts. In this way for a six-cylinder engine with flywheel and alternator there are three masses and two shafts as compared with eight masses and seven shafts when using the concentrated-mass system.

The accuracy of the combination equivalent arrangement has been examined and compared with the results obtained by the concentrated-mass system. These results are given in the form of curves and will be discussed later in the paper.

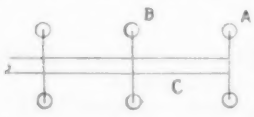


FIG. 1

A fundamental idea in the development of the method may be explained as follows: In Fig. 1, consider that A is a concentrated mass at the beginning of a shaft arrangement, that B is the next concentrated mass, and that C is a shaft without mass. Then at a given frequency, the effect of A acting upon B through the elastic shaft C can be obtained at B by adding a certain fraction of A to B. This sum can again be considered as the first mass, whose effect can be reduced to the next concentrated mass. This process of reduction is aided by tables that give the fraction of the first mass to be added to the next one. If the shaft C has both mass and elasticity, the reduction of both A and the shaft C to B is made by means of a formula, or for the case mass A equals zero, the reduction is also aided by tables.

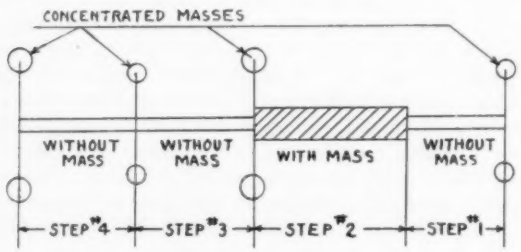


FIG. 2

The masses may be reduced from the beginning to the end, or backward from the end to the beginning, or from both ends up to some section. A natural frequency is obtained when the sum of the masses reduced from both ends up to any section added to the mass at that section is zero. At this frequency the elastic forces in the shafts and the inertia forces of the masses are in a state of equilibrium so that the system when once set into motion at this frequency will continue to vibrate indefinitely (assuming the non-existence of damping influences).

This reduction method has certain features that make a very rapid calculation for the natural frequencies when used with the combination equivalent system for multi-cylinder installations.

In using this method a certain symbolism is necessary which clearly represents the various parts of the process and shows at a glance how far the process has been carried.

The various elastic shafts used in the equivalent system with or without mass will be called "steps" of the line-up, and will be numbered in the order in which they occur. This is illustrated in Fig. 2. The step that is cross-hatched represents a step with mass, while the others without hatching lines represent steps without mass. This same convention will be used throughout the paper. Concentrated masses will be placed only between steps, but may or may not be placed at the beginning or end of steps with mass.

The letter k will mean the number of a typical step under discussion. When so used the characteristics of step No. 1 are obtained by putting $k = 1$, and of step No. 2 by putting $k = 2$, etc. The number of the end step will sometimes be denoted by the letter e.

The *length* of each step will be denoted by l (inches). If no subscript is attached it will refer to any step or length of shaft under discussion.

The rotating *inertia factor* for the various masses in the system will be denoted by J (lb-in.^{2}). This factor is similar to the mass polar moment of inertia coefficient except that it is considered in weight and length units. For this reason it will be called the "weight polar moment of inertia" of the mass, or for simplicity the "inertia factor" of the mass. The symbol J_w will represent the weight polar moment of inertia factor per unit length of a step which has mass. In this case the whole inertia of the step is $J = lJ_w$.

The *stiffness factor* of a step or certain length of shaft is denoted by C. This factor is the constant representing the proportional relationship of any twisting moment M (lb.-in.) and the resulting angular deflection θ (radians) of the shaft such that $M = C\theta$. Thus $C = \dfrac{M \text{ (lb-in.)}}{\theta \text{ (radians)}}$. For a circular shaft of constant diameter, $C = GJ_s/l$ where G(lb./in.^{2}) is the shearing modulus of elasticity and J_s (in.^{4}) is the sectional polar moment of inertia of the area of the shaft. If d is the shaft diameter in inches, $J_s = \pi d^4/32$. For steel, $G = 11{,}800{,}000$ lb. per sq. in., so that in this case

$$C = 1{,}158{,}460\,\frac{d^4}{l}\,\frac{\text{(lb-in.)}}{\text{(radians)}} \qquad [1]$$

Subscripts are used to indicate the part of the equivalent system referred to. A single subscript refers to a step. For example: in Fig. 3, l_1, C_1 are the length and stiffness factor for step No. 1; l_2, J_2, C_2 are the length and inertia and stiffness factors for step No. 2, etc. If the subscript is made up of two numbers differing by unity such as 01, 12, 23, . . . the element with the subscript occurs between the steps whose numbers are given. This is also illustrated in Fig. 3, where J_{01}, J_{23}, J_{34} . . . are the concentrated masses occurring at the beginning of the first step, between the second and third, third and fourth, etc.

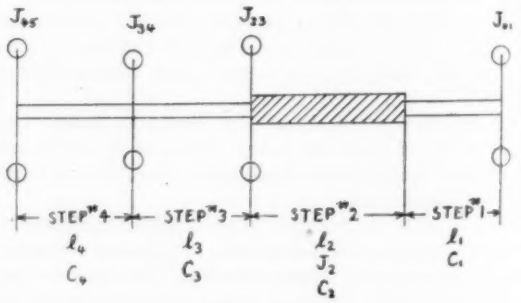


FIG. 3

In using the reduction process in making a solution, a prime mark (′) means that the effect of the masses from the beginning of the system has been reduced up to a certain section. A double prime mark (″) means that the effect of the masses from the end of the system has been reduced back to a certain section. Subscripts are then employed to indicate the section to which the reduction has been made. The subscripts 01, 12, 23 . . . mean the end of steps Nos. 1, 2, 3, etc., and 02, 03, 04 . . . mean the beginning of steps Nos. 2, 3, 4, etc. When reducing from the beginning of the system to the end of any step, such as the kth step, the reduced inertia factor is J'_{0k}. This factor does not

include the concentrated mass between this step (kth) and the next one ($k + 1$th), that is, J'_{1k} does not include $J_{k,k+1}$. The reduced factor at the beginning of the next step ($k + 1$th), however, does include $J_{k,k+1}$. Thus, inertia factors reduced from the beginning are

$$\left.\begin{aligned} J'_{01} &= J_{01} \\ J'_{02} &= J'_{11} + J_{12} \\ J'_{03} &= J'_{12} + J_{23} \\ &\cdots\cdots \\ J'_{0,k+1} &= J'_{1k} + J_{k,k+1} \end{aligned}\right\} \quad \cdots\cdots\cdots\cdots [2]$$

In reducing the effect of the masses backward from the end of the system to the beginning of any step, such as the kth step, the reduced inertia factor is J''_{0k}. This factor does not include the concentrated mass between this step (kth) and the next one (k–1th) toward the beginning of the system, that is, J''_{0k} does not include $J_{k-1,k}$. The reduced factor at the end of the next step (k–1th) toward the beginning, however, does include $J_{k-1,k}$. Thus the inertia factors reduced from the end are

$$\left.\begin{aligned} J''_{1e} &= J_{e,e+1} \\ J''_{1,e-1} &= J''_{0e} + J_{e-1,e} \\ &\cdots\cdots \\ J''_{1,k-1} &= J''_{0k} + J_{k-1,k} \end{aligned}\right\} \quad \cdots\cdots\cdots\cdots [3]$$

The factors ϕ and ψ are also used in the solution. The value of ϕ is given by

$$\phi = 360\sqrt{\frac{J}{gC}} = 18.325\sqrt{\frac{J}{C}}\ \text{(sec.} \times \text{deg.)} \cdots\cdots [4]$$

where J and C refer to a step with mass and g, the gravitational constant, is taken as 385.92 in./sec.^{2} (which is 32.16 ft./sec.^{2}).

The value of ψ is given by

$$\psi = \sqrt{\frac{J}{C}} \cdots\cdots\cdots\cdots [5]$$

where J is a reduced inertia factor. The same subscripts are used with ψ as are used with the reduced inertia factor J. The subscript for C is the same as the step number in the subscript for ψ. To complete the similarity of the symbols, prime and double-prime marks will be used with ψ to indicate that the direction of the reduction is toward the end or back toward the beginning, respectively.

Hence

$$\left.\begin{aligned} &\psi'_{01} = \sqrt{\frac{J'_{01}}{C_1}}, \quad \psi'_{02} = \sqrt{\frac{J'_{02}}{C_2}} \cdots \psi'_{0k} = \sqrt{\frac{J'_{0k}}{C_k}} \\ &\psi''_{1e} = \sqrt{\frac{J''_{1e}}{C_e}}, \cdots \psi''_{1k} = \sqrt{\frac{J''_{1k}}{C_k}} \end{aligned}\right\} \quad [6]$$

The frequency of vibration will be denoted by n in vibrations per second.

The formulas necessary for the solution of the *natural frequency* for any equivalent system are listed below.

I Reducing the Inertia Factors From the Beginning Toward the End.

a For steps without mass:

$$J'_{1k} = J'_{0k}T_0(\psi'_{0k}n) \cdots\cdots\cdots\cdots [7]$$

b For steps with mass:

1 If the first step has no concentrated mass at the beginning,

$$J'_{11} = J_1T(\phi_1 n) \cdots\cdots\cdots\cdots [8]$$

2 If the first step has a concentrated mass at the beginning or for any other step,

$$\tan\gamma'_k = \frac{\phi_k n J'_{0k}}{57.30J_k}, \quad J'_{1k} = \frac{57.30J_k}{\phi_k n}\tan(\phi_k n + \gamma'_k) \cdots [9]$$

II Reducing the Inertia Factors From the End Toward the Beginning.

a For steps without mass:

$$J''_{0k} = J''_{1k}T_0(\psi''_{1k}n) \cdots\cdots\cdots\cdots [10]$$

b For steps with mass:

1 If the last step has no concentrated mass at the end,

$$J''_{0e} = J_eT(\phi_e n) \cdots\cdots\cdots\cdots [11]$$

2 If the last step has a concentrated mass at the end or for any other step,

$$\tan(\phi_k n + \gamma''_k) = -\frac{\phi_k n J''_{1k}}{57.30J_k}, \quad J''_{0k} = -\frac{57.30J_k}{\phi_k n}\tan\gamma''_k \cdots [12]$$

III At a Natural Frequency.

$$J'_{1,k-1} + J''_{0k} + J_{k-1,k} = 0, \text{ or } J'_{1k} + J''_{0,k+1} + J_{k,k+1} = 0 \cdots [13]$$

which is the same as:

$$J'_{0k} + J''_{0k} = 0, \text{ or } J'_{1k} + J''_{1k} = 0, \text{ or } \frac{J'_{0k}}{J'_{1k}} = \frac{J''_{0k}}{J''_{1k}} \cdots [14]$$

The proofs for the above formulas are given in an appendix.

The γ-constants are defined by the equations and need no further explanation. At a natural frequency of the system the γ-constants are such that $\gamma'' - \gamma' = 0$. In this case the prime and double-prime marks are sometimes omitted on these constants.

The relative-amplitudes curve of the vibration at any section of the system follows almost immediately from the above solution. The amplitudes are represented by θ (radians) or $\theta°$ (degrees) and are given the same subscripts that are given to the reduced inertia factor at the same section. In this case, however, the amplitude at the end of any step is equal to that at the beginning of the next step, thus:

$$\theta_{11} = \theta_{02}, \quad \theta_{12} = \theta_{03}, \ldots \theta_{1k} = \theta_{0,k+1}, \ldots \theta_{1e} = \theta_{0,e+1} \cdots [15]$$

The relative amplitude at any point of a step with mass will be denoted by θ with the step number as a subscript. It will be expressed as a function of x, which is the length variable for the step with its origin at the beginning of the step.

The formulas to obtain the relative-amplitudes curve at a natural frequency, are:

1 For steps without mass:

$$\frac{\theta_{1k}}{\theta_{0k}} = \frac{J'_{0k}}{J'_{1k}} = \frac{J''_{0k}}{J''_{1k}} = \frac{1}{T_0(\psi'_{0k}n)} = T_0(\psi''_{1k}n) \cdots\cdots [16]$$

2 For steps with mass:

$$\theta°_k = \alpha_k\cos\left(\frac{\phi_k n}{l_k}x + \gamma_k\right) \cdots\cdots\cdots\cdots [17]$$

where

$$\theta°_{0k} = \alpha_k\cos\gamma_k \quad \text{and} \quad \theta°_{1k} = \alpha_k\cos(\phi_k n + \gamma_k) \cdots [18]$$

which define the α-constants.

The relative vibration twisting moments for the various sections of the system, therefore, are:

1 For steps without mass:

TABLE 1
GENERAL TABULAR FORMS FOR REDUCING FROM BEGINNING TOWARD END

TABLE 1A

FIRST STEP WITHOUT MASS				
(1)	(2)	(3)	(4)	(5)
n	$\psi'_{01}n$	$T_0(2)$	J'_{21} (3)·J_{01}	J'_{02} (4)+J_{12}

TABLE 1B

k-TH STEP WITHOUT MASS			
a	(b)	(c)	(d)
$\psi'_{0k}n$ $n\cdot\sqrt{\frac{J'_{0k}}{C_k}}$	$T_0(a)$	J'_{2k} (b)·J'_{0k}	$J'_{0,k+1}$ (c)+$J_{k,k+1}$

TABLE 1C

FIRST STEP WITH MASS (IF $J_{01}=0$)				
(1)	(2)	(3)	(4)	(5)
n	$\phi'_1 n$	$T(2)$	J'_{21} (3)·J_1	J'_{02} (4)+J_{12}

TABLE 1D

k-STEP WITH MASS (FOR FIRST STEP IF $J_{01}\neq 0$, AND FOR ANY OTHER STEP)						
(a)	(b)	(c)	(d)	(e)	(f)	(g)
$\phi'_k n$	$\frac{(a)\cdot J'_{0k}}{57.30 J_k}$	γ'_k $\tan^{-1}(b)$	$\phi'_k n+\gamma'_k$ (a)+(c)	$\tan(d)$	J'_{2k} $57.30 J_k\frac{(e)}{(a)}$	$J'_{0,k+1}$ (f)+$J_{k,k+1}$

TABLE 2
GENERAL TABULAR FORMS FOR REDUCING FROM END BACK TOWARD BEGINNING

TABLE 2A

LAST STEP WITHOUT MASS				
(1)	(2)	(3)	(4)	(5)
n	$\psi''_{2e}n$	$T_0(2)$	J''_{0e} (3)·$J_{e,e+1}$	$J''_{2,e-1}$ (4)+$J_{e-1,e}$

TABLE 2B

k-STEP WITHOUT MASS			
(a)	(b)	(c)	(d)
$\psi''_{2k}n$ $n\cdot\sqrt{\frac{J''_{2k}}{C_k}}$	$T_0(a)$	J''_{0k} (b)·J''_{2k}	$J''_{2,k-1}$ (c)+$J_{k-1,k}$

TABLE 2C

LAST STEP WITH MASS (IF $J_{e,e+1}=0$)				
(1)	(2)	(3)	(4)	(5)
n	$\phi_e n$	$T(2)$	J''_{0e} (3)·J_0	$J''_{2,e-1}$ (4)+$J_{e-1,e}$

TABLE 2D

k-TH STEP WITH MASS (FOR LAST STEP $J_{e,e+1}\neq 0$, AND FOR ANY OTHER STEP)						
(a)	(b)	(c)	(d)	(e)	(f)	(g)
$\phi_k n$	$\frac{-(a)\cdot J''_{2k}}{57.30 J_k}$	γ''_k $\tan^{-1}(b)$	$\phi_k n+\gamma''_k$ (c)−(a)	$\tan(d)$	J''_{0k} $-57.30 J_k\frac{(e)}{(a)}$	$J''_{2,k-1}$ (f)+$J_{k-1,k}$

TABLE 3
TABULAR FORMS FOR SOLVING FOR C_k OF A STEP WITHOUT MASS TO GIVE A CERTAIN FREQUENCY. SOLVE FOR J'_{0k} AND J''_{2k} AND USE EITHER TABLE 3A OR 3B

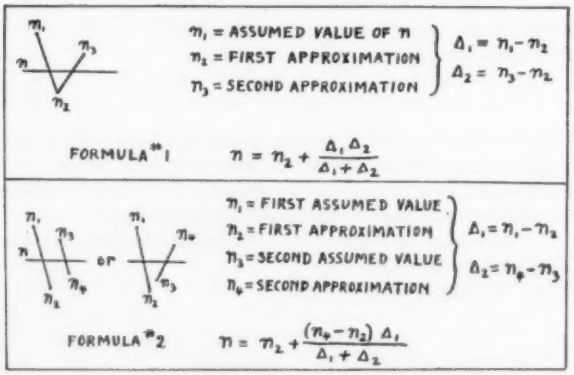
TABLE 3A

(a)	(b)	(c)	(d)
$-\frac{J'_{0k}}{J''_{2k}}$	$T_0^{-1}(a)$	$\left(\frac{(b)}{n}\right)^2$	C_k $\frac{J''_{2k}}{(c)}$

TABLE 3B

(a)	(b)	(c)	(d)
$-\frac{J''_{2k}}{J'_{0k}}$	$T_0^{-1}(a)$	$\left(\frac{(b)}{n}\right)^2$	C_k $\frac{J'_{2k}}{(c)}$

TABLE 4
INTERPOLATION FORMULAS FOR NATURAL FREQUENCY n



$$M = C_k(\theta_{0,k+1} - \theta_{0,k}) = \frac{C_k}{57.30}(\theta^\circ_{0,k+1} - \theta^\circ_{0,k}) \ldots [19]$$

2 For steps with mass:

$$M = -\left(\frac{\pi}{180}\right)^2 C\phi n\alpha \sin\left(\frac{\phi n x}{l} + \gamma\right) \ldots\ldots [20]$$

The functions $T_0(\psi n)$ and $T(\phi n)$ are defined by the equations

$$T_0(\psi n) = \left[1 - \frac{4\pi^2}{g}(\psi n)^2\right]^{-1} = [1 - 0.10229689(\psi n)^2]^{-1}. [21]$$

$$T(\phi n) = \frac{180}{\pi \phi n} \tan \phi n \ldots\ldots\ldots\ldots\ldots [22]$$

where ϕn is an angle in degrees.

One of the time-saving devices of the reduction method is the use of the tabulated values for $T_0(\psi n)$ and $T(\phi n)$ (Tables 12–14). Two tables (12 and 13) are given for the function $T_0(\psi n)$; the first for positive and the second for negative values of the reduced inertia factor. In the latter case it will be noted that ψ is an imaginary number. The tables have been constructed to give as far as practicable a maximum error of 5 parts in 20,000 in the tabular value. To do this it was necessary to give five significant figures for numbers beginning with the numeral 1, and 4 significant figures for numbers beginning with numerals 2 to 9,

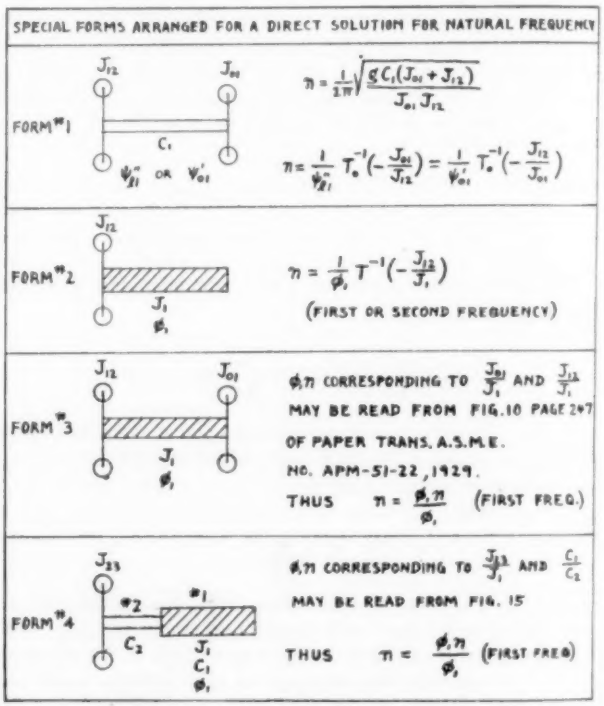


FIG. 4

inclusive. Where the tabular value ends in $\bar{5}$ or 5, the value is actually a little less or a little greater than 5, respectively. The average differences of each line are given where possible. If this average difference is inclosed in parentheses, it refers to the values after being contracted one digit. In working with the tables, $T_0^{-1}(A)$ or $T^{-1}(A)$ mean the values of ψn or ϕn, respectively, corresponding to the listed value A.

A tabular form is used in making a solution. In the general case, these forms for the various steps are illustrated in Tables 1 and 2.

Parentheses placed around letters or numbers in the tables refer to columns designated by the letter or number.

If the value of the inertia factor $J_{k,k+1}$ of a concentrated mass to give a certain natural frequency is desired, it may be obtained from Equation [13] by first solving for J'_{1k} and $J''_{0,k+1}$ by means of a tabular solution; then

$$J_{k,k+1} = -(J'_{1k} + J''_{0,k+1})$$

If the value of the stiffness factor C_k for a step without mass to give a certain natural frequency is desired, it may be obtained by solving for J'_{0k} and J''_{1k} and using either Table 3A or Table 3B.

Four special equivalent systems are given in Fig. 4 whose natural frequencies may be obtained directly from the tables for T_0 or T, or from the charts referred to.

The solution for form No. 1 is given directly by a formula. It is a little easier, however, to use the tables than the formula. Marine installations with engines amidships often approximate this form for the first natural frequency of the arrangement. This type of system was discussed in detail in a previous paper by the author.[7] The factors α and β used in this previous paper are the same as the present values $[T_0(\psi'_{01}n)]^{-1}$ and $T_0(\psi''_{12}n)$, respectively. Thus β corresponds to the tabular values for $T_0(\psi n)$ and α corresponds to the reciprocals of the tabular values.

Forms 2 to 4 may often be used to give a direct solution for certain installations. For example, a multi-cylinder engine driving a flywheel and rotor of an electric machine stiffly connected to the flywheel or made integral with it may be solved directly by one of these forms.

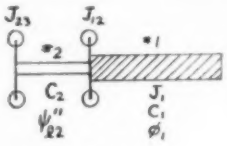


Fig. 5

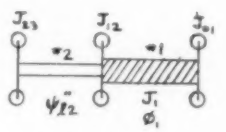


Fig. 6

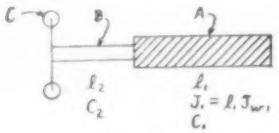


Fig. 7

In making solutions for more complicated systems, it often happens that some part of the system corresponding to either of the simpler forms, No. 1 or 2, has a larger influence in deciding one of the natural frequencies than any of the other parts. This characteristic is usually not difficult to recognize. As an example of this, consider a marine installation consisting of an engine, flywheel, propeller shafting, and propeller. It is easy to see that the first natural frequency is determined for the most part by the flywheel, propeller shafting, and propeller corresponding to form No. 1, whereas the second natural frequency will be determined for the most part by the flywheel and engine corresponding to form No. 2.

Taking advantage of the characteristic just described, the inertia factors may be reduced along from either end for an assumed frequency (n_1) until the remaining system corresponds to the form (1 or 2) having the greater influence upon the frequency.

A first approximation (n_2) for the natural frequency is thus obtained. Using this value and repeating the process a second approximation (n_3) will be obtained. If the simpler system does have the greater influence upon the frequency n_1, n_2, n_3 will form a converging series of numbers alternately on one side and then on the other of the actual frequency. In this case the frequency will be obtained with little error by using the interpolation formula No. 1 of Table 4.

It is not necessary to use the first approximation (n_2) in repeating the calculation a second time. A second assumed value (n_3) may be used which will give a second approximate value (n_4). In this case the actual frequency will be obtained by using the interpolation formula No. 2 of Table 4.

The value of the natural frequency found by the interpolation formulas of Table 4, are remarkably accurate even when the series n_1, n_2, n_3, or n_4 are slightly divergent.

Another procedure to solve for n in the general case, which may often be used with convenience is to reduce the inertia factors from each end to a step that has mass, and with several assumed values for n seek for the value that makes $\gamma'' - \gamma' = 0$. In this case a linear interpolation may be used between n and $\gamma'' - \gamma'$.

In dealing with the system shown in Fig. 5, the solution may be made by reducing from the beginning as in Table 5A or from the end as in Table 5B. If the way to proceed is not directly apparent, a reference to the curves in Fig. 15 will usually indicate the procedure for this system. For example, consider that J_{12} is removed. Then if $\phi_1 n$ in Fig. 15 is found to be less than 90 deg., Table 5A should be used for the first frequency; if $\phi_1 n$ is greater than 90 deg., Table 5B should be used; and if $\phi_1 n = 90$ deg., the first frequency is given by $90/\phi_1$ since J_{12} is at the node and has no effect on this frequency. If the first frequency is obtained rapidly from one of these tabular solutions, the second frequency will usually follow rapidly from the other.

Fig. 6 and Tables 6A and 6B show two tabular forms for another type of system. Other forms may be easily written down for any system by a reference to Tables 1 and 2. The proper procedure in making a solution, if not directly apparent, will soon become so after a few trial solutions.

TABLE 5A

REDUCING FROM BEGINNING UP TO STEP #2 BY TABLE 1C AND SOLVING FOR APPROXIMATE VALUES OF n BY FORM #1 OF FIG. 4.

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
n ASSUMED	$\phi_1 n$	$T(2)$	J'_{21} (3)·J_1	J'_{02} (4)+J_{12}	$-\frac{J'_{02}}{J_{23}}$	$T_0^{-1}(6)$	$n = \frac{(7)}{\psi''_{12}}$

TABLE 5B

REDUCING FROM END BACK TO STEP #1 BY TABLE 2A, AND SOLVING FOR APPROXIMATE VALUES OF n BY FORM #2 OF FIG. 4.

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
n ASSUMED	$\psi''_{21} n$	$T_0(2)$	J''_{02} (3)·J_{23}	J''_{21} (4)+J_{12}	$-\frac{J''_{21}}{J_1}$	$T^{-1}(6)$	$n = \frac{(7)}{\phi_1}$

TABLE 6A

REDUCING FROM BEGINNING UP TO STEP #2 BY TABLE 1D, AND SOLVING APPROXIMATE VALUES OF n BY FORM #1 OF FIG. 4

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)
n ASSUMED	$\phi_1 n$	$\frac{(2)\cdot J_{01}}{57.30 J_1}$	γ'_1 $\tan^{-1}(3)$	$\phi_1 n + \gamma'_1$ (2)+(4)	$\tan(5)$	$\frac{J'_{21}}{57.30 J_1}\frac{(6)}{(2)}$	J'_{02} (7)+J_{12}	$-\frac{J'_{02}}{J_{23}}$	$T_0^{-1}(8)$	$n = \frac{(10)}{\psi''_{12}}$

TABLE 6B

REDUCING FROM END BACK TO STEP #1 BY TABLE 2A AND SOLVING BY FINDING $\gamma''_1 - \gamma'_1 = 0$

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)
n	$\psi''_{21} n$	$T_0(2)$	J''_{02} (3)·J_{23}	J''_{21} (4)+J_{12}	$\phi_1 n$	$-\frac{(6)\cdot(5)}{57.30 J_1}$	$\phi_1 n + \gamma''_1$ $\tan^{-1}(7)$	γ''_1 (8)−(6)	$\frac{(6)\cdot J_{01}}{57.30 J_1}$	γ'_1 $\tan^{-1}(10)$	$\gamma''_1 - \gamma'_1$ (9)−(11)

Determination of Equivalent System

As previously mentioned, the equivalent-shaft system more conveniently used with the reduction method of solution is a combination of the concentrated-mass system and the uniform-shaft system. The inertias of the engine parts are averaged into a uniform shaft with mass. If the cylinders are equally spaced, the inertia at each crank is spread over the shaft for half a cylinder distance on each side of the cylinder center line. If the cylinders are not equally spaced, but occur in equal groups the inertia of each group is spread uniformly over the shafting for each group. The flywheel and other isolated heavy masses that are attached to the shaft over a relatively short length are considered as concentrated masses. Light shafts between heavy masses are considered without mass unless, as described for the engine, several equal masses connected by equal elasticities are averaged into a uniform shaft with mass.

Following this method, it will often be found (see Fig. 7) that there is a short length of shaft (B) between the uniform

[7] Trans. A.S.M.E., 1929, Paper No. APM-51-22.

shaft (A) representing the engine and the concentrated mass (C) representing the flywheel that has a much smaller inertia factor (J_w) per unit length than A. In this case the length of B is changed to have the same stiffness per unit length as A. This is done by making the length of B equal to

$$l_2 = \frac{l_1 C_1}{C_2}$$

where l_1 = length of A
C_1 = stiffness factor of A
C_2 = stiffness factor for B.

Then sufficient inertia is deducted from the flywheel (C) to make the value of J_w over B equal to its value for A. The limit to

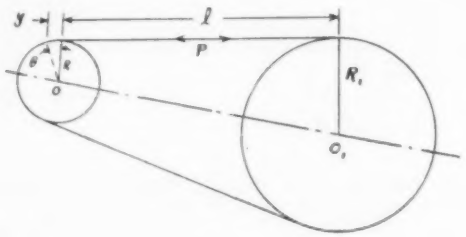


FIG. 8

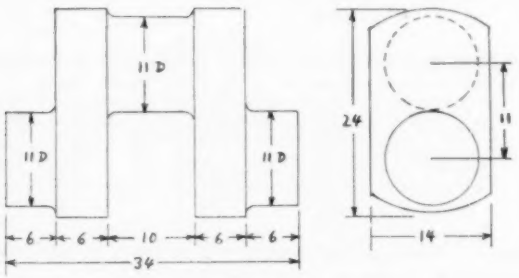


FIG. 9

which this can be carried before the accuracy is appreciably affected is illustrated by the curves in Fig. 16.

Another simplification of the equivalent system is obtained by spreading out a small concentrated mass occurring at the end of the engine opposite the flywheel so that the length of the uniform shaft representing the engine is increased by the inertia factor of the small concentrated mass divided by the value of J_w for the engine shaft. The limit to which this can be carried without appreciably affecting the accuracy is illustrated in Fig. 17.

In multi-cylinder engine installations considerable simplification is obtained by averaging the engine inertia values into a single uniform shaft instead of considering separately the inertia values at each cylinder as concentrated masses. To show that it is unnecessary to consider the engine masses by the longer concentrated-mass method to obtain accuracy, a comparison has been made of the results obtained by the two methods for the first frequency of an installation comprising an engine and flywheel. The results of the comparison are shown in Fig. 18. In an actual installation where the engine and flywheel are connected to other masses and shafts, the percentage differences in frequency will be smaller than the values given for frequencies with one node in the engine. For engines with four or more cylinders, the two methods give very little difference in frequency. For engines with two or three cylinders the frequency with one node in the engine is usually so high that noticeable critical speeds of this frequency are rarely met with in practice. Nevertheless since engines with a small number of cylinders usually have counterweights at each crank web, it seems probable that the actual installation is more like the uniform—shaft system than the system of concentrated masses. Even for the two-cylinder engine, however, the difference in this frequency from the two methods for the usual flywheel sizes will be less than two per cent. For the six-cylinder engine this difference is less than 0.5 per cent, and for a ten-cylinder engine it is less than 0.1 per cent for the usual flywheels.

In making the detailed calculations for the inertia and stiffness factors of the installation, the author has found the following working rules give reliable results.

Connecting Rod. If W is the weight of the rod and K is the distance of the center of mass of the rod from the crankpin center line divided by the connecting-rod length between the crankpin and wristpin center lines, then

Rotating part of connecting rod = $(1 - K)W$
Reciprocating part of connecting rod = KW

Reciprocating Mass. If W is the weight of the reciprocating masses and K is the radius of the crank divided by the length of the connecting rod, then

$$\left(\frac{1}{2} + \frac{K^2}{8}\right) W$$

considered as a rotating weight concentrated on the crankpin center line gives approximately the same effect as the reciprocating masses.

Geared Drives. If a mass having an inertia factor J is geared to a shaft such that the speed ratio of the mass to the shaft is h, then the inertia factor reduced to the shaft is h^2J.

Belt Drives. If a mass having an inertia factor J is belted to a shaft, it may be reduced to the shaft as follows:

In Fig. 8, let o_1 be the center of the shaft and o the center of the mass. Let R_1 be the pulley radius on the shaft and R the pulley radius attached to the mass. If a torque M acting on the mass causes a tension P on one side of the belt such that the stretch of the length l of the belt is y, then[8]

$$M = C\theta = C\frac{y}{R} = \frac{C}{R}\frac{Pl}{AE} = PR$$

where A = cross-sectional area of the belt and
E = modulus of elasticity in tension of the belt material.

Therefore

$$C = \frac{R^2AE}{l} \quad \ldots\ldots\ldots\ldots\ldots\ldots [23]$$

The modulus of elasticity for leather[9] varies between 35,500 and 71,000 lb. per sq. in.

If h is the pulley ratio R_1/R, let

$$\psi = \sqrt{\frac{h^2J}{C}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [24]$$

Hence the equivalent inertia factor at the shaft is

$$J' = h^2JT_0(\psi n) \quad \ldots\ldots\ldots\ldots\ldots\ldots [25]$$

Flexible Spokes. The effect of a heavy ring connected to a shaft by flexible spokes has been considered previously by the author.[10]

The stiffness factor for spokes with a uniform cross-section is

$$C = \frac{12NEIr}{l}\left(r + \frac{1}{3r} - 1\right) \quad \ldots\ldots\ldots\ldots [26]$$

[8] See p. 77, Holzer, loc. cit.
[9] Hütte, vol. I, 1925.
[10] Trans. A.S.M.E., 1928, Paper No. APM-50-14.

where N = number of spokes
E = modulus of elasticity in tension (lb. per sq. in.)
I = moment of inertia of the spoke section (in.^{4})
r = R/l where
R = outer radius of spokes (in.) and
l = length of spokes (in.)

If J is the inertia factor of the rim, and $\psi = \sqrt{J/C}$, then the equivalent inertia factor at the shaft is

$$J' = JT_v(\psi n) \quad [27]$$

Propeller. The inertia factor for a marine propeller is usually increased about 25 per cent to approximate the effect of the entrained water. For a rapid calculation the radius of gyration of a propeller of the usual design may be taken as 0.23 times the propeller diameter. It is of course better practice to calculate the actual inertia factor directly from a drawing of the propeller.

Crankshaft Stiffness. A crankshaft having the general proportions shown in Fig. 9 when turning in its bearings under operating conditions is about 3 per cent stiffer than a uniform shaft having the same length and the same section as the journal. For results of twisting tests on actual shafts refer to the author's previous paper (Trans. A.S.M.E., 1928, Paper No. APM-50-8, loc. cit.).

Changes in Shaft Diameter. When a shaft is composed of two uniform circular sections of different diameters such as shown in Fig. 10, the small diameter d effectively replaces D over the length λ in calculating the stiffness factor.

For D/d greater than 1.5, $\lambda = 0.1\ d$
For D/d between 1.5 and 1, λ is between 0.1 d and 0.
(Refer to Fig. 23, p. 48, Paper No. APM-50-8, loc. cit.)

Shrunk or Clamped Fits. For hubs or other members that are shrunk or clamped on a shaft, the shaft effectively enters the hub from 0.2 d to 0.5 d. The actual value of this effect will vary considerably even for installations that are supposed to be identical. For tightly shrunk-on members the factor may be taken as 0.2 d. For split hubs that are clamped on and keyed, an average value of 0.35 d may be taken.

Keyed Couplings. For keyed couplings that are merely "size and size," the shaft may be considered elastic halfway through the key length, with the limit that the increase in shaft length when considered in this manner shall not be more than one diameter of the shaft.

Flanged Couplings. In determining the elasticity of flanged couplings that are relatively short in an axial direction, the shaft may be considered to penetrate to a depth of one-half the axial length of the flange.

Short Flange. The stiffening effect of a short flange (having an axial length of less than $^1/_4\ d$) such as a thrust ring is assumed to be negligible.

Continuous Sleeve or Liner. In the case of a continuous bronze sleeve shrunk on a propeller shaft, etc., for a considerable length, the diameter of the shaft in calculating its effective stiffness in way of the sleeve is increased by the thickness of the sleeve, i.e., the radius of shaft increased by one-half the thickness of the sleeve. In considering the stiffening effect of sleeves of any length, and particularly shorter ones, two conditions should be recognized: (1) In new installations the sleeves, when carefully shrunk into place, are probably tight and do increase the stiffness of the shaft. (2) In old installations the sleeves may have worked themselves loose and do not increase the stiffness of the shaft. Thus both possibilities have to be considered.

Numerical Example. Consider a 6-cylinder, 2-cycle engine connected to a flywheel and alternator. The principal dimensions of the crankshaft are shown in Fig. 9.

Weight of piston and pin, complete, including cooling gear and oil = 1000 lb.

Weight of connecting rod, complete = 600 lb.

The center of gravity of the rod occurs at 0.38 times the center-line length of the rod from the crankpin center line. The crank-to-connecting-rod ratio = 1/5.

The formula for the inertia factor of the journals is that of a solid cylinder about its own center line; thus

$$\frac{ld^4}{36}\ [\text{lb-in.}^2] \quad [28]$$

where l is the length of the journal, d the diameter, and the weight of steel is taken as 0.2830 lb. per sq. in.

The formula for the inertia factor of the crankpin is that of a solid cylinder about an axis at a distance r from its own center line, thus

$$\frac{ld^2}{4.5}\left(\frac{d^2}{8} + r^2\right)\ [\text{lb-in.}^2] \quad [29]$$

where l is the length of the crankpin, d the diameter, and r the crank radius.

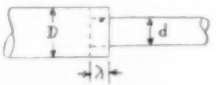


Fig. 10

A close approximation for the inertia factor for the webs of this crank may be obtained by considering each as a rectangular parallelopiped having the same width and thickness, and a height to give the same cross-sectional area. Thus the formula for the webs will be

$$\frac{bdh}{3.534}\left[\frac{b^2 + h^2}{12} + \left(\frac{r}{2}\right)^2\right]\ [\text{lb-in.}^2] \quad [30]$$

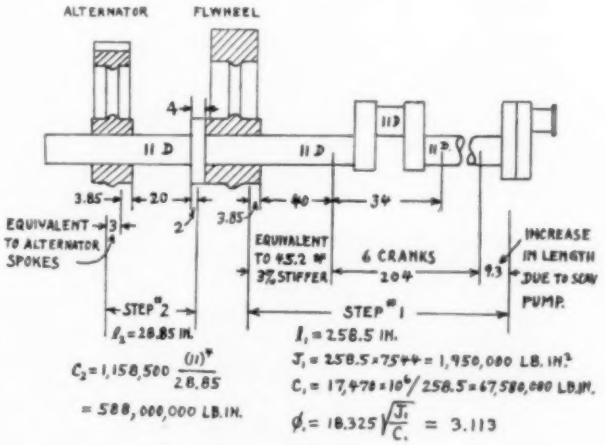


Fig. 11

where b is the breadth, d the thickness, and h the height of the equivalent rectangular section, and r is the crank radius.

Therefore the inertia factors are:

Journal: $12\ \dfrac{(11)^4}{36}$ = ... 4880 lb-in.^{2}

Pin: $10\ \dfrac{(11)^2}{4.5}\left[\dfrac{(11)^2}{8} + (11)^2\right]$ = ... 36,600

Two webs:

$$\frac{14 \times 12 \times 22.5}{3.534}\left[\frac{(14)^2 + (22.5)^2}{12} + (5.5)^2\right] = 94{,}950$$

Piston and connecting rod:

$$(11)^2[0.62 \times 600 + 0.5050(0.38 \times 600 + 1000)] = 120{,}050$$

Total inertia factor per cylinder = sum = 256,480

$$J_w = \frac{256{,}480}{34} = 7544 \text{ lb-in.}^2 \text{ per inch}$$

In engines that have counterweights and other attached rotating mass per cylinder, a careful calculation should be made for each part and added into the total.

Considering the crank 3 per cent stiffer than a uniform shaft having the same length and the same section as the journal, the value of Cl may be calculated from Equation [1]: thus

$$Cl = 1{,}158{,}500(11)^4 1.03 = 17{,}470 \times 10^6 \text{ lb-in.}^2$$

TABLE 7 (FLYWHEEL WITH STIFF SPOKES)

FIRST FREQUENCY CALCULATIONS

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
n	$\psi''_{12}n$	$T_0(2)$	$J''_{02}\times10^{-6}$	$J''_{21}\times10^{-6}$	$-J''_{21}/J_1$	$T^{-1}(6)$	$n=\frac{(7)}{3.113}$
	.09221 n		(3)×5.00	(4)+8.00	−(5)/1.950		
30	2.766	4.601	23.01	31.01	−15.903	92.24	29.63
29.63	2.732	4.229	21.15	29.15	−14.949	92.38	29.68

USING INTERPOLATION FORMULA "I TABLE 4 n = 29.67 V.P.S.

SECOND FREQUENCY CALCULATIONS

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
n	$\phi_{23}n$	$T(2)$	$J'_{21}\times10^{-6}$	$J'_{02}\times10^{-6}$	$-J'_{02}/J_{23}$	$T^{-1}(6)$	$n=\frac{(7)}{.09221}$
	3.113 n		(3)×1.950	(4)+8.00	−(5)/5.00		
43	133.86	−.4454	−.8685	7.132	−1.4264	4.078	44.23
44.23	137.69	−.3788	−.7387	7.261	−1.4522	4.063	44.06

USING INTERPOLATION FORMULA "I TABLE 4 n = 44.07 V.P.S.

TABLE 8 (FLYWHEEL WITH FLEXIBLE SPOKES)

FIRST FREQUENCY CALCULATION

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)
n	$\psi''_{12}n$	$T_0(2)$	$J''_{02}\times10^{-6}$	.07623n	$T_0(5)$	$J_{12}\times10^{-6}$	$J''_{20}\times10^{-6}$	$-J''_{21}/J_1$	$T^{-1}(9)$	$n=\frac{(10)}{3.113}$
	.09221n		(3)×5.00			(6)×8.00	(4)+(7)	−8/1.950		
29.6	2.729	4.199	21.00	2.256	2.086	16.688	37.69	−19.328	91.85	29.51
29.51	2.721	4.122	20.61	2.250	2.074	16.592	37.20	−19.077	91.87	29.51

n = 29.51 V.P.S.

SECOND FREQUENCY CALCULATION

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)
n	$\phi_{12}n$	$T(2)$	$J'_{21}\times10^{-6}$	.07623n	$T_0(5)$	$J_{12}\times10^{-6}$	$J'_{02}\times10^{-6}$	$-J'_{02}/J_{12}$	$T_0^{-1}(9)$	$n=\frac{(10)}{.09221}$
	3.113n		(3)×1.950			(6)×8.00	(4)+(7)	−(8)/5.00		
40	124.52	−.6690	−1.3046	3.049	20.905	163.24	161.94	−32.39	3.174	34.42
34.42	107.15	−.7328	−1.3779	2.624	3.382	27.06	23.68	−4.736	3.441	37.32

USING INTERPOLATION FORMULA "I TABLE 4 n = 36.31 V.P.S.

In the same way that the inertia factor for the crank was calculated, the total inertia factor for the scavenge pump including the shaft up to the center line of the first bearing is 70,000 lb-in.^{2} Therefore 70,000/7544 = 9.3 in. corresponds to the increase in length of the main cranks due to the scavenge pump.

The remainder of the calculations is shown in Fig 11 and the resulting equivalent system is shown in Fig. 12*a* where the alternator inertia factor is given as 5×10^6 lb-in.^{2} and the flywheel inertia is 8×10^6 lb-in.^{2} after making the deduction to make step No. 1 extend to the flywheel. For the first illustration the flywheel spokes will be assumed to be stiff.

The procedure in solving for the natural frequencies is as follows: Remove the flywheel J_{12} and obtain $C_1/C_2 = 0.1149$ and $J_{23}/J_1 = 2.564$. From the curves in Fig. 15, the value of $\phi_1 n$ is 94. Thus Table 5B will be used for the first frequency and the first assumed value of n will be 94/3.113 = 30. For the second frequency Table 5A will be used. The necessary calculations are given in Table 7.

The relative-amplitude curves for the first and second frequencies may then be obtained from Equations [16], [17], and [18]. Thus for $\theta''_{01} = 1$ deg., $\alpha_1 = 1$, and $\gamma_1 = 0$. For the first frequency,

$$\theta''_1 = \cos \frac{3.113 \times 29.67}{258.5} x = \cos 0.3573x$$

and $\theta''_{12}/\theta''_{02} = T_0(0.09221 \times 29.67) = T_0(2.736) = 4.269$

For the second frequency

$$\theta''_1 = \cos \frac{3.113 \times 44.07}{258.5} x = \cos 0.5307x$$

TABLE 9

FIRST FREQUENCY SOLUTION

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
n	$\phi_{23}n$	$T(2)$	$J'_{21}\times10^{-6}$	$J'_{02}\times10^{-6}$	$-J'_{02}/J_{23}$	$T_0^{-1}(6)$	$n=\frac{(7)}{.10980}$
	1.6923 n		(3)×.2104	(4)+.9835	−(5)/.7369		
35	57.98	1.5634	.3289	1.3124	−1.7810	3.907	35.58
35.58	58.93	1.5958	.3358	1.3193	−1.7903	3.903	35.55

THEREFORE BY INTERPOLATION FORMULA "I, TABLE 4, n=35.55 V.P.S.

SECOND FREQUENCY SOLUTION

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
n	$\psi''_{23}n$	$T_0(2)$	$J''_{02}\times10^{-6}$	$J''_{21}\times10^{-6}$	$-J''_{21}/J_1$	$T^{-1}(6)$	$n=\frac{(7)}{1.6922}$
	.10980 n		(3)×.7369	(4)+.9835	−(5)/.2104		
60	6.588	−.2907	−.2142	.7693	−3.656	99.00	60.28
60.28	6.619	−.2872	−.2116	.7719	−3.669	98.97	60.27

THEREFORE BY INTERPOLATION n = 60.27 V.P.S.

TABLE 10

FIRST FREQUENCY SOLUTION

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
n	$\phi_{12}n$	$T(2)$	$J'_{21}\times10^{-6}$	$J'_{02}\times10^{-6}$	$-J'_{02}/J_{23}$	$T_0^{-1}(6)$	$n=\frac{(7)}{.17014}$
	3.309 n		.21775(3)	(4)+.31825	−(5)/.5025		
22	72.80	2.542	.5535	.8718	−1.7349	3.926	23.08
23.08	76.37	3.094	.6737	.9920	−1.9741	3.836	22.56

THEREFORE BY INTERPOLATION FORMULA "I, TABLE 4, n = 22.73 V.P.S

SECOND FREQUENCY SOLUTION

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
n	$\psi''_{23}n$	$T_0(2)$	$J''_{02}\times10^{-6}$	$J''_{21}\times10^{-6}$	$-J''_{21}/J_1$	$T^{-1}(6)$	$n=\frac{(7)}{3.309}$
	.17014 n		.5025(3)	(4)+.31825	−(5)/.21775		
37	6.295	−.3275	−.16457	.15368	−.7058	123.35	37.28
37.28	6.343	−.3209	−.16125	.15700	−.7220	122.86	37.13

THEREFORE BY INTERPOLATION n = 37.18 V.P.S

THIRD FREQUENCY SOLUTION

80	13.611	−.05571	−.02800	.29025	−1.3330	278.79	84.25
84.25	14.334	−.04995	−.02510	.29315	−1.3463	278.71	84.23

THEREFORE BY INTERPOLATION n = 84.23 V.P.S.

and $\theta''_{12}/\theta''_{02} = T_0(0.09221 \times 44.07) = T_0(4.064) = -1.4503$

These curves are shown in Fig. 12*b*.

If the flywheel were considered to have six flexible spokes with a mean elliptical section of 4 and 6$^3/_8$ in. minor and major diameters, with 74$^1/_4$ in. outside diameter and 15$^1/_4$ in. inside diameter, then in Equation [26], $N = 6$, $E = 15 \times 10^6$ lb. per sq. in. for good-grade cast iron, $I = \frac{\pi}{64} 4(6.625)^3 = 57.09$ in.^{4} $R = 37.125$ in., $l = 29.5$ in., $r = 1.2585$, so that

$$C = \frac{12 \times 6 \times 15 \times 10^6 \times 57.09 \times 1.2585}{29.5} \left(1.2585 + \frac{1}{3 \times 1.2585} - 1\right)$$

$= 1376.7 \times 10^6$

Thus $\psi = \sqrt{8/1376.7} = 0.07623$

Therefore $J_{12} = 8 \times 10^6 T_0(0.07623n)$

The frequency solution considering the flexibility of these spokes on the flywheel is given in Table 8.

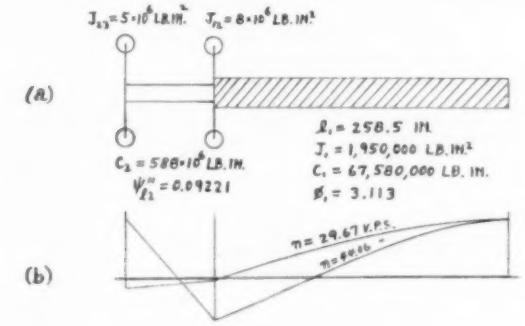


FIG. 12

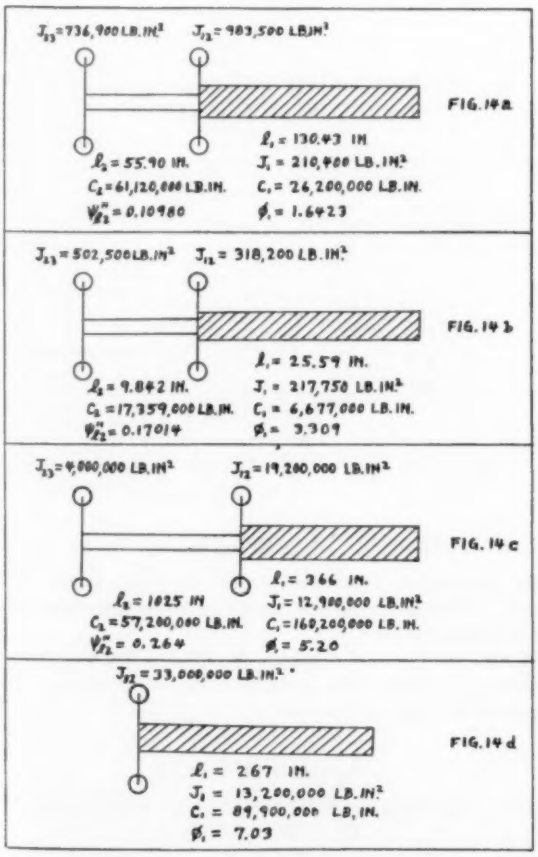


FIG. 14

Comparing the frequencies for the stiff flywheel spokes with those for the flexible ones described, it is seen that first frequency for the stiff spokes is 0.5 per cent and the second frequency is 21.4 per cent higher than the frequencies for the flexible spokes. Thus it is apparent that if the flywheel has flexible spokes it is important to consider their effect.

If a certain-sized engine with its flywheel is to be used in several different installations with various alternators and alternator shafts, curves such as shown in Fig. 13 may be drawn to give the natural frequencies of any likely combination. The curves in Fig. 13 were calculated by reducing the inertia factors to the beginning of step No. 2 for a number of frequencies by using Table 1C, and then solving for the value of the stiffness factor C_2 of the alternator shaft between the flywheel and alternator rim for the various values of the alternator inertia J_{12} by using Table 3A. Similar curves may be constructed to show the effect of different flywheels.

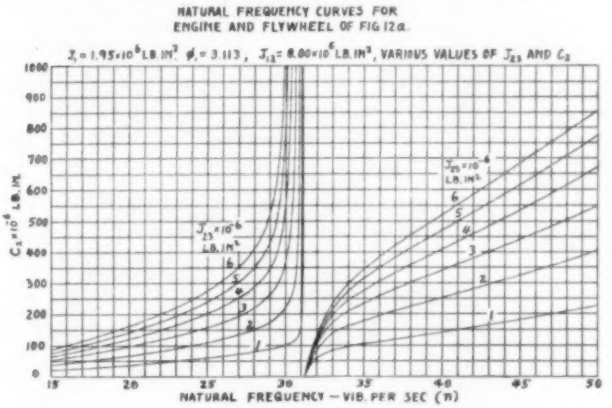


FIG. 13

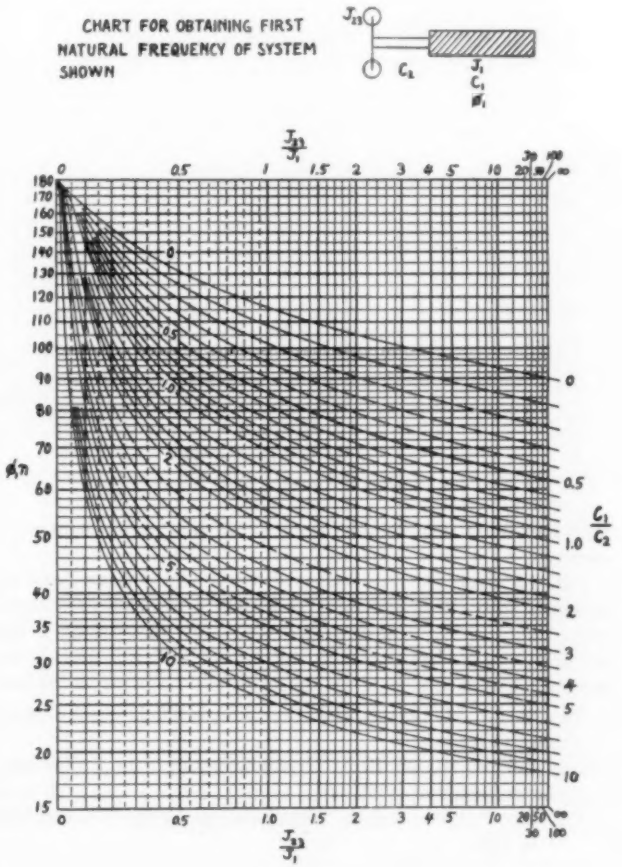


FIG. 15

Comparison With Other Methods

The relative ease of the method of solution described in this

TABLE 11

FIRST FREQUENCY SOLUTION

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
n	$\phi_1 n$	$T(2)$	$J'_{01} \times 10^{-6}$	$J'_{12} \times 10^{-6}$	$-J'_{12}/J_{01}$	$T_0^{-1}(6)$	n
	$5.20\,n$		(5)×12.90	(5)+19.2	$-(5)/4.00$		$\frac{(7)}{.264}$
12	62.4	1.756	22.65	41.85	-10.46	3.273	12.40
12.40	64.5	1.862	24.02	43.22	-10.80	3.268	12.38

THEREFORE $n = 12.38$ V.P.S.

paper will be illustrated by comparing it with other methods that have been published. In making this comparison the original equivalent system and the original calculations required for its solution will not be repeated here. To complete the comparison,

PER CENT ERROR IN NATURAL FREQUENCY IN SOLVING FIG. b INSTEAD OF FIG. a.

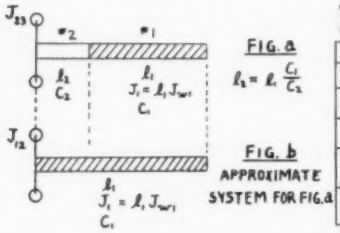


FIG. a

$l_2 = l_2 \frac{C_1}{C_2}$

FIG. b

APPROXIMATE SYSTEM FOR FIG. a

VALUES FOR FIG. b IN TERMS OF THOSE IN FIG. a

FIG. b	FIG. a
J_{w1}	J_{w1}
l_1	$l_1 + l_2$
J_1	$J_{w1}(l_1 + l_2)$
C_1	$\frac{l_1}{l_1 + l_2} C_1$
J_{12}	$J_{23} - J_{w1} l_2$

CURVES INDICATE PER CENT ERROR

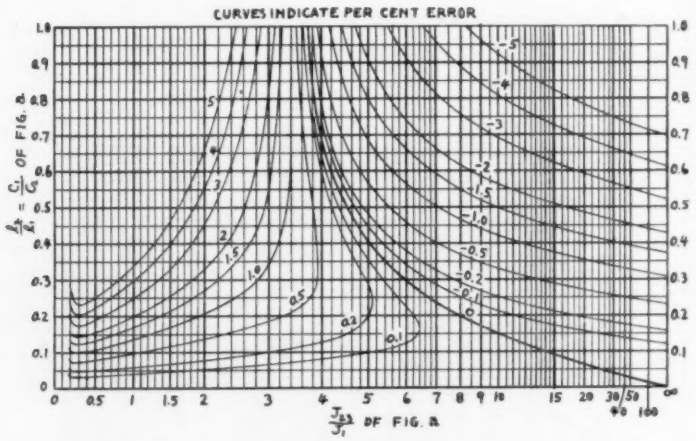


FIG. 16

Solution for Fig. 14b. A reference to Fig. 15 shows that Table 5A should be used for the first frequency. The solution for the first three frequencies is given in Table 10. The third frequency with two nodes in the engine is rarely observed in practice. The results of this solution compared with those given by Wydler are: first frequency, 0.2 per cent lower; second frequency, 0.5 per cent higher; third frequency, 2.4 per cent higher.

Solution for Fig. 14c. The solution is given in Table 11. The result obtained is 2 per cent higher than the value given by Purday. This is due to the approximate method used by Purday to obtain his results.

PER CENT ERROR IN NATURAL FREQUENCY IN SOLVING FIG. b INSTEAD OF FIG. a

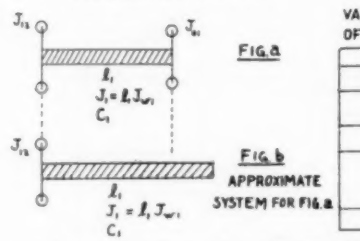


FIG. a

FIG. b

APPROXIMATE SYSTEM FOR FIG. a

VALUES FOR FIG. b IN TERMS OF THOSE IN FIG. a

FIG. b	FIG. a
J_{w1}	J_{w1}
l_1	$l_1 + \frac{J_{01}}{J_{w1}}$
J_1	$J_1 + J_{01}$
C_1	$\frac{l_1}{l_1 + \frac{J_{01}}{J_{w1}}} C_1$
J_{12}	J_{12}

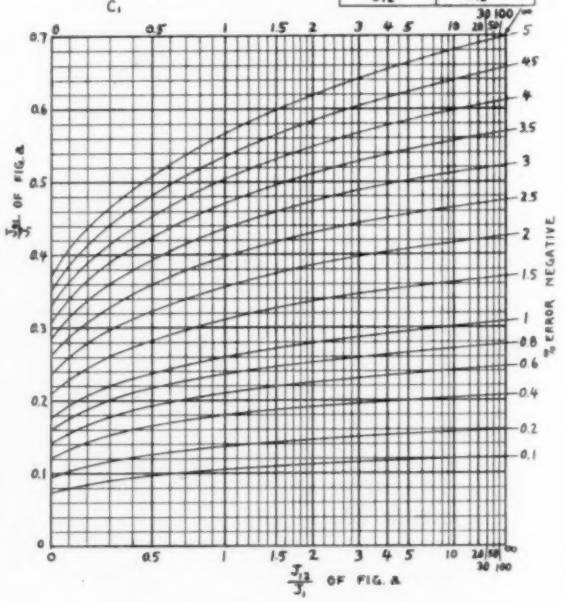


FIG. 17

Solution for Fig. 14d. Since $-J_{12}/J_1 = -2.5$, and $T^{-1}(-2.5) = 102.6$, $n = 102.6/7.03 = 14.60$ v.p.s. $= 876$ v.p.m. Purday obtains 875 v.p.m. in this case.

Appendix No. 1

SIMPLE PROOF FOR THE PRINCIPAL EQUATIONS USED IN THE SOLUTION

a Consider an irregular shaft system vibrating torsionally in simple harmonic motion. If a concentrated mass occurs at the beginning of the system having an inertia factor J_{01} and is connected to a section A (end of first step) by an elastic shaft having no mass, the twisting moment due to the inertia of the mass is

$$M = \frac{J_{01}}{g} \frac{d^2\theta_{01}}{dt^2}$$

or, since it is simple harmonic motion,

therefore, reference should be made to the original source indicated.

Fig. 14 gives four equivalent systems which have been rearranged into an equivalent system where the engine is represented by a uniform shaft with mass. The dimensions of the units have also been transformed into the inch-pound-second system. For each system references should be made as follows:

Fig. 14*a:* P. 34, Holzer, "Die Berechnung der Drehschwingungen," 1921

Fig. 14*b:* P. 20, Wydler, "Drehschwingungen in Kolbenmaschenenanlagen," 1922

Fig. 14*c:* P. 109, Purday, "Diesel Engine Design," Third Edition, 1928

Fig. 14*d:* P. 347, Purday.

Solution for Fig. 14a. Since $C_1/C_2 = 0.43$ and $J_{23}/J_1 = 3.5$, a reference to Fig. 15 shows that for the first frequency Table 5A should be used. For the second frequency Table 5B will be used. The solution is given in Table 9.

The results of this solution compared with those given by Holzer are: first frequency, the same; and second frequency, 0.84 per cent higher.

$$M = -\frac{J_{01}}{g} p^2 \theta_{01}$$

where $p = 2\pi n$.

The twisting moment in the shaft is

$$M = C_1(\theta_{11} - \theta_{01})$$

These two values of M must be equal for equilibrium, so that

$$\frac{\theta_{11}}{\theta_{01}} = 1 - \frac{J_{01}}{gC_1} p^2$$

If a concentrated mass is assumed to be placed at section A to give the same twisting moment at this section, it must have an inertia factor such that

$$M = \frac{J'_{11}}{g} \frac{d^2\theta_{11}}{dt^2} = -\frac{J'_{11}}{g} p^2\theta_{11} = C_1(\theta_{11} - \theta_{01}) = -\frac{J_{01}}{g} p^2\theta_{01}$$

Thus

$$\frac{J'_{11}}{J_{01}} = \frac{1}{1 - \frac{J_{01}}{gC_1} p^2} = \frac{\theta_{01}}{\theta_{11}}$$

b Consider the case similar to that just given except that the shaft is uniform and has mass.

For equilibrium, since the only external forces assumed to be in action are due to the inertia forces, it is necessary that the sum of all the twisting moments due to the inertia forces of the masses on one side of any section is equal to the internal resisting moment in the shaft.

The twisting moment due to the inertia of the mass at the beginning is

$$M_{01} = \frac{J_{01}}{g} \frac{d^2\theta_{01}}{dt^2} = -\frac{J_{01}}{g} p^2\theta_{01}$$

The resisting moment in the shaft at any section is

$$M_1 = C_1 l_1 \frac{d\theta_1}{dx}$$

Hence at any point in the shaft between the concentrated mass at the beginning and the section A,

$$\int_0^x -\frac{J_1}{gl_1} p^2\theta_1 dx - \frac{J_{01}}{g} p^2\theta_{01} = C_1 l_1 \frac{d\theta_1}{dx}$$

or differentiating with respect to x,

$$-\frac{J_1}{gl_1} p^2\theta_1 = C_1 l_1 \frac{d^2\theta_1}{dx^2}$$

$$\frac{d^2\theta_1}{dx^2} = -\frac{J_1 p^2}{gC_1 l_1^2} \theta_1$$

The general solution of this differential equation is

$$\theta_1 = a \cos\left(\sqrt{\frac{J_1}{gC_1}}\, p \frac{x}{l_1} + b\right)$$

where a and b are the constants of intergration and are evaluated by the end conditions. Since it is more convenient in performing numerical computations to deal with angles in degrees, the equation for θ_1 will be stated in degrees. In this way

$$\theta^\circ_1 = \alpha'_1 \cos\left(\phi_1 n \frac{x}{l_1} + \gamma'_1\right)$$

where $\phi_1 n = \dfrac{360\sqrt{J_1}}{\sqrt{gC_1}}$ n, α'_1 and γ'_1 are all angles in degrees.

Differentiating θ_1, and putting the result into M_1 gives the twisting moment in the shaft at any section. Putting $x = 0$ in M_1

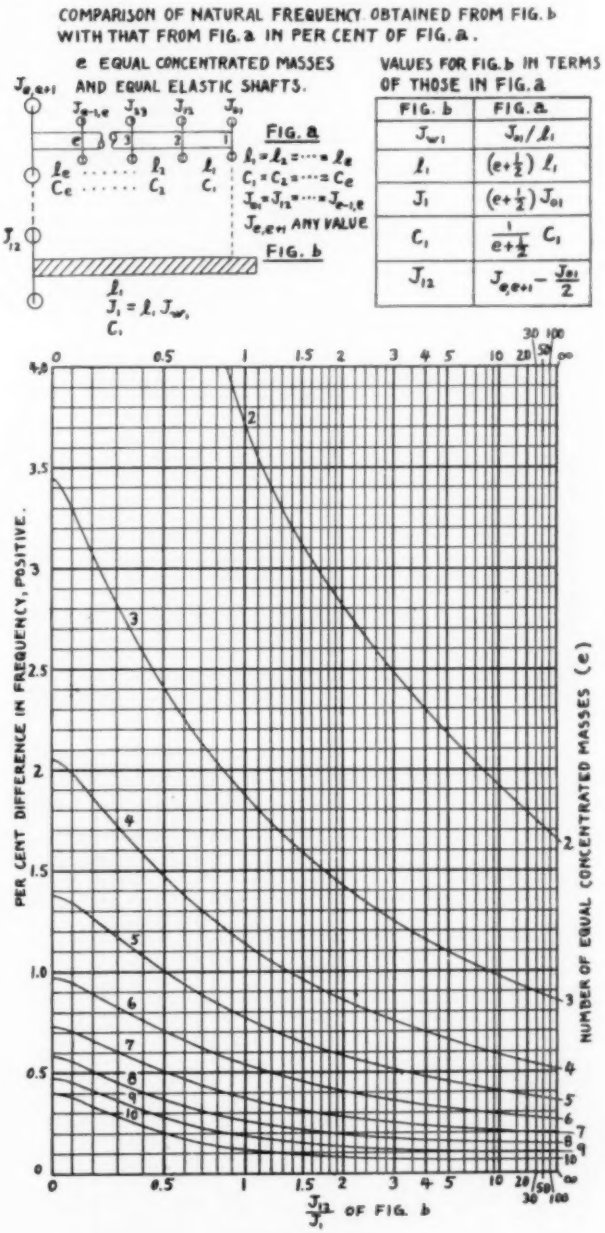


FIG. 18

gives M_{01}, which may be equated to the previous value for M_{01}. In this way it will be found that

$$\tan \gamma'_1 = \frac{\pi}{180} \phi_1 n \frac{J_{01}}{J_1}$$

If a concentrated mass is assumed to be placed at section A to give the same twisting moment at this section as stated by M_1 when $x = l_1$, it must have an inertia factor such that

$$M_{l1} = -\frac{J'_{l1}}{g} p^2\theta_{l1}$$

TABLE 12

0 to 2.700 — $T_o(\psi\eta)$ for Positive J

$\psi\eta$	0	.01	.02	.03	.04	.05	.06	.07	.08	.09	Ave D
.0	1.0000	1.000,0	1.000,0	1.000,1	1.000,2	1.000,3	1.000,4	1.000,5	1.000,7	1.000,8	1
.1	1.001,0	1.001,2	1.001,3	1.001,7	1.002,0	1.002,3	1.002,6	1.003,0	1.003,3	1.003,7	3
.2	1.0041	1.0045	1.0050	1.0054	1.0059	1.0064	1.0070	1.0075	1.0081	1.0087	5
.3	1.0093	1.0099	1.0106	1.0113	1.0120	1.0127	1.0134	1.0142	1.0150	1.0158	7
.4	1.0166	1.0175	1.0184	1.0193	1.0202	1.0212	1.0221	1.0231	1.0241	1.0252	9
.5	1.0262	1.0273	1.0284	1.0296	1.0307	1.0319	1.0331	1.0344	1.0356	1.0369	12
.6	1.0382	1.0396	1.0409	1.0423	1.0437	1.0452	1.0466	1.0481	1.0497	1.0512	14
.7	1.0528	1.0544	1.0560	1.0577	1.0593	1.0611	1.0628	1.0646	1.0664	1.0682	17
.8	1.0701	1.0719	1.0739	1.0758	1.0778	1.0798	1.0819	1.0839	1.0860	1.0882	20
.9	1.0903	1.0926	1.0948	1.0971	1.0994	1.1017	1.1041	1.1065	1.1089	1.1114	23
1.0	1.1140	1.1165	1.1191	1.1217	1.1244	1.1271	1.1299	1.1327	1.1355	1.1384	27
1.1	1.1413	1.1442	1.1472	1.1502	1.1533	1.1565	1.1596	1.1628	1.1661	1.1694	31
1.2	1.1728	1.1762	1.1796	1.1831	1.1867	1.1902	1.1939	1.1976	1.2014	1.2052	36
1.3	1.2090	1.2129	1.2169	1.2209	1.2250	1.2292	1.2334	1.2376	1.2419	1.2463	41
1.4	1.2508	1.2553	1.2599	1.2645	1.2692	1.2740	1.2789	1.2838	1.2888	1.2938	48
1.5	1.2990	1.3042	1.3093	1.3149	1.3203	1.3259	1.3315	1.3372	1.3430	1.3488	55
1.6	1.3548	1.3608	1.3670	1.3732	1.3796	1.3860	1.3925	1.3992	1.4059	1.4128	64
1.7	1.4197	1.4268	1.4340	1.4413	1.4487	1.4562	1.4639	1.4716	1.4795	1.4876	75
1.8	1.4958	1.5041	1.5125	1.5211	1.5298	1.5387	1.5478	1.5570	1.5663	1.5758	(9)
1.9	1.5855	1.5959	1.6056	1.6156	1.6260	1.6366	1.6474	1.6584	1.6696	1.6810	(11)
2.0	1.6926	1.7044	1.7163	1.7288	1.7413	1.7541	1.7671	1.7804	1.7940	1.8078	(13)
2.1	1.8219	1.8363	1.8510	1.8661	1.8814	1.8971	1.9131	1.9294	1.9461	1.9632	(16)
2.2	1.981	1.999	2.017	2.035	2.053	2.072	2.092	2.112	2.132	2.152	20
2.3	2.173	2.193	2.215	2.237	2.259	2.282	2.305	2.329	2.353	2.378	24
2.4	2.403	2.429	2.455	2.482	2.509	2.537	2.565	2.594	2.624	2.654	29
$\psi\eta$	0	.01	.02	.03	.04	.05	.06	.07	.08	.09	

$\psi\eta$	0	.001	.002	.003	.004	.005	.006	.007	.008	.009	
2.50	2.773	2.777	2.781	2.785	2.789	2.793	2.797	2.801	2.805	2.809	4
2.51	2.813	2.817	2.821	2.825	2.829	2.833	2.837	2.842	2.846	2.850	4
2.52	2.854	2.858	2.863	2.867	2.871	2.875	2.880	2.884	2.888	2.892	4
2.53	2.897	2.901	2.906	2.910	2.914	2.919	2.923	2.928	2.932	2.937	4
2.54	2.941	2.945	2.950	2.955	2.959	2.964	2.968	2.973	2.977	2.982	5
2.55	2.987	2.991	2.996	3.001	3.005	3.010	3.015	3.020	3.024	3.029	5
2.56	3.034	3.039	3.044	3.049	3.054	3.058	3.063	3.068	3.073	3.078	5
2.57	3.083	3.088	3.093	3.098	3.103	3.108	3.114	3.119	3.124	3.129	5
2.58	3.134	3.139	3.145	3.150	3.155	3.160	3.166	3.171	3.176	3.182	5
2.59	3.187	3.192	3.198	3.203	3.209	3.214	3.220	3.225	3.231	3.236	5
2.60	3.242	3.247	3.253	3.259	3.264	3.270	3.276	3.281	3.287	3.293	6
2.61	3.299	3.305	3.310	3.316	3.322	3.328	3.334	3.340	3.346	3.351	6
2.62	3.358	3.364	3.370	3.376	3.382	3.389	3.395	3.401	3.407	3.413	6
2.63	3.420	3.426	3.432	3.439	3.445	3.451	3.458	3.464	3.471	3.477	6
2.64	3.484	3.491	3.497	3.504	3.510	3.517	3.524	3.531	3.537	3.544	7
2.65	3.551	3.558	3.565	3.572	3.578	3.585	3.592	3.599	3.607	3.614	7
2.66	3.621	3.628	3.635	3.642	3.650	3.657	3.664	3.671	3.679	3.686	7
2.67	3.694	3.701	3.709	3.716	3.724	3.731	3.739	3.747	3.754	3.762	8
2.68	3.770	3.778	3.786	3.793	3.801	3.809	3.817	3.825	3.833	3.841	8
2.69	3.850	3.858	3.866	3.874	3.882	3.891	3.899	3.908	3.916	3.925	8
$\psi\eta$	0	.001	.002	.003	.004	.005	.006	.007	.008	.009	Ave D

2.700 to 3.200 — $T_o(\psi\eta)$ for Positive J

$\psi\eta$	0	.001	.002	.003	.004	.005	.006	.007	.008	.009	Ave D
2.70	3.933	3.942	3.950	3.959	3.968	3.976	3.985	3.994	4.003	4.012	9
2.71	4.021	4.030	4.039	4.048	4.057	4.066	4.075	4.084	4.094	4.103	9
2.72	4.112	4.122	4.131	4.141	4.150	4.160	4.170	4.179	4.189	4.199	10
2.73	4.209	4.219	4.229	4.239	4.249	4.259	4.269	4.279	4.290	4.300	10
2.74	4.310	4.321	4.331	4.342	4.353	4.363	4.374	4.385	4.396	4.406	11
2.75	4.417	4.428	4.439	4.451	4.462	4.473	4.484	4.496	4.507	4.519	11
2.76	4.530	4.542	4.553	4.565	4.577	4.589	4.601	4.613	4.625	4.637	12
2.77	4.649	4.662	4.674	4.686	4.699	4.711	4.724	4.737	4.750	4.762	13
2.78	4.776	4.788	4.801	4.815	4.828	4.841	4.855	4.868	4.882	4.895	13
2.79	4.909	4.923	4.937	4.951	4.965	4.979	4.993	5.007	5.022	5.036	14
2.80	5.051	5.065	5.080	5.095	5.110	5.125	5.140	5.155	5.171	5.186	15
2.81	5.201	5.217	5.233	5.249	5.264	5.280	5.297	5.313	5.329	5.346	16
2.82	5.362	5.379	5.395	5.412	5.429	5.446	5.464	5.481	5.498	5.516	17
2.83	5.534	5.551	5.569	5.587	5.605	5.623	5.642	5.661	5.679	5.698	18
2.84	5.717	5.736	5.755	5.775	5.794	5.814	5.833	5.853	5.873	5.894	20
2.85	5.914	5.934	5.955	5.976	5.997	6.018	6.039	6.060	6.082	6.104	21
2.86	6.125	6.148	6.170	6.192	6.215	6.237	6.260	6.283	6.307	6.330	23
2.87	6.354	6.377	6.401	6.426	6.450	6.474	6.499	6.524	6.549	6.575	25
2.88	6.600	6.626	6.652	6.678	6.705	6.731	6.758	6.785	6.813	6.840	27
2.89	6.868	6.896	6.924	6.953	6.981	7.010	7.040	7.069	7.099	7.129	29
2.90	7.159	7.190	7.220	7.252	7.283	7.315	7.346	7.379	7.411	7.444	32
2.91	7.477	7.511	7.544	7.578	7.613	7.648	7.683	7.718	7.754	7.790	35
2.92	7.826	7.862	7.900	7.938	7.975	8.014	8.052	8.091	8.131	8.171	38
2.93	8.211	8.251	8.292	8.334	8.376	8.418	8.461	8.504	8.548	8.592	42
2.94	8.637	8.682	8.727	8.773	8.820	8.867	8.915	8.963	9.012	9.061	47
2.95	9.111	9.161	9.212	9.264	9.316	9.368	9.422	9.476	9.530	9.586	53
2.96	9.642	9.698	9.756	9.819	9.872	9.932	9.992	10.053	10.113	10.177	59
2.97	10.291	10.303	10.370	10.436	10.502	10.570	10.638	10.708	10.778	10.849	68
2.98	10.921	10.993	11.069	11.144	11.221	11.298	11.376	11.456	11.537	11.619	77
2.99	11.702	11.786	11.872	11.959	12.047	12.137	12.228	12.320	12.414	12.509	(9)
3.00	12.606	12.704	12.804	12.906	13.009	13.114	13.220	13.329	13.439	13.551	(10)
3.01	13.665	13.781	13.899	14.019	14.141	14.266	14.392	14.521	14.653	14.787	(12)
3.02	14.923	15.062	15.203	15.348	15.495	15.643	15.798	15.956	16.118	16.285	(15)
3.03	16.941	16.611	16.784	16.960	17.141	17.325	17.513	17.706	17.903	18.104	(18)
3.04	18.311	18.522	18.738	18.959	19.185	19.417	19.653	19.894	20.139	20.389	(23)
3.05	20.67	20.94	21.22	21.50	21.79	22.09	22.40	22.72	23.05	23.39	(3)
3.06	23.73	24.09	24.46	24.84	25.24	25.64	26.06	26.49	26.93	27.38	(9)
3.07	27.88	28.38	28.90	29.43	29.99	30.56	31.16	31.79	32.44	33.11	(6)
3.08	33.82	34.55	35.32	36.13	36.97	37.85	38.78	39.75	40.76	41.86	(9)
3.09	42.99	44.20	45.47	46.81	48.24	49.76	51.38	53.11	54.96	56.99	(17)
3.10	59.08	61.38	63.87	66.56	69.50	72.71	76.23	80.11	84.41	89.20	
3.11	99.57	100.6	107.5	115.9	129.6	135.3	198.1	163.5	182.6	206.6	
3.12	238.0	280.7	342.0	437.4	607.5	993.1	2721.				
3.13	THE FOLLOWING VALUES ARE ALL NEGATIVE							3674.	1096.	699.3	
3.13	498.1	352.0	287.9	293.0	210.3	185.3	165.6	149.7	136.6	125.6	
3.14	116.19	108.12	101.89	96.92	89.96	84.59	80.23	76.29	72.71	69.96	
3.15	66.99	63.75	61.29	58.91	56.75	54.75	52.88	51.13	49.50	47.96	
3.16	46.52	45.16	43.86	42.67	41.52	40.49	39.90	38.92	37.99	36.60	(11)
3.17	35.75	34.99	34.17	33.42	32.71	32.03	31.38	30.75	30.15	29.57	(7)
3.18	29.01	28.48	27.96	27.46	26.98	26.51	26.06	25.62	25.20	24.79	(5)
3.19	24.40	24.02	23.65	23.29	22.94	22.60	22.27	21.95	21.64	21.34	(3)
$\psi\eta$	0	.001	.002	.003	.004	.005	.006	.007	.008	.009	Ave D

3.200 to 3.700 — $-T_o(\psi\eta)$ for Positive J

$\psi\eta$	.000	.001	.002	.003	.004	.005	.006	.007	.008	.009	Ave D
3.20	21.09	20.76	20.98	20.21	19.99	19.89	19.99	19.19	18.95	18.72	26
3.21	18.492	18.270	18.053	17.892	17.633	17.933	17.235	17.092	16.853	16.668	(20)
3.22	16.987	16.309	16.136	15.966	15.800	15.637	15.977	15.321	15.167	15.017	(16)
3.23	14.869	14.729	14.593	14.493	14.367	14.172	14.091	13.911	13.789	13.666	(13)
3.24	13.537	13.417	13.298	13.182	13.069	12.955	12.845	12.736	12.629	12.529	(11)
3.25	12.421	12.319	12.219	12.120	12.023	11.928	11.839	11.749	11.650	11.560	(10)
3.26	11.472	11.385	11.299	11.214	11.131	11.049	10.968	10.888	10.809	10.732	(8)
3.27	10.655	10.580	10.505	10.432	10.360	10.288	10.218	10.148	10.080	10.012	(7)
3.28	9.945	9.879	9.814	9.750	9.686	9.629	9.569	9.501	9.440	9.381	(6)
3.29	9.322	9.266	9.207	9.150	9.094	9.038	8.989	8.929	8.876	8.823	55
3.30	8.771	8.719	8.668	8.618	8.568	8.519	8.470	8.421	8.379	8.327	49
3.31	8.280	8.239	8.188	8.143	8.098	8.059	8.019	7.967	7.929	7.882	44
3.32	7.890	7.798	7.757	7.716	7.676	7.636	7.597	7.558	7.519	7.481	40
3.33	7.443	7.405	7.368	7.331	7.295	7.259	7.223	7.187	7.152	7.117	36
3.34	7.083	7.049	7.015	6.982	6.948	6.916	6.883	6.851	6.819	6.787	33
3.35	6.756	6.724	6.694	6.663	6.633	6.603	6.573	6.543	6.514	6.485	30
3.36	6.456	6.428	6.399	6.371	6.389	6.316	6.289	6.261	6.235	6.208	28
3.37	6.181	6.155	6.129	6.103	6.078	6.052	6.027	6.002	5.977	5.953	25
3.38	5.928	5.904	5.880	5.856	5.833	5.809	5.786	5.763	5.740	5.717	23
3.39	5.695	5.672	5.650	5.628	5.606	5.584	5.563	5.541	5.520	5.499	22
3.40	5.478	5.457	5.436	5.416	5.396	5.375	5.355	5.335	5.316	5.296	20
3.41	5.277	5.257	5.238	5.219	5.200	5.181	5.162	5.144	5.125	5.107	19
3.42	5.089	5.071	5.053	5.035	5.017	5.000	4.982	4.965	4.948	4.931	18
3.43	4.914	4.897	4.880	4.863	4.847	4.830	4.814	4.798	4.782	4.766	16
3.44	4.750	4.734	4.718	4.703	4.687	4.672	4.656	4.641	4.626	4.611	15
3.45	4.596	4.581	4.566	4.552	4.537	4.522	4.508	4.494	4.479	4.465	15
3.46	4.451	4.437	4.423	4.410	4.396	4.382	4.369	4.355	4.342	4.328	14
3.47	4.315	4.302	4.289	4.276	4.263	4.250	4.237	4.224	4.212	4.199	13
3.48	4.187	4.174	4.162	4.149	4.137	4.125	4.113	4.101	4.089	4.077	12
3.49	4.065	4.053	4.042	4.030	4.019	4.007	3.996	3.984	3.973	3.962	12
3.50	3.950	3.939	3.928	3.917	3.906	3.895	3.884	3.874	3.863	3.852	11
3.51	3.842	3.831	3.821	3.810	3.800	3.789	3.779	3.769	3.759	3.748	10
3.52	3.738	3.728	3.718	3.708	3.698	3.689	3.679	3.669	3.659	3.650	10
3.53	3.640	3.631	3.621	3.612	3.602	3.593	3.584	3.574	3.565	3.556	9
3.54	3.547	3.538	3.529	3.520	3.511	3.502	3.493	3.484	3.475	3.467	9
3.55	3.458	3.449	3.441	3.432	3.423	3.415	3.406	3.398	3.390	3.381	9
3.56	3.373	3.365	3.357	3.348	3.340	3.332	3.324	3.316	3.308	3.300	8
3.57	3.292	3.284	3.276	3.268	3.261	3.253	3.245	3.237	3.230	3.222	8
3.58	3.215	3.207	3.200	3.192	3.185	3.177	3.170	3.162	3.155	3.148	7
3.59	3.141	3.133	3.126	3.119	3.112	3.105	3.098	3.091	3.084	3.077	7
3.60	3.070	3.063	3.056	3.049	3.042	3.035	3.029	3.022	3.015	3.008	7
3.61	3.002	2.995	2.988	2.982	2.975	2.969	2.962	2.956	2.949	2.943	7
3.62	2.937	2.930	2.924	2.917	2.911	2.905	2.899	2.892	2.886	2.880	6
3.63	2.874	2.868	2.862	2.856	2.850	2.844	2.838	2.832	2.826	2.820	6
3.64	2.814	2.808	2.802	2.796	2.790	2.785	2.779	2.773	2.767	2.762	6
3.65	2.756	2.750	2.745	2.739	2.733	2.728	2.722	2.717	2.711	2.706	6
3.66	2.700	2.695	2.689	2.684	2.679	2.673	2.668	2.663	2.657	2.652	5
3.67	2.647	2.641	2.636	2.631	2.626	2.621	2.616	2.610	2.605	2.600	5
3.68	2.595	2.590	2.585	2.580	2.575	2.570	2.565	2.560	2.555	2.550	5
3.69	2.545	2.540	2.536	2.531	2.526	2.521	2.516	2.511	2.507	2.502	5
$\psi\eta$	.000	.001	.002	.003	.004	.005	.006	.007	.008	.009	Ave D

3.700 to 7.50 — $-T_o(\psi\eta)$ for Positive J

$\psi\eta$	.000	.001	.002	.003	.004	.005	.006	.007	.008	.009	Ave D
3.70	2.497	2.493	2.488	2.483	2.478	2.474	2.469	2.465	2.460	2.455	5
3.71	2.451	2.446	2.442	2.437	2.433	2.428	2.424	2.419	2.415	2.410	4
3.72	2.406	2.402	2.397	2.393	2.389	2.384	2.380	2.376	2.371	2.367	4
3.73	2.363	2.358	2.354	2.350	2.346	2.342	2.337	2.333	2.329	2.325	4
3.74	2.321	2.317	2.313	2.308	2.304	2.300	2.296	2.292	2.288	2.284	4
3.75	2.280	2.276	2.272	2.268	2.264	2.260	2.257	2.253	2.249	2.245	4
3.76	2.241	2.237	2.233	2.229	2.226	2.222	2.218	2.214	2.210	2.207	4
3.77	2.203	2.199	2.195	2.192	2.188	2.184	2.181	2.177	2.173	2.170	4
3.78	2.166	2.162	2.159	2.155	2.152	2.148	2.145	2.141	2.137	2.134	4
3.79	2.130	2.127	2.123	2.120	2.116	2.113	2.109	2.106	2.103	2.099	3
$\psi\eta$	.000	.001	.002	.003	.004	.005	.006	.007	.008	.009	

$\psi\eta$	.00	.01	.02	.03	.04	.05	.06	.07	.08	.09	
3.8	2.0957	2.0621	2.0294	1.9977	1.9668	1.9369	1.9077	1.8794	1.8518	1.8249	(30)
3.9	1.7988	1.7733	1.7485	1.7242	1.7006	1.6776	1.6552	1.6332	1.6118	1.5909	(23)
4.0	1.5705	1.5505	1.5310	1.5120	1.4933	1.4751	1.4573	1.4398	1.4227	1.4060	(18)
4.1	1.3896	1.3736	1.3579	1.3425	1.3274	1.3127	1.2982	1.2840	1.2700	1.2563	(15)
4.2	1.2430	1.2298	1.2169	1.2043	1.1918	1.1796	1.1676	1.1558	1.1443	1.1329	(12)
4.3	1.1217	1.1108	1.1000	1.0894	1.0790	1.0687	1.0586	1.0487	1.0390	1.0294	(10)
4.4	1.0199	1.0106	1.0015	.9925	.9836	.9749	.9663	.9579	.9495	.9413	(9)
4.5	.9333	.9253	.9175	.9097	.9021	.8946	.8872	.8799	.8727	.8656	(8)
4.6	.8587	.8518	.8450	.8383	.8317	.8251	.8187	.8124	.8061	.7999	65
4.7	.7938	.7878	.7819	.7760	.7702	.7645	.7588	.7533	.7478	.7423	57
4.8	.7370	.7317	.7264	.7213	.7161	.7111	.7061	.7012	.6963	.6915	51
4.9	.6867	.6820	.6774	.6728	.6683	.6638	.6593	.6550	.6506	.6463	45
5.0	.6421	.6379	.6337	.6296	.6256	.6216	.6176	.6137	.6098	.6059	40
5.1	.6022	.5984	.5947	.5910	.5873	.5837	.5801	.5766	.5731	.5696	36
5.2	.5661	.5628	.5595	.5561	.5528	.5496	.5464	.5432	.5400	.5369	33
5.3	.5338	.5307	.5276	.5246	.5216	.5187	.5157	.5128	.5100	.5071	30
5.4	.5043	.5015	.4987	.4960	.4933	.4906	.4879	.4852	.4826	.4800	27
5.5	.4774	.4749	.4724	.4699	.4674	.4649	.4625	.4600	.4576	.4553	25
5.6	.4529	.4506	.4482	.4459	.4437	.4414	.4391	.4369	.4347	.4325	23
5.7	.4304	.4282	.4261	.4240	.4219	.4198	.4177	.4157	.4136	.4116	21
5.8	.4096	.4076	.4057	.4037	.4018	.3999	.3980	.3961	.3942	.3923	19
5.9	.3905	.3886	.3868	.3850	.3832	.3815	.3797	.3779	.3762	.3745	18
6.0	.3728	.3711	.3694	.3677	.3660	.3644	.3628	.3611	.3595	.3579	16
6.1	.3563	.3547	.3532	.3516	.3501	.3485	.3470	.3455	.3440	.3425	15
6.2	.3410	.3396	.3381	.3367	.3353	.3338	.3324	.3310	.3296	.3282	14
6.3	.3268	.3254	.3240	.3227	.3213	.3200	.3187	.3174	.3161	.3148	13
6.4	.3135	.3122	.3109	.3096	.3084	.3071	.3059	.3047	.3034	.3022	12
6.5	.3010	.2998	.2986	.2974	.2963	.2951	.2939	.2928	.2916	.2905	12
6.6	.2893	.2882	.2871	.2860	.2849	.2838	.2827	.2816	.2805	.2795	11
6.7	.2784	.2773	.2763	.2752	.2742	.2732	.2721	.2711	.2701	.2691	10
6.8	.2681	.2671	.2661	.2651	.2641	.2632	.2622	.2612	.2603	.2593	10
6.9	.2584	.2574	.2565	.2556	.2546	.2537	.2528	.2519	.2510	.2501	9
7.0	.2492	.2483	.2474	.2466	.2457	.2448	.2440	.2431	.2423	.2414	9
7.1	.2406	.2397	.2389	.2381	.2372	.2364	.2356	.2348	.2340	.2332	8
7.2	.2324	.2316	.2308	.2300	.2292	.2285	.2277	.2269	.2262	.2254	8
7.3	.2246	.2239	.2231	.2224	.2217	.2209	.2202	.2195	.2187	.2180	7
7.4	.2173	.2166	.2159	.2152	.2145	.2138	.2131	.2124	.2117	.2110	7
$\psi\eta$	.00	.01	.02	.03	.04	.05	.06	.07	.08	.09	Ave D

TABLE 12 (*continued*)

7.50 to 30 — $T_o(\psi n)$ for Positive J

ψn	.00	.01	.02	.03	.04	.05	.06	.07	.08	.09	Avg D
7.5	.21034	.20966	.20899	.20832	.20765	.20699	.20633	.20567	.20502	.20437	66
7.6	.20372	.20308	.20244	.20180	.20117	.20053	.19991	.19926	.19866	.19806	63
7.7	.19793	.19681	.19620	.19560	.19499	.19439	.19380	.19320	.19261	.19202	60
7.8	.19143	.19085	.19027	.18969	.18912	.18854	.18797	.18741	.18684	.18628	57
7.9	.18572	.18517	.18461	.18406	.18351	.18297	.18243	.18188	.18135	.18081	55
8.0	.18028	.17975	.17922	.17869	.17817	.17765	.17713	.17661	.17610	.17559	52
8.1	.17508	.17457	.17407	.17357	.17307	.17257	.17207	.17158	.17109	.17060	50
8.2	.17011	.16963	.16915	.16867	.16819	.16771	.16724	.16677	.16630	.16583	48
8.3	.16536	.16490	.16444	.16398	.16352	.16307	.16262	.16218	.16171	.16127	46
8.4	.16082	.16038	.15994	.15950	.15906	.15862	.15819	.15776	.15733	.15690	44
8.5	.15647	.15605	.15562	.15520	.15478	.15436	.15395	.15353	.15312	.15271	42
8.6	.15230	.15190	.15149	.15109	.15068	.15028	.14988	.14949	.14909	.14870	40
8.7	.14831	.14791	.14753	.14714	.14675	.14637	.14599	.14560	.14522	.14485	38
8.8	.14447	.14409	.14372	.14335	.14298	.14261	.14224	.14188	.14151	.14115	37
8.9	.14079	.14043	.14007	.13971	.13935	.13900	.13865	.13830	.13795	.13760	35
9.0	.13725	.13690	.13656	.13621	.13587	.13553	.13519	.13485	.13452	.13418	34
9.1	.13385	.13351	.13318	.13285	.13252	.13220	.13187	.13154	.13122	.13090	33
9.2	.13058	.13026	.12994	.12962	.12930	.12899	.12867	.12836	.12805	.12774	32
9.3	.12743	.12712	.12681	.12650	.12620	.12590	.12559	.12529	.12499	.12469	30
9.4	.12439	.12410	.12380	.12351	.12321	.12292	.12263	.12234	.12205	.12176	29
9.5	.12147	.12119	.12090	.12062	.12033	.12005	.11977	.11949	.11921	.11893	28
9.6	.11866	.11838	.11811	.11783	.11756	.11729	.11702	.11675	.11648	.11621	27
9.7	.11594	.11567	.11541	.11514	.11488	.11462	.11436	.11410	.11384	.11358	26
9.8	.11332	.11306	.11281	.11255	.11230	.11204	.11179	.11154	.11129	.11104	25
9.9	.11079	.11054	.11029	.11005	.10980	.10956	.10931	.10907	.10883	.10857	24
ψn	.00	.01	.02	.03	.04	.05	.06	.07	.08	.09	

ψn	.0	.1	.2	.3	.4	.5	.6	.7	.8	.9	
10.	.10035	.10598	.10370	.10150	.09936	.09729	.09529	.09335	.09148	.08965	(21)
11	.08789	.08618	.08452	.08290	.08133	.07982	.07839	.07690	.07551	.07415	(15)
12	.07283	.07156	.07029	.06908	.06789	.06674	.06561	.06452	.06345	.06241	(12)
13	.06139	.06040	.05944	.05850	.05758	.05668	.05580	.05494	.05411	.05329	(9)
14	.05249	.05171	.05095	.05020	.04947	.04876	.04806	.04738	.04671	.04606	(7)
15	.04542	.04479	.04418	.04358	.04299	.04241	.04185	.04130	.04075	.04022	(6)
16	.03970	.03919	.03869	.03820	.03772	.03726	.03678	.03632	.03588	.03544	47
17	.03501	.03459	.03417	.03377	.03337	.03297	.03259	.03221	.03184	.03147	39
18	.03111	.03076	.03041	.03007	.02973	.02940	.02908	.02876	.02844	.02813	33
19	.02783	.02753	.02724	.02695	.02667	.02639	.02611	.02584	.02557	.02531	28
20	.02505	.02480	.02455	.02430	.02405	.02382	.02358	.02335	.02312	.02289	24
21	.02267	.02245	.02223	.02202	.02181	.02160	.02140	.02120	.02100	.02081	21
22	.02061	.02042	.02024	.02005	.01987	.01969	.01951	.01934	.01917	.01899	18
23	.01882	.01866	.01849	.01833	.01817	.01800	.01785	.01770	.01754	.01739	(16)
24	.01726	.01711	.01697	.01683	.01669	.01655	.01641	.01628	.01615	.01601	(14)
25	.01589	.01576	.01563	.01550	.01538	.01526	.01513	.01502	.01490	.01478	(12)
26	.01467	.01456	.01445	.01434	.01423	.01412	.01402	.01392	.01382	.01372	(11)
27	.01362	.01353	.01343	.01334	.01325	.01316	.01308	.01299	.01291	.01283	(10)
28	.01274	.01266	.01258	.01250	.01243	.01235	.01228	.01221	.01214	.01207	(9)
29	.01200	.01193	.01187	.01181	.01175	.01169	.01163	.01157	.01151	.01145	(8)
ψn	.0	.1	.2	.3	.4	.5	.6	.7	.8	.9	Avg D

30 to 200 — $T_o(\psi n)$ for Positive J

ψn	.0	.1	.2	.3	.4	.5	.6	.7	.8	.9	Avg D
30	.010981	.010907	.010834	.010762	.010691	.010620	.010550	.010481	.010412	.010343	71
31	.010277	.010210	.010144	.010079	.010014	.009950	.009886	.009823	.009761	.009699	64
32	.009638	.009578	.009518	.009458	.009400	.009341	.009283	.009226	.009170	.009114	58
33	.009058	.009003	.008948	.008894	.008840	.008787	.008734	.008682	.008631	.008579	53
34	.008528	.008478	.008428	.008379	.008330	.008281	.008233	.008185	.008138	.008091	49
35	.008044	.007998	.007952	.007907	.007862	.007817	.007773	.007729	.007686	.007643	45
36	.007600	.007558	.007516	.007474	.007433	.007392	.007351	.007311	.007271	.007231	41
37	.007192	.007153	.007114	.007076	.007038	.007000	.006963	.006926	.006889	.006852	38
38	.006816	.006780	.006744	.006709	.006674	.006639	.006604	.006570	.006536	.006502	35
39	.006469	.006435	.006402	.006370	.006337	.006305	.006273	.006241	.006210	.006178	32
40	.006147	.006116	.006086	.006055	.006025	.005995	.005966	.005936	.005907	.005878	30
41	.005849	.005821	.005792	.005764	.005736	.005708	.005681	.005653	.005626	.005599	28
42	.005573	.005546	.005520	.005493	.005467	.005441	.005416	.005390	.005365	.005339	26
43	.005314	.005289	.005266	.005241	.005217	.005193	.005169	.005145	.005122	.005098	24
44	.005075	.005052	.005029	.005006	.004983	.004961	.004939	.004916	.004894	.004873	22
45	.004851	.004829	.004808	.004786	.004765	.004744	.004723	.004703	.004682	.004662	21
46	.004641	.004621	.004601	.004581	.004561	.004542	.004522	.004503	.004483	.004464	20
47	.004445	.004426	.004407	.004389	.004370	.004351	.004333	.004315	.004297	.004279	18
48	.004261	.004243	.004225	.004208	.004191	.004173	.004156	.004139	.004122	.004105	17
49	.004088	.004071	.004055	.004038	.004022	.004006	.003989	.003973	.003957	.003941	16
ψn	.0	.1	.2	.3	.4	.5	.6	.7	.8	.9	

ψn	0	1	2	3	4	5	6	7	8	9	
50	.003926	.003773	.003628	.003492	.003364	.003242	.003127	.003018	.002914	.002816	(72)
60	.002723	.002639	.002560	.002487	.002419	.002354	.002294	.002237	.002183	.002133	(57)
70	.002085	.002040	.001997	.001957	.001918	.001881	.001845	.001811	.001778	.001746	(45)
80	.001715	.001685	.001655	.001627	.001600	.001573	.001548	.001523	.001499	.001476	(35)
90	.001453	.001431	.001410	.001389	.001369	.001350	.001331	.001312	.001294	.001277	(28)
100	.001260	.001243	.001227	.001211	.001196	.001180	.001165	.001151	.001137	.001123	(22)
110	.001110	.001097	.001085	.001073	.001061	.001050	.001039	.001028	.001017	.001007	(17)
120	.000997	.000987	.000977	.000967	.000958	.000948	.000939	.000930	.000921	.000913	(13)
130	.000904	.000896	.000888	.000880	.000872	.000864	.000857	.000850	.000843	.000836	(10)
140	.000829	.000823	.000816	.000810	.000803	.000797	.000791	.000785	.000779	.000773	(8)
150	.000768	.000762	.000757	.000752	.000747	.000742	.000737	.000732	.000727	.000722	(6)
160	.000717	.000713	.000708	.000704	.000699	.000695	.000691	.000687	.000683	.000679	44
170	.000675	.000671	.000667	.000663	.000659	.000655	.000652	.000648	.000645	.000641	37
180	.000638	.000634	.000631	.000628	.000625	.000622	.000619	.000616	.000613	.000610	31
190	.000607	.000604	.000601	.000598	.000595	.000593	.000590	.000587	.000585	.000582	27
200	.000580										
ψn	0	1	2	3	4	5	6	7	8	9	Avg D

FOR HIGHER VALUES OF ψn, THE TABULAR IS $\frac{9.775}{(\psi n)^2}$.

TABLE 13

0 to 5 — $T_o(\psi n)$ for Negative J

$\psi n/\lambda$	.00	.01	.02	.03	.04	.05	.06	.07	.08	.09	Avg D
.0	1.0000	1.0000	1.0000	.9999	.9998	.9997	.9996	.9995	.9993	.9992	1
.1	.9990	.9988	.9985	.9983	.9980	.9977	.9974	.9971	.9967	.9963	3
.2	.9959	.9955	.9951	.9946	.9941	.9936	.9931	.9926	.9920	.9915	5
.3	.9909	.9903	.9896	.9890	.9883	.9876	.9869	.9862	.9854	.9847	7
.4	.9839	.9831	.9823	.9814	.9806	.9797	.9788	.9779	.9770	.9760	9
.5	.9751	.9741	.9731	.9721	.9710	.9700	.9689	.9678	.9667	.9656	10
.6	.9645	.9633	.9622	.9610	.9598	.9586	.9573	.9561	.9548	.9536	12
.7	.9523	.9510	.9496	.9483	.9470	.9456	.9442	.9428	.9414	.9400	14
.8	.9386	.9371	.9356	.9342	.9327	.9312	.9297	.9281	.9266	.9250	15
.9	.9235	.9219	.9203	.9187	.9171	.9155	.9138	.9122	.9105	.9089	16
1.0	.9072	.9055	.9038	.9021	.9004	.8986	.8969	.8952	.8934	.8916	17
1.1	.8899	.8881	.8863	.8845	.8827	.8808	.8790	.8772	.8753	.8735	18
1.2	.8716	.8697	.8679	.8660	.8641	.8622	.8603	.8584	.8565	.8545	19
1.3	.8526	.8507	.8487	.8468	.8448	.8429	.8409	.8389	.8370	.8350	20
1.4	.8330	.8310	.8290	.8270	.8250	.8230	.8210	.8190	.8169	.8149	20
1.5	.8129	.8109	.8088	.8068	.8048	.8027	.8007	.7986	.7966	.7945	20
1.6	.7925	.7904	.7884	.7863	.7842	.7822	.7801	.7780	.7760	.7739	21
1.7	.7718	.7697	.7677	.7656	.7635	.7614	.7594	.7573	.7552	.7531	21
1.8	.7511	.7490	.7469	.7448	.7428	.7407	.7386	.7365	.7345	.7324	21
1.9	.7303	.7282	.7262	.7241	.7220	.7200	.7179	.7158	.7138	.7117	21
2.0	.7096	.7076	.7055	.7035	.7014	.6993	.6973	.6952	.6932	.6912	21
2.1	.6891	.6871	.6850	.6830	.6810	.6789	.6769	.6749	.6729	.6709	20
2.2	.6688	.6668	.6648	.6628	.6608	.6588	.6568	.6548	.6528	.6508	20
2.3	.6489	.6469	.6449	.6429	.6410	.6390	.6370	.6351	.6331	.6312	20
2.4	.6292	.6273	.6254	.6234	.6215	.6196	.6176	.6157	.6138	.6119	19
2.5	.6100	.6081	.6062	.6043	.6024	.6005	.5987	.5968	.5949	.5930	19
2.6	.5912	.5893	.5875	.5856	.5838	.5819	.5801	.5783	.5765	.5746	18
2.7	.5728	.5710	.5692	.5674	.5656	.5638	.5620	.5603	.5585	.5567	18
2.8	.5549	.5532	.5514	.5497	.5479	.5462	.5444	.5427	.5410	.5393	17
2.9	.5375	.5358	.5341	.5324	.5307	.5290	.5273	.5257	.5240	.5223	17
3.0	.5207	.5190	.5173	.5157	.5140	.5124	.5108	.5091	.5075	.5059	16
3.1	.5043	.5027	.5011	.4995	.4979	.4963	.4947	.4931	.4915	.4900	16
3.2	.4884	.4868	.4853	.4837	.4822	.4807	.4791	.4776	.4761	.4745	15
3.3	.4730	.4715	.4700	.4685	.4670	.4655	.4641	.4626	.4611	.4596	15
3.4	.4582	.4567	.4553	.4539	.4524	.4509	.4495	.4481	.4467	.4452	14
3.5	.4438	.4424	.4410	.4396	.4382	.4368	.4355	.4341	.4327	.4313	14
3.6	.4300	.4286	.4273	.4259	.4246	.4232	.4219	.4206	.4192	.4179	13
3.7	.4166	.4153	.4140	.4127	.4114	.4101	.4088	.4075	.4062	.4050	13
3.8	.4037	.4024	.4012	.3999	.3987	.3974	.3962	.3949	.3937	.3925	12
3.9	.3912	.3900	.3888	.3876	.3864	.3852	.3840	.3828	.3816	.3804	12
4.0	.3793	.3781	.3769	.3757	.3746	.3734	.3723	.3711	.3700	.3688	11
4.1	.3677	.3666	.3654	.3643	.3632	.3621	.3610	.3599	.3588	.3577	11
4.2	.3566	.3555	.3544	.3533	.3522	.3512	.3501	.3490	.3480	.3469	11
4.3	.3458	.3448	.3437	.3427	.3417	.3406	.3396	.3386	.3376	.3365	10
4.4	.3355	.3345	.3335	.3325	.3315	.3305	.3295	.3285	.3275	.3266	10
4.5	.3256	.3246	.3236	.3227	.3217	.3207	.3198	.3188	.3179	.3169	10
4.6	.3160	.3151	.3141	.3132	.3123	.3113	.3104	.3095	.3086	.3077	9
4.7	.3068	.3059	.3050	.3041	.3032	.3023	.3014	.3005	.2996	.2988	9
4.8	.2979	.2970	.2962	.2953	.2944	.2936	.2927	.2919	.2910	.2902	9
4.9	.2893	.2885	.2877	.2868	.2860	.2852	.2844	.2835	.2827	.2819	8
$\psi n/\lambda$	.00	.01	.02	.03	.04	.05	.06	.07	.08	.09	Avg D

5 to 10 — $T_o(\psi n)$ for Negative J

$\psi n/\lambda$	.00	.01	.02	.03	.04	.05	.06	.07	.08	.09	Avg D
5.0	.2811	.2803	.2795	.2787	.2779	.2771	.2763	.2755	.2747	.2739	8
5.1	.2732	.2724	.2716	.2708	.2701	.2693	.2685	.2678	.2670	.2663	8
5.2	.2655	.2648	.2640	.2633	.2625	.2618	.2611	.2603	.2596	.2589	7
5.3	.2582	.2574	.2567	.2560	.2553	.2546	.2539	.2532	.2525	.2518	7
5.4	.2511	.2504	.2497	.2490	.2483	.2476	.2469	.2463	.2456	.2449	7
5.5	.2442	.2436	.2429	.2422	.2416	.2409	.2402	.2396	.2389	.2383	7
5.6	.2376	.2370	.2363	.2357	.2351	.2344	.2338	.2332	.2325	.2319	6
5.7	.2313	.2307	.2300	.2294	.2288	.2282	.2276	.2270	.2263	.2258	6
5.8	.2252	.2246	.2240	.2234	.2228	.2222	.2216	.2210	.2204	.2198	6
5.9	.2193	.2187	.2181	.2175	.2169	.2164	.2158	.2152	.2147	.2141	6
6.0	.2136	.2130	.2124	.2119	.2113	.2108	.2102	.2097	.2091	.2086	5
6.1	.2081	.2075	.2070	.2064	.2059	.2054	.2048	.2043	.2038	.2033	5
6.2	.2027	.2023	.2017	.2012	.2007	.2001	.1996	.1991	.1986	.1981	5
6.3	.19762	.19712	.19662	.19612	.19562	.19513	.19463	.19419	.19365	.19316	49
6.4	.19268	.19219	.19171	.19122	.19070	.19027	.18979	.18931	.18889	.18837	48
6.5	.18790	.18793	.18696	.18650	.18603	.18557	.18511	.18965	.18919	.18370	46
6.6	.18328	.18283	.18238	.18193	.18148	.18103	.18059	.18015	.17970	.17926	45
6.7	.17882	.17839	.17795	.17752	.17708	.17665	.17622	.17579	.17536	.17494	43
6.8	.17451	.17409	.17367	.17325	.17283	.17241	.17200	.17158	.17117	.17076	42
6.9	.17035	.16994	.16953	.16912	.16872	.16832	.16791	.16751	.16711	.16672	40
7.0	.16632	.16592	.16553	.16519	.16479	.16435	.16397	.16358	.16319	.16281	39
7.1	.16242	.16204	.16166	.16128	.16090	.16052	.16015	.15977	.15940	.15902	38
7.2	.15865	.15828	.15791	.15755	.15718	.15681	.15645	.15609	.15573	.15536	36
7.3	.15501	.15465	.15429	.15393	.15358	.15323	.15287	.15252	.15217	.15182	35
7.4	.15147	.15113	.15078	.15044	.15009	.14975	.14941	.14907	.14873	.14839	34
7.5	.14806	.14772	.14739	.14705	.14672	.14639	.14606	.14573	.14540	.14507	33
7.6	.14475	.14442	.14410	.14377	.14345	.14313	.14281	.14249	.14217	.14186	32
7.7	.14154	.14122	.14091	.14060	.14028	.13997	.13966	.13935	.13905	.13874	31
7.8	.13843	.13813	.13782	.13752	.13722	.13692	.13661	.13631	.13602	.13572	30
7.9	.13542	.13513	.13483	.13454	.13424	.13395	.13366	.13337	.13308	.13279	29
8.0	.13250	.13222	.13193	.13164	.13136	.13108	.13079	.13051	.13023	.12995	28
8.1	.12967	.12939	.12912	.12884	.12857	.12829	.12802	.12774	.12747	.12720	27
8.2	.12693	.12666	.12639	.12612	.12585	.12559	.12532	.12506	.12479	.12453	27
8.3	.12427	.12400	.12374	.12348	.12322	.12297	.12271	.12295	.12219	.12194	26
8.4	.12168	.12143	.12118	.12092	.12067	.12042	.12017	.11992	.11967	.11942	25
8.5	.11918	.11893	.11868	.11844	.11819	.11795	.11771	.11746	.11722	.11698	24
8.6	.11674	.11650	.11626	.11603	.11579	.11555	.11532	.11508	.11485	.11461	24
8.7	.11438	.11415	.11391	.11368	.11345	.11322	.11299	.11277	.11254	.11231	23
8.8	.11208	.11186	.11163	.11141	.11118	.11096	.11074	.11052	.11030	.11007	22
8.9	.10985	.10964	.10942	.10920	.10898	.10876	.10855	.10833	.10812	.10790	22
9.0	.10769	.10748	.10726	.10705	.10684	.10663	.10642	.10621	.10600	.10579	21
9.1	.10558	.10538	.10517	.10496	.10476	.10455	.10435	.10414	.10394	.10374	20
9.2	.10354	.10334	.10313	.10293	.10273	.10253	.10234	.10214	.10194	.10174	20
9.3	.10155	.10135	.10116	.10096	.10077	.10057	.10038	.10019	.09999	.09980	19
9.4	.09961	.09942	.09923	.09904	.09885	.09866	.09848	.09829	.09810	.09792	19
9.5	.09773	.09754	.09736	.09718	.09699	.09681	.09663	.09644	.09626	.09608	19
9.6	.09590	.09572	.09555	.09536	.09518	.09500	.09483	.09465	.09447	.09429	18
9.7	.09412	.09394	.09377	.09359	.09342	.09324	.09307	.09290	.09273	.09255	17
9.8	.09238	.09221	.09204	.09187	.09170	.09153	.09136	.09120	.09103	.09086	17
9.9	.09069	.09053	.09036	.09020	.09003	.08987	.08970	.08954	.08938	.08921	16
$\psi n/\lambda$	.00	.01	.02	.03	.04	.05	.06	.07	.08	.09	Avg D

TABLE 13 (*continued*)

10 to 100 — $T_e(\psi n)$ for Negative J

$\psi n/\lambda$	0	.1	.2	.3	.4	.5	.6	.7	.8	.9	Ave D
10	.08903	.08795	.08589	.08937	.08289	.08189	.08009	.07867	.07733	.07602	(10)
11	.07475	.07351	.07230	.07111	.06996	.06883	.06773	.06665	.06560	.06457	(11)
12	.06357	.06259	.06163	.06067	.05978	.05888	.05800	.05714	.05631	.05548	(9)
13	.05468	.05389	.05312	.05237	.05163	.05091	.05020	.04950	.04882	.04816	(7)
14	.04751	.04687	.04624	.04562	.04502	.04443	.04385	.04328	.04272	.04217	(6)
15	.04169	.04111	.04059	.04009	.03959	.03910	.03862	.03815	.03768	.03723	49
16	.03678	.03634	.03591	.03549	.03507	.03466	.03426	.03386	.03348	.03309	41
17	.03272	.03235	.03199	.03163	.03128	.03093	.03059	.03026	.02993	.02961	35
18	.02929	.02897	.02867	.02836	.02806	.02777	.02748	.02719	.02691	.02664	29
19	.02636	.02610	.02583	.02557	.02532	.02506	.02481	.02457	.02433	.02409	25
20	.02386	.02362	.02340	.02317	.02295	.02273	.02252	.02230	.02210	.02189	22
21	.02169	.02148	.02129	.02109	.02090	.02071	.02052	.02034	.02015	.01997	19
22	.01979	.01962	.01949	.01929	.01910	.01894	.01878	.01861	.01845	.01830	(17)
23	.01814	.01799	.01783	.01768	.01754	.01739	.01724	.01709	.01696	.01682	(15)
24	.01668	.01655	.01641	.01628	.01615	.01602	.01589	.01577	.01565	.01552	(13)
25	.01540	.01527	.01516	.01505	.01496	.01481	.01469	.01458	.01447	.01436	(12)
26	.01425	.01419	.01409	.01396	.01383	.01372	.01362	.01352	.01342	.01332	(10)
27	.01323	.01313	.01304	.01295	.01285	.01276	.01267	.01258	.01249	.01241	(9)
28	.01231	.01222	.01213	.01205	.01197	.01189	.01181	.01173	.01165	.01157	(8)
29	.01149	.01142	.01133	.01125	.01118	.01110	.01103	.01096	.01088	.01081	75
30	.01075	.01067	.01060	.01053	.01046	.01039	.01033	.01026	.01019	.01013	68
31	.01007	.01000	.00994	.00988	.00981	.00975	.00969	.00963	.00957	.00951	62
32	.00945	.00939	.00933	.00928	.00922	.00917	.00911	.00906	.00900	.00895	56
33	.00889	.00884	.00879	.00874	.00869	.00863	.00858	.00853	.00848	.00843	51
34	.00838	.00833	.00828	.00824	.00819	.00814	.00809	.00805	.00800	.00796	47
35	.00791	.00787	.00782	.00778	.00774	.00769	.00765	.00761	.00757	.00752	43
36	.00748	.00744	.00740	.00736	.00732	.00728	.00724	.00720	.00716	.00712	40
37	.00708	.00705	.00701	.00697	.00694	.00690	.00686	.00683	.00679	.00675	37
38	.00672	.00668	.00665	.00662	.00658	.00655	.00651	.00648	.00645	.00642	34
39	.00638	.00635	.00632	.00628	.00625	.00622	.00619	.00616	.00613	.00610	31
40	.00607	.00604	.00601	.00598	.00595	.00592	.00589	.00586	.00583	.00580	29
41	.00578	.00575	.00572	.00569	.00567	.00564	.00561	.00559	.00556	.00553	27
42	.00551	.00548	.00545	.00543	.00540	.00538	.00535	.00533	.00530	.00528	25
43	.00525	.00523	.00521	.00518	.00516	.00513	.00511	.00509	.00507	.00504	24
44	.00502	.00500	.00497	.00495	.00493	.00491	.00488	.00486	.00484	.00482	22
45	.00480	.00478	.00476	.00474	.00472	.00470	.00468	.00466	.00464	.00462	21
46	.00459	.00457	.00455	.00453	.00452	.00450	.00448	.00446	.00444	.00443	19
47	.00441	.00439	.00437	.00435	.00433	.00432	.00430	.00428	.00426	.00425	18
48	.00423	.00421	.00419	.00418	.00416	.00414	.00413	.00411	.00410	.00408	17
49	.00405	.00403	.00402	.00400	.00399	.00397	.00396	.00394	.00393	.00391	16
$\psi n/\lambda$	.0	.1	.2	.3	.4	.5	.6	.7	.8	.9	Ave D

$\psi n/\lambda$	0	1	2	3	4	5	6	7	8	9	Ave D
50	.003893	.003799	.003702	.003648	.003561	.003221	.003107	.003000	.002897	.002800	(12)
60	.002708	.002620	.002537	.002457	.002381	.002308	.002239	.002173	.002110	.002049	(7)
70	.001991	.001935	.001882	.001831	.001782	.001735	.001690	.001646	.001609	.001569	(5)
80	.001525	.001487	.001451	.001417	.001383	.001351	.001320	.001289	.001260	.001232	(32)
90	.001205	.001179	.001153	.001129	.001105	.001082	.001059	.001037	.001016	.000996	(23)
$\psi n/\lambda$	0	1	2	3	4	5	6	7	8	9	Ave D

100 to 200 — $T_e(\psi n)$ for Negative J

$\psi n/\lambda$	0	1	2	3	4	5	6	7	8	9	Ave D
100	.0009766	.0009576	.0009387	.0009204	.0009026	.0008853	.0008683	.0008531	.0008379	.0008221	(17)
110	.0008072	.0007928	.0007787	.0007650	.0007516	.0007386	.0007257	.0007134	.0007016	.0006898	(13)
120	.0006789	.0006672	.0006563	.0006457	.0006356	.0006252	.0006159	.0006057	.0005963	.0005871	(10)
130	.0005781	.0005693	.0005607	.0005523	.0005441	.0005361	.0005282	.0005206	.0005130	.0005057	(8)
140	.0004985	.0004915	.0004846	.0004778	.0004712	.0004647	.0004584	.0004522	.0004461	.0004401	(6)
150	.0004343	.0004285	.0004229	.0004174	.0004120	.0004067	.0004015	.0003964	.0003914	.0003865	(5)
160	.0003817	.0003770	.0003723	.0003678	.0003633	.0003589	.0003546	.0003504	.0003462	.0003421	41
170	.0003381	.0003342	.0003303	.0003265	.0003228	.0003191	.0003155	.0003119	.0003084	.0003049	37
180	.0003016	.0002983	.0002950	.0002918	.0002887	.0002855	.0002825	.0002795	.0002765	.0002736	31
190	.0002707	.0002679	.0002651	.0002624	.0002597	.0002570	.0002544	.0002518	.0002493	.0002468	27
200	.0002443										
$\psi n/\lambda$	0	1	2	3	4	5	6	7	8	9	Ave D

For higher values of ψn, $T_e(\psi n) = \dfrac{9.775}{(\psi n)^2}$

Solving with M_1 where $x = l_1$, gives

$$\tan(\phi_1 n + \gamma'_1) = \frac{\pi}{180}\phi_1 n \frac{J'_{11}}{J_1}$$

In a similar manner the equations for reducing the masses back from the end of the vibrating system may be proved.

c In the above demonstrations a knowledge of the characteristics of simple harmonic motion has been assumed. In general simple harmonic motion is any motion such that the relation between displacement and time is of the following type:

$$y = a \cos pt$$

where a is the maximum displacement and $p = 2\pi n$. Differentiating y twice with respect to t gives

$$\frac{d^2y}{dt^2} = -ap^2 \cos pt = -p^2 y$$

Appendix No. 2

SOLUTION FOR INTERPOLATION FORMULAS OF TABLE 4

Formula No. 1.

Let h_1 and h_2 be ratios such that

$$n = n_2 + h_1 \Delta_1$$
$$= n_3 - h_2 \Delta_2$$

Then assuming that $h_1 = h_2 = h$, which is approximately true when n_1, n_2, n_3 do not diverge rapidly,

$$n_2 + h\Delta_1 = n_3 - h\Delta_2$$

Therefore
$$h = \frac{\Delta_2}{\Delta_1 + \Delta_2}$$

so that
$$n = n_2 + \frac{\Delta_1 \Delta_2}{\Delta_1 + \Delta_2}$$

Formula No. 2.

Let h_1 and h_2 be ratios such that

$$n = n_2 + h_1 \Delta_1$$
$$= n_4 - h_2 \Delta_2$$

Then assuming that $h_1 = h_2 = h$

$$n_2 + h\Delta_1 = n_4 - h\Delta_2$$

Therefore
$$h = \frac{n_4 - n_2}{\Delta_1 + \Delta_2}$$

so that
$$n = n_2 + \frac{(n_4 - n_2)\Delta_1}{\Delta_1 + \Delta_2}$$

Appendix No. 3

LIST OF THE MORE IMPORTANT SYMBOLS USED IN THE TEXT

- C = stiffness factor (lb-in./radian) of a shaft representing the proportional relationship of any twisting moment M (lb-in.) and the resulting angular deflection θ (radians) such that $M = C\theta$. Refer to Equation [1]
- e = step number for the end step
- g = gravitational constant, which is taken as 385.92 in./sec.^{2}
- G = shearing modulus of elasticity (lb. per sq. in.). For steel this is 11,800,000 lb. per sq. in.
- J = inertia factor or the weight polar moment of inertia of a mass (lb-in.^{2})
- J_s = sectional polar moment of inertia of the area of a shaft (in.^{4})
- J_w = weight polar moment of inertia per unit length of a step with mass (lb-in.^{2}/in.)
- k = number of a typical step
- l = length (in.)
- M = twisting moment (lb-in.)
- n = frequency of vibration in vibrations per second
- p = frequency constant—$2\pi n$
- t = time variable (seconds)
- $T(\phi n)$ = Defined by Equation [22]. Values of this expression corresponding to values of ϕn are listed in the tables. This function applies to a step with mass

TABLE 14

$T(\phi n)$

0° to 45°

ϕn	″0	″1	″2	″3	″4	″5	″6	″7	″8	″9	Ave D
0°	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0001	1.0001	
1	1.0001	1.0001	1.0001	1.0001	1.0002	1.0002	1.0002	1.0003	1.0003	1.0004	
2	1.0004	1.0004	1.0005	1.0005	1.0006	1.0006	1.0007	1.0007	1.0008	1.0009	
3	1.0009	1.0010	1.0010	1.0011	1.0012	1.0012	1.0013	1.0014	1.0015	1.0015	1
4	1.0016	1.0017	1.0018	1.0019	1.0020	1.0021	1.0022	1.0023	1.0023	1.0024	1
5	1.0025	1.0026	1.0028	1.0029	1.0030	1.0031	1.0032	1.0033	1.0034	1.0035	1
6	1.0037	1.0038	1.0039	1.0040	1.0042	1.0043	1.0044	1.0046	1.0047	1.0049	1
7	1.0050	1.0052	1.0053	1.0054	1.0056	1.0058	1.0059	1.0061	1.0062	1.0064	2
8	1.0065	1.0067	1.0069	1.0071	1.0072	1.0074	1.0076	1.0078	1.0079	1.0081	2
9	1.0083	1.0085	1.0087	1.0089	1.0091	1.0093	1.0095	1.0097	1.0099	1.0101	2
10°	1.0103	1.0105	1.0107	1.0109	1.0111	1.0113	1.0116	1.0118	1.0120	1.0122	2
11	1.0125	1.0127	1.0129	1.0132	1.0134	1.0136	1.0139	1.0141	1.0144	1.0146	2
12	1.0149	1.0151	1.0154	1.0157	1.0159	1.0162	1.0165	1.0167	1.0170	1.0173	3
13	1.0175	1.0178	1.0181	1.0184	1.0186	1.0189	1.0192	1.0195	1.0198	1.0201	3
14	1.0204	1.0207	1.0210	1.0213	1.0216	1.0219	1.0222	1.0225	1.0229	1.0232	3
15	1.0235	1.0238	1.0241	1.0245	1.0248	1.0251	1.0255	1.0258	1.0261	1.0265	3
16	1.0268	1.0272	1.0275	1.0279	1.0282	1.0286	1.0290	1.0293	1.0297	1.0300	4
17	1.0304	1.0308	1.0312	1.0315	1.0319	1.0323	1.0327	1.0331	1.0335	1.0339	4
18	1.0343	1.0346	1.0350	1.0355	1.0359	1.0363	1.0367	1.0371	1.0375	1.0379	4
19	1.0383	1.0388	1.0392	1.0396	1.0401	1.0405	1.0409	1.0414	1.0418	1.0423	4
20°	1.0427	1.0431	1.0436	1.0441	1.0445	1.0450	1.0454	1.0459	1.0464	1.0468	5
21	1.0473	1.0478	1.0483	1.0488	1.0493	1.0497	1.0502	1.0507	1.0512	1.0517	5
22	1.0522	1.0527	1.0532	1.0538	1.0543	1.0548	1.0553	1.0558	1.0564	1.0569	5
23	1.0574	1.0580	1.0585	1.0590	1.0596	1.0601	1.0607	1.0612	1.0618	1.0623	5
24	1.0629	1.0635	1.0640	1.0646	1.0652	1.0658	1.0663	1.0669	1.0675	1.0681	6
25	1.0687	1.0693	1.0699	1.0705	1.0711	1.0717	1.0723	1.0729	1.0736	1.0742	6
26	1.0748	1.0754	1.0761	1.0767	1.0773	1.0780	1.0786	1.0793	1.0799	1.0806	6
27	1.0812	1.0819	1.0826	1.0832	1.0839	1.0846	1.0853	1.0860	1.0866	1.0873	7
28	1.0880	1.0887	1.0894	1.0901	1.0908	1.0915	1.0923	1.0930	1.0937	1.0944	7
29	1.0952	1.0959	1.0966	1.0974	1.0981	1.0989	1.0996	1.1004	1.1011	1.1019	7
30°	1.1027	1.1034	1.1042	1.1050	1.1058	1.1066	1.1073	1.1081	1.1089	1.1097	8
31	1.1105	1.1114	1.1122	1.1130	1.1138	1.1146	1.1155	1.1163	1.1172	1.1180	8
32	1.1188	1.1197	1.1205	1.1214	1.1223	1.1231	1.1240	1.1249	1.1258	1.1266	9
33	1.1275	1.1284	1.1293	1.1302	1.1311	1.1320	1.1330	1.1339	1.1348	1.1357	9
34	1.1367	1.1376	1.1385	1.1395	1.1404	1.1414	1.1424	1.1433	1.1443	1.1453	10
35	1.1463	1.1472	1.1482	1.1492	1.1502	1.1512	1.1522	1.1532	1.1543	1.1553	10
36	1.1563	1.1574	1.1584	1.1594	1.1605	1.1616	1.1626	1.1637	1.1647	1.1658	11
37	1.1669	1.1680	1.1691	1.1702	1.1713	1.1724	1.1735	1.1746	1.1757	1.1769	11
38	1.1780	1.1792	1.1803	1.1814	1.1826	1.1838	1.1849	1.1861	1.1873	1.1885	12
39	1.1897	1.1909	1.1921	1.1933	1.1945	1.1957	1.1969	1.1982	1.1994	1.2007	12
40°	1.2019	1.2032	1.2044	1.2057	1.2070	1.2083	1.2096	1.2109	1.2122	1.2135	13
41	1.2148	1.2161	1.2174	1.2188	1.2201	1.2215	1.2228	1.2242	1.2256	1.2269	13
42	1.2283	1.2297	1.2311	1.2325	1.2339	1.2353	1.2368	1.2382	1.2396	1.2411	14
43	1.2425	1.2440	1.2455	1.2469	1.2484	1.2499	1.2514	1.2529	1.2544	1.2560	15
44	1.2575	1.2590	1.2606	1.2621	1.2637	1.2653	1.2668	1.2684	1.2700	1.2716	16
ϕn	″0	″1	″2	″3	″4	″5	″6	″7	″8	″9	Ave D

$-T(\phi n)$

90° to 135°

ϕn	″0	″1	″2	″3	″4	″5	″6	″7	″8	″9	Ave D
90°	∞	364.9	181.97	121.18	90.78	72.58	60.39	51.76	45.19	40.12	
91	36.07	32.76	29.99	27.65	25.68	23.91	22.39	21.05	19.860	18.793	
92	17.834	16.966	16.176	15.455	14.795	14.187	13.626	13.106	12.629	12.175	
93	11.756	11.363	10.996	10.650	10.325	10.019	9.730	9.456	9.196	8.950	
94	8.717	8.499	8.283	8.081	7.888	7.704	7.528	7.359	7.197	7.042	(18)
95	6.894	6.751	6.613	6.481	6.354	6.231	6.112	5.998	5.888	5.781	(12)
96	5.678	5.579	5.483	5.389	5.299	5.211	5.126	5.044	4.964	4.886	(9)
97	4.811	4.737	4.666	4.597	4.529	4.463	4.398	4.337	4.277	4.218	(7)
98	4.160	4.104	4.049	3.995	3.943	3.892	3.842	3.794	3.746	3.700	(5)
99	3.654	3.610	3.566	3.524	3.482	3.441	3.401	3.362	3.324	3.286	41
100°	3.249	3.213	3.178	3.143	3.109	3.076	3.043	3.011	2.980	2.949	33
101	2.918	2.889	2.859	2.831	2.802	2.775	2.747	2.720	2.694	2.668	28
102	2.643	2.618	2.593	2.569	2.545	2.521	2.498	2.476	2.453	2.431	23
103	2.409	2.388	2.367	2.346	2.326	2.306	2.286	2.267	2.247	2.228	20
104	2.210	2.191	2.173	2.155	2.137	2.120	2.103	2.086	2.069	2.053	17
105	2.036	2.020	2.004	1.9870	1.9735	1.9583	1.9439	1.9289	1.9138	1.8993	(15)
106	1.8850	1.8709	1.8570	1.8432	1.8296	1.8162	1.8030	1.7898	1.7769	1.7641	(13)
107	1.7513	1.7390	1.7266	1.7144	1.7023	1.6904	1.6786	1.6670	1.6554	1.6440	(12)
108	1.6328	1.6216	1.6106	1.5997	1.5889	1.5782	1.5677	1.5573	1.5469	1.5367	(11)
109	1.5266	1.5166	1.5067	1.4969	1.4872	1.4776	1.4681	1.4587	1.4494	1.4402	96
110°	1.4311	1.4221	1.4131	1.4043	1.3955	1.3868	1.3782	1.3697	1.3613	1.3530	87
111	1.3447	1.3365	1.3284	1.3204	1.3124	1.3045	1.2967	1.2890	1.2813	1.2737	79
112	1.2662	1.2587	1.2513	1.2440	1.2367	1.2295	1.2224	1.2154	1.2083	1.2014	72
113	1.1945	1.1877	1.1809	1.1742	1.1676	1.1610	1.1544	1.1480	1.1415	1.1352	66
114	1.1288	1.1226	1.1164	1.1102	1.1041	1.0980	1.0920	1.0861	1.0801	1.0743	61
115	1.0684	1.0627	1.0569	1.0513	1.0456	1.0400	1.0345	1.0290	1.0236	1.0181	56
116	1.0127	1.0074	1.0021	.9968	.9916	.9865	.9813	.9762	.9711	.9661	52
117	.9611	.9562	.9512	.9464	.9415	.9367	.9319	.9272	.9225	.9178	48
118	.9132	.9086	.9040	.8995	.8950	.8905	.8861	.8817	.8773	.8729	45
119	.8686	.8643	.8601	.8558	.8516	.8474	.8433	.8392	.8351	.8310	42
120°	.8270	.8230	.8190	.8150	.8111	.8072	.8033	.7995	.7957	.7918	39
121	.7881	.7843	.7806	.7769	.7732	.7695	.7659	.7623	.7587	.7551	37
122	.7516	.7481	.7446	.7411	.7376	.7342	.7308	.7274	.7240	.7206	34
123	.7173	.7140	.7107	.7074	.7042	.7009	.6977	.6946	.6913	.6882	32
124	.6850	.6819	.6788	.6757	.6727	.6696	.6666	.6636	.6606	.6576	30
125	.6546	.6517	.6487	.6458	.6429	.6400	.6372	.6343	.6315	.6287	29
126	.6259	.6231	.6203	.6176	.6148	.6121	.6094	.6067	.6040	.6013	27
127	.5987	.5961	.5934	.5908	.5882	.5856	.5831	.5805	.5780	.5754	26
128	.5729	.5704	.5679	.5655	.5630	.5605	.5581	.5557	.5533	.5509	25
129	.5485	.5461	.5437	.5414	.5390	.5367	.5344	.5321	.5298	.5275	23
130°	.5252	.5230	.5207	.5185	.5163	.5141	.5119	.5097	.5075	.5053	22
131	.5031	.5010	.4988	.4967	.4946	.4925	.4904	.4883	.4862	.4841	21
132	.4821	.4800	.4780	.4759	.4739	.4719	.4699	.4679	.4659	.4639	20
133	.4620	.4600	.4581	.4561	.4542	.4523	.4503	.4484	.4465	.4447	19
134	.4428	.4409	.4390	.4372	.4353	.4335	.4317	.4298	.4280	.4262	18
ϕn	.0°	″1	″2	″3	″4	″5	″6	″7	″8	″9	Ave D

$T(\phi n)$

45° to 90°

ϕn	.0	.1	.2	.3	.4	.5	.6	.7	.8	.9	Ave D
45°	1.2732	1.2749	1.2765	1.2781	1.2798	1.2814	1.2831	1.2846	1.2864	1.2881	17
46	1.2898	1.2915	1.2932	1.2950	1.2967	1.2984	1.3002	1.3019	1.3037	1.3055	17
47	1.3073	1.3091	1.3109	1.3127	1.3145	1.3164	1.3182	1.3201	1.3219	1.3238	18
48	1.3257	1.3276	1.3295	1.3314	1.3333	1.3353	1.3372	1.3392	1.3412	1.3431	19
49	1.3451	1.3471	1.3491	1.3512	1.3532	1.3552	1.3573	1.3594	1.3615	1.3635	20
50°	1.3656	1.3678	1.3699	1.3720	1.3742	1.3763	1.3785	1.3807	1.3829	1.3851	22
51	1.3873	1.3896	1.3918	1.3941	1.3964	1.3987	1.4010	1.4033	1.4056	1.4079	23
52	1.4103	1.4127	1.4150	1.4174	1.4198	1.4223	1.4247	1.4272	1.4296	1.4321	24
53	1.4346	1.4371	1.4396	1.4421	1.4447	1.4473	1.4499	1.4525	1.4551	1.4577	26
54	1.4604	1.4631	1.4657	1.4684	1.4711	1.4739	1.4766	1.4794	1.4822	1.4849	27
55	1.4878	1.4906	1.4934	1.4963	1.4992	1.5021	1.5050	1.5079	1.5109	1.5139	29
56	1.5169	1.5199	1.5229	1.5260	1.5290	1.5321	1.5352	1.5384	1.5415	1.5447	31
57	1.5479	1.5511	1.5543	1.5575	1.5608	1.5641	1.5674	1.5708	1.5741	1.5775	33
58	1.5809	1.5843	1.5878	1.5912	1.5947	1.5983	1.6018	1.6054	1.6090	1.6126	35
59	1.6162	1.6199	1.6236	1.6273	1.6310	1.6348	1.6386	1.6424	1.6462	1.6501	38
60°	1.6540	1.6579	1.6619	1.6658	1.6698	1.6739	1.6779	1.6820	1.6862	1.6903	40
61	1.6945	1.6987	1.7028	1.7072	1.7115	1.7159	1.7202	1.7246	1.7291	1.7335	43
62	1.7380	1.7426	1.7471	1.7517	1.7564	1.7610	1.7657	1.7705	1.7752	1.7801	47
63	1.7849	1.7898	1.7947	1.7997	1.8047	1.8097	1.8148	1.8199	1.8251	1.8303	50
64	1.8355	1.8408	1.8461	1.8515	1.8569	1.8624	1.8679	1.8734	1.8790	1.8846	55
65	1.8903	1.8961	1.9018	1.9077	1.9135	1.9195	1.9254	1.9314	1.9375	1.9436	59
66	1.9498	1.9561	1.9623	1.9687	1.9751	1.9815	1.9880	1.9946	2.0012	2.0079	66
67	2.015	2.021	2.028	2.035	2.042	2.049	2.056	2.064	2.071	2.078	7
68	2.085	2.093	2.100	2.108	2.116	2.123	2.131	2.139	2.147	2.155	8
69	2.163	2.171	2.180	2.188	2.196	2.205	2.214	2.222	2.231	2.240	9
70°	2.249	2.258	2.267	2.276	2.286	2.295	2.305	2.314	2.324	2.334	10
71	2.344	2.354	2.364	2.374	2.384	2.395	2.406	2.416	2.427	2.438	10
72	2.449	2.460	2.472	2.483	2.495	2.506	2.518	2.530	2.542	2.555	12
73	2.567	2.580	2.593	2.605	2.618	2.632	2.645	2.659	2.672	2.686	13
74	2.700	2.714	2.729	2.743	2.758	2.773	2.788	2.804	2.819	2.835	15
75	2.851	2.867	2.884	2.900	2.917	2.934	2.952	2.969	2.987	3.005	17
76	3.024	3.042	3.061	3.080	3.100	3.120	3.140	3.160	3.181	3.202	20
77	3.223	3.245	3.267	3.289	3.312	3.335	3.358	3.382	3.406	3.431	23
78	3.456	3.481	3.507	3.533	3.560	3.587	3.615	3.643	3.672	3.701	27
79	3.731	3.761	3.792	3.824	3.856	3.889	3.922	3.956	3.990	4.026	33
80°	4.062	4.099	4.136	4.174	4.213	4.253	4.294	4.336	4.378	4.422	40
81	4.466	4.512	4.558	4.606	4.654	4.704	4.755	4.807	4.861	4.916	(5)
82	4.972	5.029	5.088	5.149	5.211	5.275	5.341	5.408	5.478	5.549	(6)
83	5.623	5.698	5.775	5.855	5.938	6.023	6.110	6.200	6.293	6.390	(8)
84	6.490	6.593	6.699	6.809	6.929	7.042	7.165	7.292	7.429	7.562	(12)
85	7.705	7.853	8.008	8.170	8.339	8.515	8.699	8.892	9.093	9.305	(16)
86	9.528	9.761	10.007	10.267	10.540	10.830	11.136	11.461	11.807	12.174	
87	12.566	12.985	13.435	13.917	14.437	14.998	15.605	16.266	16.987	17.774	
88	18.643	19.605	20.67	21.86	23.20	24.72	26.46	28.46	30.80	33.57	
89	36.88	40.93	46.00	52.51	61.20	73.36	91.59	121.99	182.78	365.2	
ϕn	.0	.1	.2	.3	.4	.5	.6	.7	.8	.9	Ave D

$-T(\phi n)$

135° to 180°

ϕn	″0	″1	″2	″3	″4	″5	″6	″7	″8	″9	Ave D
135°	.4244	.4226	.4208	.4191	.4173	.4155	.4138	.4120	.4103	.4086	18
136	.4068	.4051	.4034	.4017	.4000	.3983	.3966	.3950	.3933	.3916	17
137	.3900	.3883	.3867	.3851	.3835	.3818	.3802	.3786	.3770	.3754	16
138	.3738	.3723	.3707	.3691	.3676	.3660	.3645	.3629	.3614	.3598	16
139	.3583	.3568	.3553	.3538	.3523	.3508	.3493	.3478	.3463	.3449	15
140°	.3434	.3419	.3404	.3390	.3376	.3362	.3347	.3333	.3319	.3305	14
141	.3291	.3277	.3263	.3249	.3235	.3221	.3207	.3193	.3180	.3166	14
142	.3152	.3139	.3125	.3112	.3099	.3085	.3072	.3059	.3046	.3032	13
143	.3019	.3006	.2993	.2980	.2967	.2954	.2942	.2929	.2916	.2903	13
144	.2891	.2878	.2866	.2853	.2841	.2828	.2816	.2804	.2791	.2779	12
145	.2767	.2755	.2743	.2730	.2718	.2706	.2694	.2683	.2671	.2659	12
146	.2647	.2635	.2624	.2612	.2600	.2589	.2577	.2566	.2554	.2543	12
147	.2531	.2520	.2508	.2497	.2486	.2475	.2463	.2452	.2441	.2430	11
148	.2419	.2408	.2397	.2386	.2375	.2364	.2354	.2343	.2332	.2321	11
149	.2311	.2300	.2289	.2279	.2268	.2258	.2247	.2237	.2226	.2216	10
150°	.2205	.2195	.2185	.2174	.2164	.2154	.2144	.2134	.2123	.2113	10
151	.2103	.2093	.2083	.2073	.2063	.2053	.2044	.2034	.2024	.2014	10
152	.20043	.19945	.19848	.19751	.19655	.19558	.19462	.19366	.19271	.19176	96
153	.19081	.18986	.18892	.18798	.18704	.18610	.18517	.18424	.18331	.18238	94
154	.18146	.18054	.17962	.17871	.17779	.17688	.17598	.17507	.17417	.17327	91
155	.17238	.17148	.17058	.16969	.16880	.16792	.16703	.16615	.16527	.16439	89
156	.16352	.16265	.16178	.16092	.16005	.15919	.15833	.15747	.15661	.15576	86
157	.15491	.15406	.15321	.15237	.15152	.15068	.14985	.14901	.14817	.14734	84
158	.14651	.14568	.14486	.14404	.14321	.14239	.14158	.14076	.13995	.13914	82
159	.13833	.13752	.13671	.13591	.13511	.13431	.13351	.13271	.13192	.13113	80
160°	.13034	.12955	.12876	.12798	.12720	.12641	.12564	.12486	.12408	.12331	78
161	.12254	.12177	.12100	.12023	.11947	.11871	.11794	.11718	.11643	.11567	76
162	.11492	.11416	.11341	.11266	.11192	.11117	.11043	.10968	.10894	.10820	75
163	.10747	.10673	.10600	.10526	.10453	.10380	.10307	.10235	.10162	.10090	73
164	.10018	.09946	.09874	.09802	.09731	.09659	.09588	.09517	.09446	.09376	71
165	.09304	.09234	.09164	.09093	.09023	.08953	.08883	.08814	.08744	.08675	70
166	.08606	.08537	.08468	.08399	.08330	.08262	.08193	.08125	.08057	.07989	68
167	.07921	.07853	.07785	.07718	.07651	.07583	.07516	.07449	.07382	.07316	67
168	.07249	.07183	.07116	.07050	.06984	.06918	.06852	.06786	.06721	.06655	66
169	.06590	.06525	.06460	.06395	.06330	.06265	.06200	.06136	.06071	.06007	65
170°	.05943	.05879	.05815	.05751	.05687	.05623	.05560	.05497	.05433	.05370	64
171	.05307	.05244	.05181	.05118	.05056	.04993	.04930	.04868	.04806	.04744	63
172	.04682	.04620	.04558	.04496	.04434	.04373	.04311	.04250	.04189	.04128	62
173	.04066	.04006	.03945	.03885	.03823	.03763	.03702	.03642	.03581	.03521	61
174	.03461	.03401	.03341	.03281	.03221	.03162	.03102	.03042	.02983	.02924	60
175	.02864	.02805	.02746	.02687	.02628	.02569	.02511	.02452	.02393	.02335	59
176	.022746	.022181	.021598	.021016	.020435	.019855	.019275	.018696	.018118	.017541	580
177	.016965	.016389	.015819	.015240	.014666	.014093	.013521	.012950	.012380	.011810	573
178	.011241	.010672	.010109	.009537	.008971	.008405	.007840	.007276	.006712	.006149	566
179	.005587	.005026	.004465	.003906	.003345	.002786	.002227	.001668	.001109	.0005539	
ϕn	″0	″1	″2	″3	″4	″5	″6	″7	″8	″9	Ave D

TABLE 14 (*continued*)

$T(\phi n)$

180° to 360° POSITIVE FROM $\phi n = 180°$ TO 270°

ϕn	0°	1°	2°	3°	4°	5°	6°	7°	8°	9°	Arc D
180°	0	.005525	.010993	.016408	.02177	.02710	.03238	.03762	.04283	.04801	
190	.05317	.05831	.06343	.06859	.07369	.07873	.08382	.08892	.09402	.09914	
200°	.10427	.10942	.11460	.11981	.12505	.13033	.13566	.14103	.14646	.15196	
210	.15752	.16316	.16888	.17469	.18060	.18660	.19272	.19897	.20537	.21194	
220	.2185	.2254	.2324	.2396	.2470	.2546	.2625	.2707	.2791	.2878	
230	.2969	.3063	.3161	.3263	.3370	.3482	.3599	.3723	.3853	.3990	
240	.4135	.4289	.4453	.4628	.4814	.5015	.5231	.5465	.5718	.5994	
250°	.6297	.6629	.6998	.7407	.7867	.8386	.8977	.9657	1.0448	1.1381	
260	1.2498	1.3860	1.5560	1.7743	2.065	2.471	3.080	4.095	6.122	12.202	
THE FOLLOWING VALUES ARE NEGATIVE											
270	∞	12.113	6.032	4.005	2.990	2.381	1.9751	1.6896	1.4665	1.2966	
280	1.1605	1.0490	.9559	.8769	.8092	.7503	.6987	.6530	.6123	.5758	
290	.5428	.5129	.4857	.4607	.4377	.4165	.3969	.3786	.3616	.3457	
300°	.3308	.3168	.3036	.2912	.2799	.2683	.2577	.2477	.2381	.2290	
310	.2203	.2119	.2040	.19638	.18895	.18189	.17509	.16855	.16223	.15613	
320	.15024	.14458	.13902	.13367	.12848	.12349	.11855	.11379	.10915	.10469	
330	.10024	.09595	.09176	.08767	.08367	.07975	.07592	.07217	.06849	.06488	
340	.06134	.05785	.05443	.05107	.04776	.04450	.04129	.03812	.03500	.03191	
350°	.02887	.02585	.02288	.019929	.017011	.014120	.011259	.008411	.005589	.002786	
ϕn	0	1	2	3	4	5	6	7	8	9	Arc D

$T_0(\psi n)$ = Defined by Equation [21]. Values of this expression corresponding to values of ψn are listed in the tables. This function applies to a step without mass

x = length variable (inches)

α = constant in degrees. This constant is defined by Equation [18]

γ = constant in degrees. This constant is defined by Equations [9] and [12]

θ = amplitude of vibration in radians

$\theta°$ = amplitude of vibration in degrees

ϕ = constant for steps having mass. Defined by Equation [4]

ψ = a constant defined by Equations [5] and [6].

Subscripts are used with the various symbols and have the following meaning. A single subscript means that the value applies to the step having the number of the subscript. If the subscript is made up of two figures, the element applies to a certain section. These sections are designated thus:

01, 12; 23 ... between the steps whose numbers are included

01, 02, 03 ... beginning of the step whose number is given

*l*1, *l*2, *l*3 ... end of the step whose number is given

A prime mark (′) means that the masses are being reduced from the beginning.

A double-prime mark (″) means that the masses are being reduced back from the end.

Discussion

MERVIN B. HOGAN.[11] The problem of torsional vibration is one of great importance in the field of internal-combustion-engine theory. This type of prime mover is now so general and found in so many applications that it is essential that engineers be familiar with the peculiarities of systems of which it is a component part. All the present-day trends in the application and design of these engines have been conducive to torsional-vibration difficulties. The tendencies are toward higher speeds and a greater number of cylinders. The higher speeds have increased the possibility of torsional vibration by raising the running speeds up into the region of critical speeds. On the other hand, it is a well-known fact that, for a given system, an increase in mass or flexibility will lower the natural frequency. As a consequence, the increase in the number of cylinders increased the mass, and the longer crankshaft, which necessarily resulted, increased the flexibility, so that the resultant effect was a lowering of the critical speeds.

[11] Power Engineering Department, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.

In this and in his two preceding papers Mr. Porter has made a praiseworthy contribution to the literature of the subject in the English language. The mathematical theory of the subject has been developed to a nice degree and the practical application of the results has been pointed out. The problem here presented is that of reducing involved systems to simpler systems to which known solutions at hand can be applied.

The necessity of some such method is quite apparent. It is theoretically possible to establish relations between the natural frequencies and the physical properties of any system, expressing the former as a function of the latter. However, such equations are extremely complicated for systems of more than three masses, and the degree becomes of such order that a mathematician is required to undertake the solution. Even with a correct solution the probability of error in a numerical calculation is very high, and the labor required makes such exact solutions impractical in engineering.

The theory of the method here presented is very interesting. It seems that it would be advantageous in design work where many applications of Diesel engines to systems of approximately the same magnitude and arrangement are encountered. In such cases the routine calculation of critical speeds would be expedited. It is well to mention, though, that the time required for the actual calculation of the critical speeds is small compared to the time necessary to calculate the moments of inertia, flexibilities, and the other physical constants of the system. Consequently, the advantage in the saving of time or labor is slight.

A disadvantage of the method follows directly from the reduction method itself. To one who does not already possess an understanding of this phenomenon the reducing of the system to different forms merely introduces another complication which obscures the physical picture. Again, in order to carry out the calculations, the tables and data of the paper must be on hand. It is a disadvantage to be familiar with only one such method as there are times when these calculations must be made and the tables are not available. Sometimes it is desirable to have the relative angular displacements of the various masses, i.e., the so-called elastic curve. With this method another calculation must be made in order to determine it.

The writer is well acquainted with Holzer's method of calculation, having employed it in his work now for several years. As a consequence he naturally thinks of torsional vibration in terms of this method and compares other methods with it.

One of the best reasons for the use of Holzer's method is that it aids the engineer in getting a clear mental picture of the phenomenon. When idealized, each mass of the system is represented by a uniform circular mass, and the respective shaft flexibilities are indicated as elastic shafts without mass. It is then obvious that at a natural frequency the sum of the inertia torques of the system must be zero. As a result, after the engineer has arrived at his "concentrated mass system" he sees physically what each step of the calculation achieves. This is extremely valuable because as he gains experience his judgment enables him to prevent any numerical errors that may tend to creep in.

The actual operation of calculating numerically the natural frequencies by Holzer's method identifies it as essentially an engineering tool. The tabular method lends itself admirably to slide-rule calculation with accuracy of the results within the accuracy of the known constants. After the equivalent system has been calculated it takes but a short time to carry out the tabulated calculations for each frequency.

The disadvantage of the method is that, for each natural frequency, a cut-and-try method must be resorted to in order to determine the value. A value of the natural frequency which is very nearly correct may be obtained by considering the system

as one of three masses composed, say, of generator, flywheel, and engine. The values of the first and second natural frequencies are then obtained from the solution of the well-known biquadratic equation for a three-mass system. The numerical value as thus obtained for the lowest frequency is usually very close to the exact value, while that for the second is generally in error. Ordinarily these are the only frequencies of interest, as higher ones are sufficiently above the running speed to be of no concern.

Having made the calculation by this method, no special calculations are necessary in order to draw the relative displacement or elastic curve. This curve, for both frequencies, is readily plotted from the values of angular displacement which one of the columns of the tabulated calculations contains.

The matter of spoke flexibility is one of great importance, especially with the usual arrangement of generator, flywheel, and engine. In the case of fabricated synchronous generators where the arms are H-beams, the flexibility of the rotor is of much greater importance than that of the flywheel. This type of construction lends itself nicely to analytical treatment, however, so that its flexibility can be calculated much more accurately than can that of ordinary flywheels.

With the arrangement referred to, the node in the first critical speed ordinarily is at the flywheel. In such a case the flywheel flexibility does not influence the value of critical speed theoretically, and but slightly practically. The generator flexibility, however, is of the same order as that of the shafting between it and the flywheel. Since the frequency of this section of the system is the same as that of the whole system, the importance of the rotor flexibility is obvious. In many cases the error introduced by assuming the generator as a rigid mass is in the neighborhood of 30 to 40 per cent; that is, the critical speed as calculated will be far too high. A design approved on such a calculation usually results in trouble as the critical speed will be in the running range.

The engineer who starts to study the torsional-vibration problem finds that the theoretical field has been developed extensively. As is the case with many other problems the chief trouble comes in the practical application. Such questions as the effect of flanges, sharp changes in shaft sections, keyways, press fits, and other types of mechanical construction upon the flexibility of the system are matters which only experience can settle. Such judgment can be developed only by extensive calculations of units whose critical speeds can be observed.

From the standpoint of the Diesel engine itself, the major problem is that of the crankshaft flexibility. It is indeed an extremely rare occasion when the engine builder can supply the measured value of this characteristic. There are several well-known formulas which aim to bridge this difficulty for the engineer by enabling him to calculate the constant from the known crankshaft dimensions. Some of the most generally used are those of Geiger, Hort, Timoshenko, and Carter. These all have certain objectionable features. The first mentioned, that of Geiger, involves a constant which must be obtained experimentally. If it is possible to make such a test the formula ceases to be necessary. In any case a great deal of experimental data must be on hand in order to apply the formula to new crankshaft designs. Hort's expression is too simple to give sufficiently accurate results for most investigations. Dr. Timoshenko's work is based upon a rational analytical foundation, with the result that it is very complicated. The difficulty of such a problem is obvious as there are the effects of bearing constraint, bending, etc., which cannot be accurately considered. Carter's formula is really empirical, although derived with some rational foundation. Its advantage is that it is simple, involving only the dimensions of the crankshaft, and readily evaluated numerically. Tests have proved it to be sufficiently accurate for engineering calculations, especially in the Diesel-engine field. The writer feels that this formula is the best for this use that has yet been published.

H. WYDLER.[12] The author's work supplements the collection of methods for calculating natural frequencies of torsional vibrations by a further quite unique scheme. This should be welcomed, not so much because the calculations become shorter and simpler, but because it seems that we are presented here, again, with an accomplishment borne out of a superior command of the matter.

The beauty of the method is decidedly to be seen in the fact that it replaces the given system of many masses by a model system which is a far closer approximation to the actual mass distribution than any other simplification could produce, and, further, that it settles the calculation for this model system quickly and correctly within a few calculation steps. Fig. 18, for instance, shows that for a six-cylinder engine plant the natural frequencies of a combination-shaft system differ mostly less than $^1/_2$ per cent from that of the corresponding concentrated-mass system.

As to the simplicity of the matter it is felt that it will appeal more to men who are already familiar with the subject. Since the method necessitates the use of different charts and tables during the calculation process, it may require some practice before the calculation results obtained can be relied upon to be correct. However, the writer thinks that, for engineers who are used to other methods, it may pay them to get acquainted with Mr. Porter's work, even if it were only for checking calculation results obtained from other methods.

The writer would like to add another suggestion. The first elaborate scheme for calculating forced vibrations was presented in Holzer's book. Two articles published during this year by F. Neugebauer (*Technische Mechanik und Thermodynamik*, May, 1930, pp. 147 and 184) and M. Vollmar (*T. M. & T.*, Nov., 1930, p. 382) have also done fine pioneering work. It would be one of the most gratifying steps ahead to obtain complete theoretical control of the actual movements of elastic systems if these rather extensive calculations could be shortened. In the writer's book, on page 78 and following, he undertook to combine harmonics into two groups of forces and to follow the calculation through for the system thus simplified. He wonders if Mr. Porter could not amplify his method also in this direction. It is thought that the community of vibration specialists would appreciate this as a very welcome help in their work.

F. J. GEITTMAN.[13] The writer has been using the method outlined in the paper for the past year in the calculation of the natural frequencies of torsional vibration of Diesel installations. Some 200 installations have been calculated, of which more than 30 were tested with the Geiger torsiograph.

Every instance of a discrepancy larger than that of experimental error has been traced directly to the improper consideration of certain factors, such as the stiffness of flexible couplings, whose stiffness factors were unknown or subject to change in operation.

One of the outstanding features of the method developed by the author is its simplicity. The arithmetical operations involved are rarely more than 20 to 30 operations for a complete calculation. The possibility for error is practically eliminated by rechecking the calculation, and since there are relatively few operations, this does not require a great deal of time.

---

[12] Combustion Utilities Corporation, New York, N. Y.

[13] Design Calculation Work, Fairbanks Morse & Co., Beloit, Wis. Jun. A.S.M.E.

A few examples of the agreement between the actual and calculated frequencies that the writer has observed are given below:

Engine	Actual frequency v.p.m.	Calculated frequency, v.p.m.
8-Cylinder 16 × 20	1680	1696
5-Cylinder 14 × 17	1984	1960
6-Cylinder $8^3/_4 \times 10^1/_2$	3660	3690
3-Cylinder 14 × 17	2493	2520
4-Cylinder 14 × 17	1224	1228

When a new installation is being designed, the simplicity of the method makes it easy to determine the best design of the shafting or masses to give a desired location of the critical speeds. This calculation may be done by a direct solution, thus eliminating the cut-and-try methods which consume so much time.

Another advantage is the ease with which curves of reduced mass or extension-shaft stiffness may be calculated. In "long" installations consisting of the basic engine and flywheel with more than one power takeoff, there is frequently more than one frequency having dangerous harmonics near or in the operating range. By reducing the masses from the end to the flywheel and the masses from the beginning through the flywheel for every frequency from, say, 10 vibrations per second to 60 vibrations per second at intervals of 5 vibrations per second, and plotting both values on the same curve sheet with vibrations per second as abscissas and reduced mass as ordinates, every intersection of the curves represents a natural frequency. In this way none of the natural frequencies that will cause a dangerous vibration near or in the operating speed will be missed.

Work of this type is greatly simplified by the tables published with the paper, which are in a way comparable to the trigonometry tables, although necessarily a bit more complicated since more variables are involved. A calculation such as mentioned would be a very long and tedious job with any method not employing a simplification such as that represented by the tables.

BERNHARD H. ANDERSON.[14] The methods for calculation of natural frequencies described in this paper have undoubtedly a decided advantage over any other methods published on this subject, and the paper is indeed a valuable contribution to engineers working on torsional vibration problems.

The importance of this problem varies of course with the type and construction of the engines.

For some types of engines a torsional-vibration calculation has to be made for almost every new installation to assure a successful operation, while other types only need to be given consideration if the installations are much out of the ordinary.

For a four-cycle engine, of the trunk-piston type and of moderate speed, for instance, it is rather a simple matter to design the crankshaft so as to bring the dangerous criticals above the working range for regular marine and generating purposes. The experience of the company that the writer represents is that the additional cost caused by increasing the size of the crankshaft for this purpose is a good investment.

However, with the tendency of increasing speed for Diesel engines there is a growing need for a simple and rapid way of solving the problem of torsional vibration for commercial purpose, and whenever a tabulation of the natural frequency has to be made, the accuracy of the method applied is of prime importance. For this reason most of the short approximate methods in existence are more or less without value.

The simplification of the methods is only justified if it can be obtained without sacrificing accuracy, and after having made an investigation of the methods given in this paper the writer feels certain that this result has been accomplished by the author.

The writer has applied these methods on a number of installations and in order to investigate the reliability, the results of the calculations have been compared with the natural frequencies arrived at by the Holzer or "Concentrated Mass Method," the accuracy of which is recognized, and proved to give very reliable results for general application on Diesel installation. The installations investigated are:

(1) Six-cylinder 18 × 22 in., engine, flywheel, generator (regular generating installation).

(2) Eight-cylinder 16 × 20 in., engine and heavy flywheel (special marine engine connected to water brake on test in shop).

(3) Six-cylinder 14 × 18 in., engine, flywheel, generator (regular generating installation).

(4) Eight-cylinder 11 × 14 in., engine, combined clutch and flywheel, propeller and generator driven through forward extension shaft (special marine installation for trawler).

(5) Eight-cylinder 9 × 12 in., engine, flywheel, propeller (regular marine installation).

The results in natural-frequency v.p.s. are:

Installation	Porter method	Holzer method	Difference in per cent
1	42.634	42.572	0.145
2	38.990	38.970	0.052
3	54.910	54.870	0.073
4	25.295	25.300	0.000
5	35.600	35.600	0.000

It is interesting to note that the differences in frequencies calculated by the two methods are all smaller than the values given for "per cent error" on the charts of Figs. 16 and 18 of the paper, which they essentially should be, because any mass added to the systems shown on these charts will tend to reduce the error made by using the short method instead of the concentrated-mass method.

The slightly larger values of the frequencies calculated by the Porter method are also in accordance with the charts; that is, the errors read from the curves for the installations considered will be found to be positive.

The only conclusion to be drawn from this investigation is that the difference in natural frequency as obtained by the Porter method and the Holzer method is negligible, and so far as accuracy is concerned, either one of these two methods may be applied.

To calculate the natural frequency by the Holzer method, or by accurate methods given by other investigators (analytical or graphical), is always a laborous undertaking with a tedious procedure of "trial and error." A great saving of time by using the Porter method was very soon apparent, and the simplicity in application is distinctively emphasized by comparing the work involved in solving this problem by the different methods, and furthermore, as there are only a few arithmetical operations involved in the tabulation by the Porter method, the chances to make errors are considerably reduced.

PHILIP MALLOZZI.[15] The paper adds a very valuable method of determining the frequency of any complicated shaft arrangement when in the design stage. The writer would like to call attention to some of the relatively complicated installations found in submarines, as shown by the sketch in Fig. 19. It

[14] Diesel Division, Cooper-Bessemer Corporation, Grove City, Pa. Assoc-Mem. A.S.M.E.

[15] Assistant Mechanical Engineer, U. S. Navy Yard, Brooklyn, N. Y. Mem. A.S.M.E.

will be noted that a heavy motor armature is connected to the engine and flywheel by a length of shaft, and the propeller is connected to the motor armature by a very long shaft.

In an installation of this kind, items such as flywheel, motor armature, propeller, and their connecting shafts may be so chosen as to place the critical speeds above or below the operating range or some of the orders of vibration may be grouped at speeds where operation is not necessary.

The object of the computer is to find the items which give a change in frequency by relatively small changes in dimensions or masses. A family of curves showing the effect on shaft frequency of changes in masses and elasticities is of great assistance in picturing the effect of such changes. The author's method gives an easy and rapid means of constructing such a family of curves.

If the uniform shaft system is used, the solution becomes

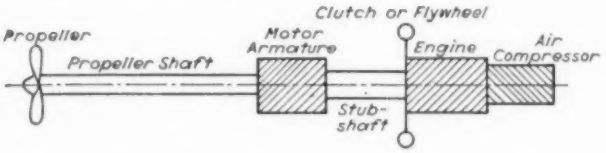


FIG. 19 TYPICAL SUBMARINE SHAFT LINE-UP

laborious because of the trigonometric equations involved. If the concentrated mass system is used, a great amount of numerical computation is required.

By using the author's method the reduction from either side of the shaft line-up is carried out more or less mechanically by the use of tables and charts or by a simple formula. This reduces the labor and chance of error.

In general, where the solution of shaft frequency only is desired, the writer finds the author's method very satisfactory. If, however, a very complete analysis is needed, including calculation of amplitudes and stresses for both synchronous and forced vibration, the uniform shaft system originated by Professor Lewis with the tabular solutions given by Mr. Porter in previous papers is found very satisfactory.

Mr. Porter's paper is a notable contribution to the art of solving torsional vibration problems, and the rapid method he now presents will be welcomed by all engineers confronted with this problem.

GEORGE DASHEFSKY.[16] The author is to be complimented for his past contributions to the technique of calculations pertaining to torsional vibration, as well as for his present valuable contribution. The paper offers several labor-saving and time-saving devices for calculating the natural frequency of complex shafting systems.

In general, a complete analysis of torsional vibration characteristics of an installation may be classified into the following elements:

(1) Calculation of natural frequencies and location of critical speeds

(2) Amplitude of vibration at critical speeds

(3) Vibration stresses occurring in shafting when in a critical speed

(4) Calculation of forced vibration in proximity to the critical speeds; namely, the resonance curves for amplitude and stress.

In each of the elements listed, the frequency solution is usually directly or incidentally involved. The particular system of calculation used will depend upon the idiosyncrasies of the particular calculator. In any case the availability of another method permits an excellent check of a calculation by comparison of results obtained by two systems.

Where a family of curves is required, to show the influence upon the location of critical speeds of various connecting shaft sizes, and the moments of inertia of the driven masses such as flywheels, generators, propellers, etc., much time and labor are involved. Any device for conserving these becomes very important. The present paper in this respect is invaluable and should be of great utility.

Of the various classifications of installations shown in the paper, most cases met in conventional shaft line-ups may be thrown into the forms shown in form 2 of Fig. 4, or the forms shown in Figs. 5 and 6. The adaptation of the system to one of these forms and use of the tables may introduce some error of approximation in the calculated frequency due to interpolation of trial values of n. Usually such slight departure from the actual frequency is in itself unimportant, with respect to stating the location of critical speeds and ranges of speeds to be prohibited to operation. However, as stated at the outset, the determination of the amplitudes of vibration and of the vibration stresses and moments are dependent upon the frequency solution. It is in this respect that the writer's experience has shown that whatever system of calculation is used, whether the concentrated-mass system, the uniform-shaft system, or crosses between them, critical values of n may be encountered, and it becomes essential that the system of equations be quite closely satisfied by the value of frequency considered to applying to the "mathematical" installation.

Referring to Equations [17], [18], and [19] of the paper, it will be observed that the amplitude of vibration and vibratory twisting moments and stresses at any points in a step with mass involves a step constant α. This constant is determined by assuming unit amplitude at the beginning of the line-up and solving for the α value for the first step, and computing in turn the α values for succeeding steps, from the following relation:

$$\alpha_{k+1} = \alpha_k \frac{\cos(\phi_k n + \gamma_k)}{\cos \gamma_{k+1}}$$

This equation follows from the relations given in Equation [18] of the paper. It often happens that values of $\phi_k n + \gamma_k$ and γ_{k+1} are close to 90 deg. or 270 deg. Even small error in the assumed value of n with respect to the value of n which satisfies the system of equations in such case may result in very considerable errors in the calculation of α. Consequently, in such cases similar error results in the calculated vibration moments and stresses. Where only a frequency solution is being sought, it is not necessary to determine n with the finesse implied, and the interpolated values are usually of sufficient accuracy.

The frequency solutions devised by the author quite thoroughly systematize the frequency calculations. The tabulated values for reduction of masses and for the direct solution of some of the simpler line-ups are indeed valuable contributions to the art.

E. A. TULUS.[17] The author has contributed another very interesting paper on the important problem of Diesel-engine vibration. The proposed method for calculating natural frequencies uses a slightly different scheme than the one the writer works with now, but it gives a speedy, accurate solution through the use of standard forms and tables. In an engine with only a few cylinders the accuracy of the proposed method will not be nearly as good as the Holzer method and it is questionable whether it will be much more rapid. For standard work the Holzer method is more routine since an addition of masses simply

[16] Assistant Mechanical Engineer, Navy Yard, New York, N. Y.

[17] Research Engineer, Power Engineering Department, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.

means a longer procedure. However, the theory is no more complicated and the results are very accurate. These factors in some cases may overweigh the advantage of simplicity in favor of Mr. Porter's proposed method.

To the writer's mind, the most important point in a torsional-vibration calculation for a Diesel-engine unit is the evaluation of the elastic constants of the shaft. It has been the writer's experience that a marked change in the inertia constants for the system will usually result in only a slight change in the calculated critical frequency. On the other hand, a slight change in the shaft flexibility, especially if it occurs in the nodal section,

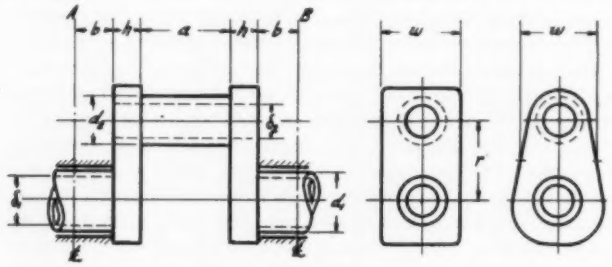


FIG. 20

will produce a large change in the natural frequency and will at times bring the unit within the dangerous range. For this reason the data presented by the author in regard to shrink fits and couplings are very important. In addition, to the valuable instructions the author has presented for calculating shrunk couplings, it would be very interesting to know if he has any data on the twist of shafts in pressed or shrunk fits and to know how this flexibility varies as a function of the length and the pressure of the fit. In short shafts this is very important because the relative value of the shrunk portion is quite large in respect to the rest of the shaft. In selecting the coupling the designer should keep in mind the application in which a Diesel drive will be used. Some of the flexible couplings have rigidity constants which vary with the load. An application of such a coupling to a constant-speed but variable-load installation, for instance, a dredge, results in unnecessary complications. Such an application would mean that the characteristics of the unit would be similar to the variable-speed unit, and hence the designer would have to consider the whole range of criticals instead of only one of them. Where the installation of the flexible coupling is necessary for some other reason, the designer should select one that has a constant rigidity coefficient or one that has very good damping properties.

Another phase of the problem of calculating rigidity constants that is important is the evaluation of the rigidity of the crankshaft. A bibliography on this subject is given at the end of this discussion.

Of the six formulas mentioned, the Geiger equation requires experimental constants and hence cannot be used directly in predetermining the crankshaft flexibility. A tabulation of all the available data and the results that have been published indicate that Carter's formula will give the most accurate results. One does not need experimental data to apply Carter's formula, and in addition the computations are relatively simple. The rigidity constant as expressed by Carter's formula is $K = \frac{G \times J}{L}$ in.-lb. per radian. G is the value of shear modulus and may be taken as 11.8×10^6 lb. per sq. in., J is the polar moment of inertia of the cross-section of the shaft and is expressed as:

$$J = \frac{\pi}{32}(d_1^4 - \delta_1^4)\text{in.}^4$$

L is the equivalent length from A to B (see Fig. 1) in terms of the journal shaft and may be found by:

$$L = (2b + 0.8h) + \frac{3}{4}a\left(\frac{d_1^4 - \delta_1^4}{d_2^4 - \delta_2^4}\right) + \frac{3}{2}r\left(\frac{d_1^4 - \delta_1^4}{h\omega^3}\right)$$

This same compilation of the crankshaft formulas shows that Constant's method usually gives closer results than Carter's. This is not surprising when one considers that Constant considers irregularities in the crank webs, whereas Carter neglects them. Nevertheless the writer prefers to use Carter's formula, because it is much simpler and gives nearly as good results.

After the calculations are completed, the designer must decide how far the operating speed must be kept from any one of the major orders of the critical. Of course, this decision will be influenced to a large extent by the type of installation. For instance, in Diesel-driven dredges there is quite a large amount damping present due to the pump impeller running under water. In such a case the limit of the difference between any one important major and the running speed may be lowered, and still the installation may be considered as safe from torsional vibrations. As another example one may mention the Diesel drive in railway service. With the present design of high-speed multiple-cylinder Diesels, the range of critical speeds is such that the actual operating range will overlap these speeds and the units have to be designed to meet this condition. Usually,

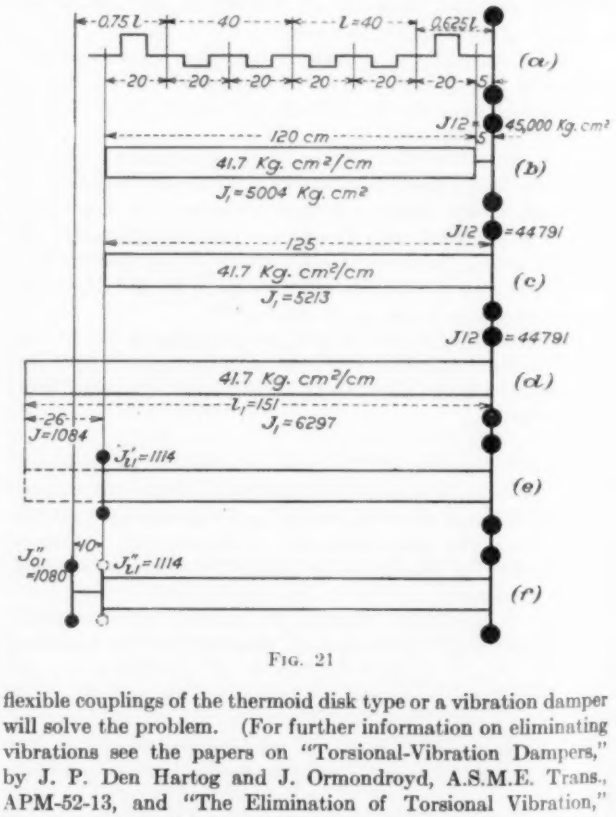


FIG. 21

flexible couplings of the thermoid disk type or a vibration damper will solve the problem. (For further information on eliminating vibrations see the papers on "Torsional-Vibration Dampers," by J. P. Den Hartog and J. Ormondroyd, A.S.M.E. Trans., APM-52-13, and "The Elimination of Torsional Vibration," by George J. Dashefsky, A.S.M.E. Trans., Penn State meeting, June, 1930.) Finally, consider a Diesel generator power installation. For this service the unit will operate at a constant speed and will probably be designed in such a way that the system will not be able to absorb the energy due to vibration by damping. In such a case the limit should be higher than in the two mentioned cases. It would be very interesting to hear of the experience of the author in similar installations. The

writer's experience is such that the difference between a major order and the running speed of a Diesel unit is varied to suit the conditions of operation, but in no case in constant-speed installations is this difference allowed to be less than 10 per cent of the operating speed.

Bibliography

1 Aeronautical Research Committee (Reports and Memoranda No. 1201), E. 29, "On the Stiffness of Crankshafts," by H. Constant, B. A.

2 "The Angular Distortion of Crankshafts," by C. A. Norman and K. W. Stinson. Ohio State University Engineering Experiment Station Bulletin No. 43.

3 "The Bending and Torsion of Multi-Throw Crankshafts on Many Supports," by S. Timoshenko, A.S.M.E. Bulletin.

4 "The Range and Severity of Torsional Vibration in Diesel Engines," by F. P. Porter, A.S.M.E. Trans., Paper APM-50-8.

5 "An Empirical Formula for Crankshaft Stiffness in Torsion," by B. C. Carter, *Engineering*, July 13, 1928.

6 "Zur Berechnung der Verdrehungsschwingungen von Wellenleitungen," V.D.I., 1921.

H. Schreck.[18] The paper is of great value and will be appreciated by all engineers who have to do with the calculation of critical speed. The methods most used today are those developed by Holzer and Dr.-Ing. Wydler, both of which require a long cut-and-try method and the use of formulas of quite considerable length.

In order to try out the simple method suggested by Porter, the writer worked out an actual example presented by Dr.-Ing. Schlaefke in the *Journal* of the Verein Deutscher Ingenieure of October 18, 1930. The article points out the difference between the calculated critical speed and the actual one as determined on this engine. The calculation had not taken into account the effect of the drive off the shaft for the camshaft, and the author in his paper determined how much mass must be acting at this gear to cause a difference in the first frequency of 11,600 per min. calculated against 9700 per min. measured on the engine.

The dimensions of the engine under consideration are given in the metric system, which will therefore also be used in this calculation. The values are given in kg. and cm. Formulas and tables given by Porter do not require any change, except in one place, which will be stated.

The first problem will be to calculate the first frequency without consideration of the gear drive. The original article reduced the six-cylinder engine to an equivalent three-cylinder system, but since a six-cylinder engine is no more complicated to calculate by the Porter method than a three-cylinder engine, the layout is reduced to its actual arrangement on the engine (see Figure 21-*a*).

From data given, it figures that 8.34 kg. acts at each crank and 450 kg. at the flywheel. These values are those reduced to a common radius, which may be assumed to be 10 cm. At each crank is then acting (the symbols used are those of the paper under discussion), J = 834 kg. cm.^{2} or for six cranks J = 5004 kg. cm.^{2}, or distributed over the length of the shaft, J_s = 41.7 kg. cm. At the flywheel is acting J = 45,000 kg. cm.^{2} The sectional polar moment of inertia of the shaft can be figured from the values given and the assumption R = 10 cm. to J_s = 452.3 cm.^{4}

The system reduces itself then to Fig. 21-*b*. A further simplification is applied, in that part of the flywheel is distributed over the 5-cm. length without mass, amounting to 41.7 × 5 = 209 kg. cm.^{2} The critical speed now figures as follows:

$$J_1 = 125 \times 41.7 = 5213$$

$$C_1 = \frac{820{,}000 \times 452.3}{125} = 2{,}967{,}000$$

$$\phi_1 = \frac{360}{\sqrt{g}} \sqrt{\frac{J_1}{C_1}} = 11.5 \sqrt{\frac{J_1}{C_1}} = 0.482$$

$$-\frac{J_{12}}{J_1} = -\frac{44791}{5213} = -8.59$$

We then read from Table 14, $\phi_1 n = 94.06$, which gives $n = \frac{94.06}{0.482} = 195$ v.p.s., or 11,700 vibrations per minute, against Dr. Schlaefke's value of 11,600 v.p.m.

Let us now find how much the possible error can be due to simplifications. It is

$$\frac{J_{23}}{J_1} = \frac{45{,}000}{5004} = 9 \quad \text{and} \quad \frac{l_2}{l_1} = \frac{5}{120} = 0.0417$$

With these values we can read from Fig. 16 that the error due to solving Fig. 21-*c* instead of *b* is negligible and from Fig. 18 due to solving Fig. 21-*c* instead of *a* is +0.3 per cent, or a total of +0.3 per cent error.

We will now proceed to answer the question put by Dr. Schlaefke as to how much the mass of the gear drive opposite the flywheel end must be to cause the actual frequency on the engine to be only 9700 v.p.s. Dr. Schlaefke takes this mass as μ times the mass of the two cranks, which is 1668 kg. cm.^{2}

Different values for μ must be assumed, but we will take at once the value 0.65, which gave the desired result. For Fig. 21-*d*, the inertia factor of the gear drive 0.65 × 1668 = 1084. Length of added step 1084/41.7 = 26 cm.; J_1 = 5213 + 1084 = 6297; $C_1 = \frac{820{,}000 \times 452.3}{151} = 2{,}456{,}000$; $\phi_1 = 11.5 \sqrt{\frac{J_1}{C_1}} = 0.587$; $-\frac{J_{12}}{J_1} = -7.113$.

We then read from Table 14, $\phi_1 n = 94.85$, which gives $n = \frac{94.85}{0.587} = 161.6$ v.p.s., or 9700 vibrations per minute.

This then means that to satisfy the measurements of the critical speed on the engine proper, the mass of the gear drive amounts to 1.3 times that of one cylinder.

In this last part of the calculation the shaft was simply extended as mass per unit length, while on the engine proper the center of the gear or the load is located 10 cm. from the center of the last bearing, as can be determined from Dr. Schlaefke's layout and $\lambda_x = 0.75\ l$. In order to find how much the error in the mass is due to this assumption, the mass will be reduced backward to a concentrated mass at the end of the original system (Fig. 21-*e*), and then this mass in turn reduced to one located on the centerline of the gear (Fig. 21-*b*). Referring to Fig. 21-*e*, it is

$$J_1 = 1084$$

$$C_1 = \frac{820{,}000 \times 452.3}{26} = 14{,}260{,}000$$

$$\phi_1 = 11.5 \sqrt{\frac{J_1}{C_1}} = 0.1002$$

$$\phi_1 n = 0.1002 \times 161.6 = 16.2$$

Then referring to Table 14, we find the table value $T(\phi_1 n)$ and it is $J'_{l1} = J_1 T(\phi_1 n) = 1084 \times 1.0275 = 1114$. Since $J'_{l1} + J''_{l1} = 0$ (Equation [14]), it is $J''_{l1} = -1114$ kg. cm.^{2}. A simple rule for knowing whether this mass has a positive or

[18] Development Engineer, Fairbanks, Morse & Co., Beloit, Wis. Mem. A.S.M.E.

negative value is that it is positive when the point to which the mass is reduced approaches a node and negative when moved away from a node.

Reducing then the mass to its proper place and referring to Fig. 21-*f*, it is $C_1 = \frac{820{,}000 \times 452.3}{10} = 37{,}100{,}000.$

$$\psi''_{11}n = \sqrt{\frac{1}{2.54}} \sqrt{\frac{J''_{11}}{C_1}}\, 161.6 = 161.6 \times 0.6275 \sqrt{\frac{-1114}{37{,}100{,}000}}$$
$$= 0.556i$$

The factor $\sqrt{\frac{1}{2.54}}$ is introduced for conversion from cm. to inch, Porter's tables being figured for the inch system.

When now reading the table value for $T_0(\psi''_{11}n)$, it must be noted that the value 0.556 is imaginary and therefore Table 13 is to be used. $J''_{01} = 1114\ T_0(0.556\ i) = 1114 \times 0.9693 = 1079.8$, or the error in the mass, by putting it at its proper distance, amounts to $\left(1 - \frac{1079.8}{1084}\right) 100 = 0.4$ per cent, or less than $^1/_2$ of 1 per cent.

The writer believes that he has proved by this discussion and the calculation of an actual case that the method suggested by Mr. Porter shortens the work from one-fifth to one-tenth of that required by present analytical methods. It should be noted that the larger part of this calculation is for the checking of percentage of error due to assumptions made by Porter. This part would not be necessary when using this method in everyday work.

To appreciate Mr. Porter's work fully, it would be worth while to compare the laborious formulas used by Dr. Schlaefke (this is not a criticism of his article, because he uses a method generally accepted today) with the few lines of calculation required for the Porter method. Mr. Porter, who has given us a number of papers on critical speed, deserves great credit for his latest work in this field.

J. Ormondroyd.[19] While the author undoubtedly found great pleasure and use from the development of his simplified methods, the writer is not so sure that this method is of such great importance to the rest of the industry. At the present time methods for calculating critical speeds are in the process of being slowly disseminated throughout the ranks of those engineers who should be able to use them. At this stage it is a far better policy to stay as close to fundamental ideas as possible without any elaboration. The graphical and numerical methods of calculating critical speeds go back, in the final analysis, to the Newtonion equations of motion for the simplified system in question. This system is assumed to be vibrating freely and is also assumed to have no damping forces acting upon it or in it.

Under these assumptions the quantities which appear in the simultaneous equations are the moments of inertia of the masses in the system, the spring constants of the elastic members in the system, the angular coordinates (in radians) of the masses and time. These fundamental equations written with these quantities involved can be recognized by any one in any country who has sufficient dynamical education. A thorough grasp of Newton's second and third laws and a small knowledge of the solution of linear differential equations are all that is needed to the final solutions. The Holzer method can be so used that nothing appears in the tabulation work but the very quantities which appear in the simultaneous equations, and the units of angular measure can be maintained as the radian. This seems to be such a simple procedure that one is surprised at the extensive use of such ideas as equivalent lengths and masses at equivalent radii. Now on top of these elaborations which seem to obscure the real problem, Mr. Porter is proposing a really complicated simplification. He is using a mass of new notations which must be memorized by any engineer who starts out with this method and which are particularly hard to understand for any engineer who has been working in the field with notations grown familiar with years of usage.

If one is starting to calculate the critical speeds of an entirely new engine, Mr. Porter's simplification refers only to a small fraction of the total work required. By far the largest job to be accomplished is the reduction of the actual engine to the equivalent simplified system. It may take anywhere from one day to one week to reduce the reciprocating parts to equivalent flywheels and to calculate the equivalent spring constants between these flywheels by using the data contained in the sketches and drawings of the proposed engine. After the equivalent system is set up, the work of making a preliminary estimate of the lowest natural frequency is very simple. This calculation might take about five minutes. After this the application of the Holzer method to arrive at the lowest natural frequency will take about four complete trials; each trial in the case of a system having eight or nine bodies might take five to seven minutes. In other words, after the system has been set up, it only takes one-half hour to calculate the lowest natural frequency and its corresponding normal elastic curve. Since the second and third natural frequencies require more accurate work in the tabular method, it might be said that two hours more should be added for their calculation when they are needed. Mr. Porter's method simplifies this work, which at the utmost can consume only about two and one-half hours, whereas the work which may take from one day to one week is still the same as before.

Mr. Porter's method of course does have merit in production methods of design or where minor changes to existing engines are very frequent. Most engineers who will have to calculate critical speeds will not run up against mass production of design even in an engine-manufacturing concern. The importance of a common language for all engineers working in a certain field is so great and the widespread use of Holzer's method is such an apparent fact, that the writer believes all efforts should be bent toward disseminating that method rather than attempting to bring out graphical methods or other simplified methods. Naturally the method used by each investigator is a matter of his own taste, and the writer certainly is not the one to oppose publishing any methods of calculation which may be devised. He merely wishes to bring out the fact that he does not consider Mr. Porter's simplified method as simple as the original Holzer method.

M. Stone.[20] The present discusser had a chance to discuss the paper with the author at a joint N.E.M.A.-D.E.M.A. meeting for torsional vibration, and hence is no longer under the handicap of having to force his way through the developments.

The author's notations are rather unnecessarily involved, and the large tables of constants are such that one has a certain hesitancy about reading the paper through. But once this is done, there are certain definite points that are well made by the author.

There is no doubt as to the method of calculating the critical frequencies of a system of n degrees of freedom. This has been

[19] Experimental Division Manager, Westinghouse Elec. & Mfg. Co., South Philadelphia, Pa.

[20] Mechanical Engineer, Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa. At present on special investigations in Europe.

done in many ways, but the particular case of torsional vibration was first solved and presented satisfactorily by H. Holzer. His assumptions were that an engine can be replaced by a series of discrete inertias (having an average value of the variable reciprocating inertias) connected by massless flexibilities represented by the crankshaft sections between cylinder centers. Calculations based on this method have been checked by actual experiments for the last fifteen years by various investigators in many parts of the world.

In the interest of shortening these calculations, but not losing any appreciable accuracy, the author has replaced the earlier dynamical picture of the engine by one of a distributed mass and flexibility. A real appreciation of what this means can only be obtained by noting the excellent comparison that he has made in Fig. 18, which forms the first convincing argument as to the value of the approximation. The author has further worked out tabulated values to take care of the effects of large concentrated masses in the torsional system, such as flywheels, etc., and has thus reduced the calculation work still further. In an actual case, however, there still remains a considerable amount of judgment that the calculator must exercise in order to represent properly his system dynamically, so that the error shall not be larger than a few per cent. This judgment is of the kind described in the paragraphs following Table 8 where the "total inertia factor for the scavenge pump including the shaft up to the center line of the first bearing is . . . , and corresponds to an increase in length of the main cranks."

Where a calculator has only one type of engine to work with—and has many calculations to do weekly—then the use of the author's tabulations should be a real help, once the proper "feel" is obtained in judging equivalent systems accurately. For calculations of all types of engines and drives, however, the slightly longer but more accurately determined Holzer method is preferable.

One of the most important parts of the calculation of torsional critical frequencies is the proper evaluation of system constants—inertias and flexibilities. While, in general, the discusser agrees with the author's evaluations, the two pertaining to (1) flexible spokes and (2) crankshaft stiffness are not accurately enough determined. In fact, for certain types of crankshafts, the author's rule of taking an equivalent shaft as "3 per cent stiffer than a uniform shaft having the same section as the journal and the same length" may be 40 per cent off. At the Westinghouse Company, the discusser has been making a comparison of all the existing published tests on crankshafts, together with a few unpublished ones, with all the existing methods of calculating crankshaft stiffness. Although the method presented by Constant[21] seems to be the most accurate, due to the correction curves that are necessary to use along with the method, the method of Carter,[22] which is practically as accurate, is preferable. Depending on the type of crankshaft, whether marine, auto, or airplane, the flexibilities, in relation to uniform shaft of section equal to that at the journal and length equal to the actual length, range from 70 per cent to 140 per cent.

The question of the flexibility of the rotor is one that has been only recently appreciated[23] in its full importance. In a case in mind the discusser remembers an instance where the introduction of the generator-rotor flexibility reduced the calculated second critical frequency from 3600 per min. to 2200 per min.; the observed frequency was 2110 per min. The expression for rotor flexibility given in a previous paper[24] has been more recently improved upon, in that it takes into consideration the bending flexibility of the rim against torque. The discusser is sorry that he does not have this formula or its derivation at hand, but it can be found in a report to the previously mentioned N.E.M.A.-D.E.M.A. committee. If the rim is very stiff, i.e., the term $I_r \longrightarrow \infty$, the formula simplifies to that of the author's. However, for the ratio of I_r to I_a (moment of inertia of rim to moment of inertia of arm) that occurs in actual machines, the value of C calculated from the author's formula may be 100 per cent off. A detailed discussion of this point, together with other similar discussion, will appear in a forthcoming paper.

[21] H. Constant, "Stiffness of Multi-Throw Crankshafts," *Engineering*, Nov. 1, 1929.

[22] B. C. Carter, "Experimental Formula for Crankshaft Stiffness in Torsion," *Engineering*, July 13, 1928.

[23] Author's discussion of Dashevsky's paper at Penn State meeting of A.S.M.E. Oil and Gas Power Division, Summer, 1930.

[24] M. Stone, "Stress Analysis in Electrical Rotating Machinery," Trans. A.S.M.E., 1928.

Author's Closure

The author is grateful to the contributors in the discussion for their favorable and stimulating remarks. Apparently the utility of the proposed method is being considered from two different points of view. One of these is the viewpoint of the engine designer who lays out the installation to accommodate satisfactorily the requirements of the engine and driven machinery. The other is the viewpoint of those who design the driven machinery and who wish to check the installation to insure the safety of their own equipment. In the latter case, some of the discussers feel that the time element in making a calculation is not an important consideration, the principal requirement being that the results are reliable. In the former case, however, it is often necessary to check the frequencies of many different combinations in order to find the best arrangement of the machinery. If this kind of work is carried out thoroughly, the time element in making the calculations becomes an important consideration. It is gratifying to see by the discussion that the engine designers are favorably inclined toward the proposed method.

There are several ways in which the designer is forced to approach the torsional-vibration problem. The following is a description of how it may be given consideration from the beginning of the design work. The description is divided into two parts: (1) preliminary calculations, and (2) calculations for special arrangements.

1 *Preliminary Calculations.* Suppose that a line of engines are to be built having a certain cylinder size and consisting of anywhere from three to eight cylinders. Suppose also that these engines are to be used in any arrangement where Diesel power is desired.

In the first steps of the design work, the weights of the moving parts will be calculated for the purpose of determining the stresses due to the reciprocating and rotating parts. These weights will also be used in designing the counterweights for the three and four cylinder sizes and for the scavenge pump and air compressor. Later the flywheel sizes are determined for the various types of service in which the engines will be used.

Thus in the preliminary design calculations, a greater part of the work in calculating the engine factors for the torsional vibration problem is already made in determining the unbalanced forces and stresses in the engine.

Suppose that many of the adaptations of these engines will be in electric installations driving generators or alternators, and also in marine installations direct connected to marine propellers. The most probable electric equipment will be assumed, as will also several typical marine installations for each of the six engines in the new line.

The critical speeds for these various arrangements will then be calculated. In this way the designer sees whether he can expect to overcome the vibration difficulties in the various engines and

which of the engines will cause him the most trouble. He can then decide if the crankshaft diameter should be changed, or if other changes are necessary to give a proper control of the critical speeds.

Finally the line of engines is put into production. The actual weights of the connecting rods and reciprocating parts are obtained, and the preliminary inertia factors of the engines are modified to agree with these values.

2 *Calculations for Special Arrangements.* When the desired adaptation of the Diesel engine is determined for each customer, the frequency calculations are made for the most obvious arrangement. In many cases, however, this is just the beginning of the work, as it often happens that the most obvious arrangement does not give favorable locations for the critical speeds.

Various arrangements are then calculated. For example, the attached shafting may be changed; the flywheel may be altered;

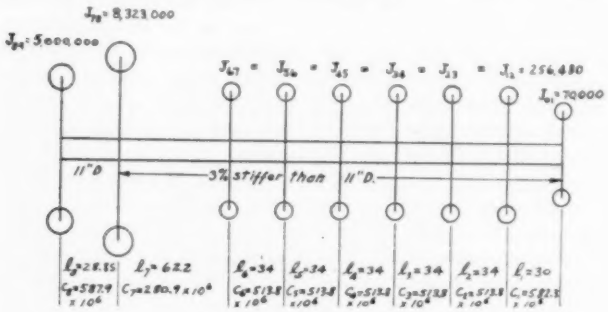


FIG. 22

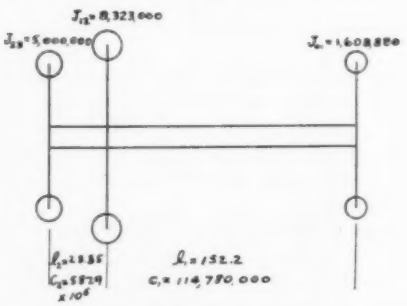


FIG. 23

a small flywheel may be located to advantage at some point, particularly at the engine end opposite the flywheel; or other changes may be considered that are apparent in special installations. Even the main-engine counterweights may be altered or, in especially difficult installations, counterweights may be added on those engines that are inherently balanced.

After a suitable arrangement has been found, the maximum amplitudes and resonance curves may be calculated as a final check on the installation.

Sometimes it is advantageous to draw curves as illustrated in Fig. 13 to give the natural frequencies of any likely combination of the driven machinery.

In the type of design work described above, the frequency calculations require a large percentage of the time given to the torsional-vibration problem. The proposed method of calculation was developed to handle work of this kind in order to meet production methods of manufacturing with speedier methods of calculation and without loss of accuracy.

Most of the discussers have acknowledged that the proposed method has merit when used as just described. Messrs. Hogan and Ormondroyd, however, feel that the amount of time required for the frequency calculations is small in comparison with the time required for the whole calculation. While this idea may be true for those making a final check on a complete installation, it is seldom true for those laying out new designs.

Referring to Mr. Hogan's comments, he states that the proposed method merely introduces another complication which obscures the physical picture, and that one of the best reasons for the use of Holzer's method is that it aids the engineer in getting a clear mental picture of the phenomenon which prevents any numerical errors that tend to creep in. He explains the approximate solution used in the Holzer method and states that the value obtained by the approximate solution is usually very close to the exact value for the first frequency, while that for the second is generally in error.

The last statement mentions an important characteristic, but, contrary to Mr. Hogan's idea, it is true about as often as it is false. A little work with the proposed method, as illustrated below, will show this and will also show the reason. Since Mr. Hogan has used Holzer's method for several years, this would indicate that Holzer's method does not give as good a mental picture of the phenomenon as the proposed method.

Fig. 22 shows the numerical example given in the paper (refer to Figs. 11 and 12) arranged for solution by Holzer's method. The flywheel inertia factor is 8,323,000 lb-in.^{2}, since the value of 8×10^6 as shown in Fig. 12 was obtained by making a deduction from the actual value to spread over a short length in step No. 1 to make the uniform shaft representing the engine extend to the flywheel. This was mentioned in the second paragraph under Table 8. The value of the deduction was $45.2 \times 7544 - 43.85\ (11)^4/36 = 323{,}000$ lb-in.^{2}

Fig. 23 shows this system arranged for the approximate solution by the well-known formula for a three-mass system. The formula will be used as it is given, page 54, Equation [121] of the author's paper APM-50-8, Trans. A.S.M.E.

$$\frac{C_1(J_{01}+J_{12})}{2J_{01}J_{12}}+\frac{C_2(J_{12}+J_{23})}{2J_{12}J_{23}}=\frac{114.78\times 9.932}{2\times 1.6089\times 8.323}+\frac{587.9\times 13.323}{2\times 8.323\times 5}=136.67$$

$$\frac{(J_{01}+J_{12}+J_{23})C_1C_2}{J_{01}J_{12}J_{23}}=\frac{14.932\times 114.78\times 587.9}{16.6089\times 8.323\times 5}=15{,}049$$

$$n = 3.1266\sqrt{136.67 \pm \sqrt{(136.67)^2-15{,}049}}$$

$$= 3.1266\sqrt{136.67 \pm 60.25} = 3.1266\times(7.642 \text{ or } 14.033)$$

$$= 23.89 \text{ or } 43.88$$

We therefore assume $p^2 = (2\pi n)^2 = 23{,}000$ for the first tabular solution by the Holzer method. The calculations are given in Table 15a. The value I in this table is equal to J/g. The assumed value of p^2 is seen to be low. Next we take $p^2 = 25{,}000$ as shown in Table 15b. The value of p^2 is still too low. The next trial value by extrapolation would be $p^2 = 36{,}440$, but anticipating that this value is high, we assume $p^2 = 30{,}000$. Table 15c gives the calculations. Again, by extrapolation, p^2 would be 35,250. Assuming $p^2 = 35{,}000$, another table (15d) is calculated. Interpolating between $p^2 = 30{,}000$ and 35,000, we get $p^2 = 34{,}780$, which gives $n = 29.68$ for the first frequency. It is to be noted, however, that even though our interpolated value of p^2 is correct, we shall have to make another tabular calculation to obtain correct θ-values for the relative-amplitudes curve.

The two tabular calculations necessary for the second frequency solution are given in Tables 15c and 15f. The interpolated value of p^2 is 76,230, which gives $n = 43.94$.

The calculations necessary to obtain these frequencies by the proposed method are given in Table 7.

Summarizing the results, we have:

VALUES OF n IN VIBRATIONS PER SECOND

	Holzer's Method		
	Approx. solution	Accurate method	Proposed method
First frequency.....	23.89	29.68	29.67
Second frequency.....	43.88	43.94	44.07

It will be observed that the first frequency by the approximate solution in Holzer's method is 19.5 per cent in error, while the second frequency is 0.9 per cent in error. The reason that this is contrary to Mr. Hogan's statement is that the engine and flywheel have a larger influence in determining the first frequency than the other parts, and that the flywheel, alternator shaft, and alternator have a larger influence in determining the second frequency than the engine. Thus the rough approximation of replacing the engine with a single concentrated mass at the center of gravity of the inertia factors on the engine side of the flywheel gives only a rough approximation to the first frequency, whereas the second frequency is obtained more accurately by the same approximation.

This fact is easily seen when using the proposed method, thus: First, consider the frequency of the engine and flywheel with the alternator cut off. In this case $n = T^{-1}(-8/1.95) \div 3.113 = 31.5$, which is only 6 per cent from the correct value of the first frequency. Then consider the frequency of the flywheel and alternator with the engine cut off. In this case $n = T_0^{-1}(-8/5) \div 0.09221 = 43.2$, which is only 2 per cent from the correct value of the second frequency.

Another way of seeing this is described at the bottom in the third paragraph below Table 8, where a reference is made to Fig. 15.

The idea is also illustrated in Fig. 13. It will be noted in this figure that portions of the curves are only slightly inclined to the natural-frequency axis and that other portions rise very swiftly from this axis. In the former case where the curves are only slightly inclined, the frequency changes rapidly for changes in the alternator-shaft stiffness, while in the latter case changes in the extension shaft have very little influence on the frequency. The former case is the one that Mr. Hogan has noted, whereas the latter case, which is equally likely to occur, corresponds to the numerical example given above.

The saving of work when using the proposed method is also illustrated by the numerical example. The complete calculations necessary to get the first and second frequencies by Holzer's method are given in Tables 15a to 15f, not counting the calculations required for the approximate solution. Each of these tables requires 41 arithmetical operations and corresponds to one line across Table 7, or 5 arithmetical operations for the proposed method. Therefore not only is the amount of work required for the proposed method less than one-eighth that for Holzer's method in this case, but the possibility of numerical errors is also reduced by the same ratio.

Mr. Hogan mentions that having the calculations by Holzer's method, no special calculations are necessary in order to draw the relative displacement or elastic curve, also that in the case of the proposed method another calculation must be made in order to determine it. The actual facts of the case, however, may be seen by a reference to the numerical example just described. In working this problem by Holzer's method we needed four tables (15a to 15d) for the first frequency and two tables (15c and 15f) for the second frequency to obtain accurate values of p. We do not have tables for these accurate values, because they were obtained by interpolation. Therefore in order to obtain the relative-amplitude curves for the first and second frequencies, we shall have to calculate two more tables having 41 arithmetical operations in each table. The proposed method requires two multiplications and one division to obtain the relative-amplitude curve for each frequency, giving the amplitudes over the engine as ordinates of a sine curve.

TABLE 15: SOLUTION FOR FIG. 22 BY HOLZER'S METHOD

$p^2 = 23{,}000$ Table 15a

Stn No	1 $C \times 10^{-6}$	2 I	3 $Ip^2 10^{-6}$	4 θ	5 $Ip^2\theta 10^{-6}$	6 $\Sigma Ip^2\theta 10^{-6}$	7 $\Sigma \frac{Ip^2\theta}{C}$
		18138	4.172	1.00000	4.172	4.172	.00716
1	582.3	66459	15.286	.99284	15.177	19.349	.03766
2	513.8	66459	15.286	.95518	14.601	33.950	.06608
3	513.8	66459	15.286	.88910	13.591	47.541	.08940
4	513.8	66459	15.286	.79970	12.224	59.765	.11632
5	513.8	66459	15.286	.68338	10.446	70.211	.13665
6	513.8	66459	15.286	.54673	8.357	78.568	.27970
7	2809	21,566.65	496033	.26703	132.456	211.024	.35894
8	5879	12,956.05	297.989	-.09191	-27.388	183.636	

$p^2 = 25{,}000$ Table 15b

Stn No	1 $C \times 10^{-6}$	2 I	3 $Ip^2 10^{-6}$	4 θ	5 $Ip^2\theta 10^{-6}$	6 $\Sigma Ip^2\theta 10^{-6}$	7 $\Sigma \frac{Ip^2\theta}{C}$
		18138	4534	1.00000	4534	4534	.00779
1	582.3	66459	16.615	.99221	16.486	21.020	.04091
2	513.8	66459	16.615	.95130	15.806	36.826	.07167
3	513.8	66459	16.615	.87963	14.615	51.441	.10012
4	513.8	66459	16.615	.77951	12.952	64.393	.12533
5	513.8	66459	16.615	.65418	10.869	75.262	.14648
6	513.8	66459	16.615	.50770	8.435	83.697	.29796
7	2809	21,566.65	539.166	.20974	113.085	196.782	.33472
8	5879	12,956.05	323.901	-.12498	-40.481	156.301	

$p^2 = 30{,}000$ Table 15c

Stn No	1 $C \times 10^{-6}$	2 I	3 $Ip^2 10^{-6}$	4 θ	5 $Ip^2\theta 10^{-6}$	6 $\Sigma Ip^2\theta 10^{-6}$	7 $\Sigma \frac{Ip^2\theta}{C}$
		18138	5.441	1.00000	5.441	5.441	.00934
1	582.3	66459	19.938	.99066	19.752	25.193	.04903
2	513.8	66459	19.938	.94163	18.774	43.967	.08557
3	513.8	66459	19.938	.85606	17.068	61.035	.11879
4	513.8	66459	19.938	.73727	14.700	75.735	.14740
5	513.8	66459	19.938	.58987	11.761	87.496	.17029
6	513.8	66459	19.938	.41958	8.366	95.862	.34127
7	2809	21,566.65	647.000	.07831	50.667	146.529	.24924
8	5879	12,956.05	388.682	-.17093	-66.437	80.092	

$p^2 = 35{,}000$ Table 15d

Stn No	1 $C \times 10^{-6}$	2 I	3 $Ip^2 10^{-6}$	4 θ	5 $Ip^2\theta 10^{-6}$	6 $\Sigma Ip^2\theta 10^{-6}$	7 $\Sigma \frac{Ip^2\theta}{C}$
		18138	6.348	1.00000	6.348	6.348	.01090
1	582.3	66459	23.261	.98910	23.007	29.355	.05713
2	513.8	66459	23.261	.93197	21.679	51.034	.09933
3	513.8	66459	23.261	.83264	19.368	70.402	.13702
4	513.8	66459	23.261	.69562	16.181	86.583	.16851
5	513.8	66459	23.261	.52711	12.261	98.844	.19238
6	513.8	66459	23.261	.33473	7.786	106.630	.37960
7	2809	21,566.65	754.833	-.04487	-33.869	72.761	.12376
8	5879	12,956.05	453.462	-.16364	-76.472	-3.711	

$p^2 = 76{,}000$ Table 15e

Stn No	1 $C \times 10^{-6}$	2 I	3 $Ip^2 10^{-6}$	4 θ	5 $Ip^2\theta 10^{-6}$	6 $\Sigma Ip^2\theta 10^{-6}$	7 $\Sigma \frac{Ip^2\theta}{C}$
		18138	13.785	1.00000	13.785	13.785	.02367
1	582.3	66459	50.509	.97633	49.313	63.098	.12281
2	513.8	66459	50.509	.85352	43.110	106.208	.20671
3	513.8	66459	50.509	.64681	32.670	138.950	.27044
4	513.8	66459	50.509	.37637	19.010	157.960	.30743
5	513.8	66459	50.509	.06894	3.482	161.442	.31421
6	513.8	66459	50.509	-.24527	-12.388	149.054	.53063
7	2809	21,566.65	1639.065	-.77590	-1271.751	-1122.207	-1.96884
8	5879	12,956.05	984.660	1.13294	1115.561	-6.646	

$p^2 = 76{,}500$ Table 15f

Stn No	1 $C \times 10^{-6}$	2 I	3 $Ip^2 10^{-6}$	4 θ	5 $Ip^2\theta 10^{-6}$	6 $\Sigma Ip^2\theta 10^{-6}$	7 $\Sigma \frac{Ip^2\theta}{C}$
		18138	13.876	1.00000	13.876	13.876	.02379
1	582.3	66459	50.841	.97621	49.631	63.507	.12360
2	513.8	66459	50.841	.85261	43.348	106.855	.20797
3	513.8	66459	50.841	.64464	32.774	139.629	.27176
4	513.8	66459	50.841	.37238	18.958	158.587	.30865
5	513.8	66459	50.841	.06423	3.266	161.853	.31501
6	513.8	66459	50.841	-.25078	-12.750	149.103	.53080
7	2809	21,566.65	1649.849	-.78358	-1291.489	-1140.386	-1.93976
8	5879	12,956.05	991.138	1.15818	1147.916	7.530	

Mr. Hogan also brings up the point that in order to carry out the calculations by the proposed method, the tables and data of the paper must be on hand. One could also say this was a disadvantage for the ordinary trigonometric tables or even for the slide rule.

The author agrees with Mr. Hogan that spoke flexibility is often of considerable importance, and has tried to point this out in the paper. Further discussion of this point is given later.

In regard to crankshaft stiffness, the author prefers using factors for similar crankshafts that he has tested rather than using a formula. For any new type of crankshaft, experience has shown that the importance of the torsional-vibration problem warrants a test to determine its stiffness.

The author appreciates Dr. Wydler's comments. He hopes that he may have opportunity to apply his method as suggested.

The tables of results given by Messrs. Geittman and Anderson are interesting. The latter gives a check between Holzer's method and the proposed method that he obtained on five different installations. The former gives a comparison between calculated results and the actual frequencies that he has also observed on five installations.

Mr. Dashefsky mentions that some error of approximation in

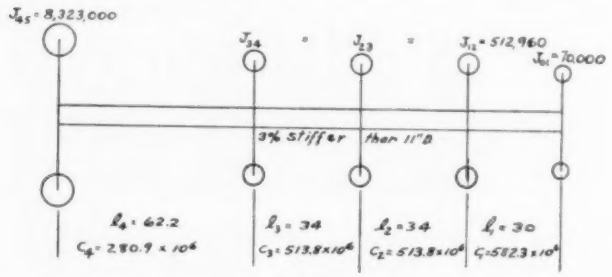


FIG. 24

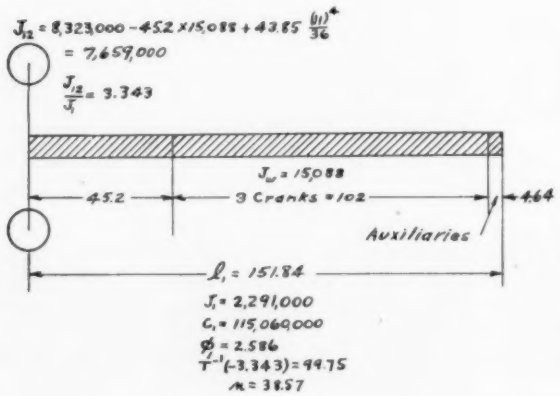


FIG. 25

the calculated frequency may be introduced due to interpolation on trial values of n. This error, however, should be no greater than five parts in 20,000, which in general is the maximum error for which the tables were designed. If there is any doubt that the interpolated value is not correct, another calculation should be made.

The critical values of n mentioned by Mr. Dashefsky are seldom troublesome. They can always be handled in the proposed tabular solutions by assuming one of the factors of the installation as a variable and computing values of n corresponding to different values of the assumed variable.

Mr. Tulus states that in an engine with only a few cylinders the accuracy of the proposed method will not be nearly as good as the Holzer method. This idea was discussed in the paper and the statement was made in the second paragraph following Fig. 9 that "since engines with a small number of cylinders usually have counterweights at each crank web, it seems probable that the actual installation is more like the uniform-shaft system than the system of concentrated masses." We have examined

TABLE 16
SOLUTION FOR FIG. 24 BY HOLZER'S METHOD

p^2 = 58,000 — Table 16a

1	2	3	4	5	6	7
Step No.	$C \times 10^{-6}$	I	$Ip^2 10^{-6}$	θ	$Ip^2\theta 10^{-6}$	$\Sigma Ip^2\theta 10^{-6}$	$\Sigma \frac{Ip^2\theta}{C}$
		181.38	10.520	1.00000	10.520	10.520	.01807
1	582.3	1329.19	77093	.98193	75.700	86.220	.16781
2	513.8	1329.19	77093	.81412	62.763	148.983	.28963
3	513.8	1329.19	77093	.52449	40.435	189.418	.67433
4	280.9	21,566.65	1250.866	-.14984	187.430	1.988	

p^2 = 59,000 — Table 16b

1	2	3	4	5	6	7
Step No.	$C \times 10^{-6}$	I	$Ip^2 10^{-6}$	θ	$Ip^2\theta 10^{-6}$	$\Sigma Ip^2\theta 10^{-6}$	$\Sigma \frac{Ip^2\theta}{C}$
		181.38	10.701	1.00000	10.701	10.701	.01838
1	582.3	1329.19	78.422	.98162	76.981	87.682	.17065
2	513.8	1329.19	78.422	.81097	63.598	151.280	.29443
3	513.8	1329.19	78.422	.51654	40.508	191.788	.68276
4	280.9	21,566.65	1272.432	-.16622	211.504	-19.716	

TABLE 17
SOLUTION FOR FIG. 26

p^2 = 58,000 — Table 17a

1	2	3	4	5	6	7
Step No.	$C \times 10^{-6}$	I	$Ip^2 10^{-6}$	θ	$Ip^2\theta 10^{-6}$	$\Sigma Ip^2\theta 10^{-6}$	$\Sigma \frac{Ip^2\theta}{C}$
		181.38	10.520	1.00000	10.520	10.520	.01325
1	794.1	471.12	27.325	.98675	26.963	37.483	.01716
2	2183.8	386.95	22.443	.96959	21.761	59.244	.02713
3	2183.8	471.12	27.325	.94246	25.753	84.997	.08757
4	9706	471.12	27.325	.85489	23.360	108.357	.04962
5	2183.8	386.95	22.443	.80527	18.073	126.430	.05789
6	2183.8	471.12	27.325	.74738	20.422	146.852	.15130
7	9706	471.12	27.325	.59608	16.288	163.140	.07470
8	2183.8	386.95	22.443	.52138	11.701	174.841	.08006
9	2183.8	471.12	27.325	.44132	12.059	186.900	.57989
10	322.3	21,566.65	1250.866	-.13857	-173.333	13.567	

p^2 = 59,000 — Table 17b

1	2	3	4	5	6	7
Step No.	$C \times 10^{-6}$	I	$Ip^2 10^{-6}$	θ	$Ip^2\theta 10^{-6}$	$\Sigma Ip^2\theta 10^{-6}$	$\Sigma \frac{Ip^2\theta}{C}$
		181.38	10.701	1.00000	10.701	10.701	.01348
1	794.1	471.12	27.796	.98652	27.421	38.122	.01746
2	2183.8	386.95	22.830	.96906	22.124	60.246	.02759
3	2183.8	471.12	27.796	.94147	26.169	86.415	.08903
4	970.6	471.12	27.796	.85244	23.694	110.109	.05042
5	2183.8	386.95	22.830	.80202	18.310	128.419	.05881
6	2183.8	471.12	27.796	.74321	20.658	149.077	.15359
7	970.6	471.12	27.796	.58962	16.389	165.466	.07577
8	2183.8	386.95	22.830	.51385	11.731	177.197	.08114
9	2183.8	471.12	27.796	.43271	12.028	189.225	.58711
10	322.3	21,566.65	1272.432	-.15440	-196.464	-7.239	

this point very carefully and believe that Mr. Tulus is not correct in his statement of the case. For example, consider a three-cylinder engine of the same design as that shown in Fig. 11, driving a flywheel-type generator having an inertia factor of 8,323,000 lb-in.^{2} The usual engine counterweights which are placed on each web practically double the inertia factor at each

cylinder. We shall therefore assume that each counterweight has an inertia factor of 128,240 lb-in.[2]

Fig. 24 shows the equivalent arrangement for solution by Holzer's method. The solution is given in Tables 16a and 16b.

The solution by the proposed method is given in Fig. 25.

Now, consider the inertia factors at each crank as follows:

(1) Inertia factor at the center line of the crankpin.

(*a*)	Piston and connecting rod =	120,050
(*b*)	8 in. of crankpin = 0.8 × 36,600 =	29,280
	Sum =	149,330

(2) Inertia factor at the center of each web.

(*a*)	One web 94,950/2 =	47,475
(*b*)	1/2 of journal = 4880/2 =	2,440
(*c*)	One inch of crankpin =	3,660
(*d*)	One counterweight =	128,240
	Sum =	181,815

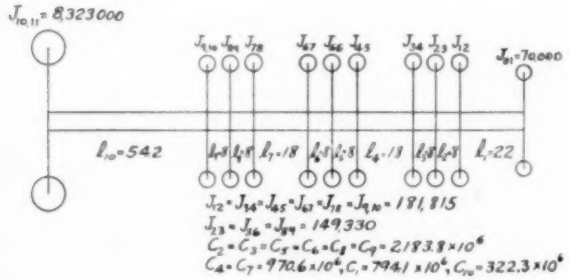


FIG. 26

The length between the center line of the piston and the center of the web equals 8 in., and between the centers of adjacent webs of different cranks is 18 in.

The equivalent arrangement is shown in Fig. 26 and the solution by Holzer's tabular solution is given in Tables 17a and 17b.

The results of the solutions for the three types of equivalent systems considered for this three-cylinder engine are given in Table 18.

Which of these methods gives the more nearly correct result is difficult to say, but it is quite apparent that either (2) or (3) of Table 18 is closer to the actual shaft arrangement than (1). This shows that, in the last analysis, actual fact is contrary to Mr. Tulus' statement.

TABLE 18

	Method	n	Per cent difference relative to (3)
(1)	Holzer's method	38.36	0.47
(2)	Proposed method	38.57	0.08
(3)	Concentrated-mass solution considering three masses per crank	38.54	..

The elastic effects of press fits, clamped fits, and shrink fits vary considerably even in identical installations. We have observed these effects in twisting tests and vibration tests, and recommend the working rules given in the paper. Some work has also been done in treating this problem theoretically.[25]

Mr. Tulus mentions the problem of how far the operating speeds must be kept from any one of the major orders of vibration. This depends upon such limiting factors as excessive stresses, undue wear of parts, noise, flickering lights, or other kinds of objectionable effects due to torsional vibration. The calculation of the range of critical speeds was considered in a previous paper.

The author appreciates Mr. Schreck's addition to the discussion. The example worked out by Mr. Schreck shows very clearly the simplification of the work involved in calculating the natural frequencies by the proposed method.

Mr. Stone states that there still remains a considerable amount of judgment to be used by the calculator in order to properly represent his sytem dynamically, so that the error shall not be larger than a few per cent, and he quotes as proof to his statement the treatment of the scavenge-pump inertia as explained in the numerical example given in the paper.

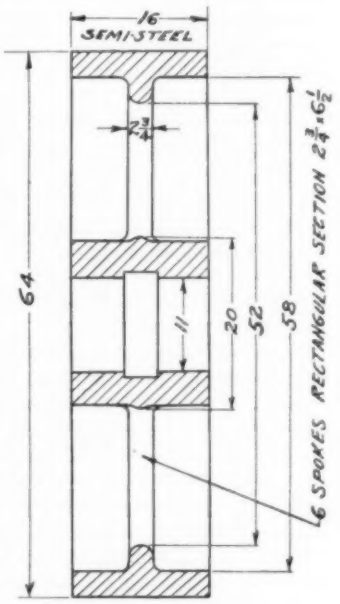


FIG. 27

Apparently Mr. Stone has overlooked Fig. 17 and the explanation of this figure given in the first paragraph below Fig. 9. When one is reducing the actual system to a simpler one for mathematical treatment, it is recommended that he read over the section of the paper entitled "Determination of Equivalent System."

The author does not agree with Mr. Stone's expression, the "more accurately determined Holzer method." The fact is that the theory of the vibration of uniform shafts with mass is just as fundamental and just as accurately determined as the theory underlying Holzer's method. In regard to the accuracy of the results, we have shown both in the paper and in the discussion that there is no practical difference between Holzer's method and the proposed method.

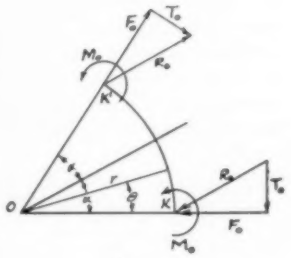


FIG. 28

Mr. Stone brings up two points which he says are not accurately enough determined: (1) flexible spokes and (2) crankshaft stiffness. In regard to crankshaft stiffness, Mr. Stone has apparently misunderstood the statements in the paper. The author would like to point out that he has quoted only the last half of a sentence, the whole of which reads, "A crankshaft having the general proportions shown in Fig. 9 when turning in its bearings under operating conditions is about 3 per cent stiffer than a uniform shaft having the same length and the same section as the journal." The author recognizes that crankshafts of widely different designs will have different values for their stiffness factors. In fact, in the sentence following the one quoted above, he refers to test results where the comparative stiffness ratio varies from less than one to 1.25.

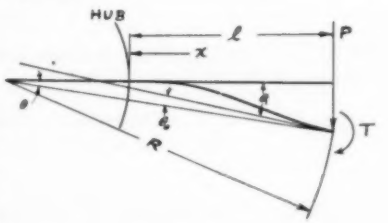


FIG. 29

Several of the discussers, including Mr. Stone, refer to Carter's

[25] Verhandlungen des 2-internationalen Kongresses für technische Mechanik, Zurich, 1926: "Ueber die Torsion von Schrumpfverbindungen," von G. Eichelberg, Winterthur.

formula for crankshaft stiffness. The statement is made that tests have proved it to be sufficiently accurate for engineering calculations, especially in the Diesel-engine field. Undoubtedly Carter's formula is a remarkably good one considering the number of variables involved, and furthermore it does have its uses in evaluating the stiffness factors for crankshafts of new designs. It should be recognized, however, that it is an approximate formula and that the stiffness values of actual shafts do vary considerably from those computed. In this respect Carter makes the following statement: "Taking the test results to be exact, the range of error in the calculated stiffness values is from 6.4 per cent low to 8.2 per cent high for the marine shafts, 10.2 per cent low and 4.8 per cent high for aircraft shafts, and 3.8 per cent low and 13 per cent high for car shafts; thus all the results lie between 10.2 per cent low and 13 per cent high."

Applying Carter's formula to the crank shown in Fig. 9, the comparative stiffness factor works out to be 0.872. This value from test results observed by the author should have been 1.03, which in this case shows an error of 15.3 per cent in the value from Carter's formula.

The large number of variables affecting the crankshaft-stiffness problem include not only the crankshaft dimensions but also the bearing restraint. The latter variable is dependent upon the stiffness of the engine frame, etc. About the only way to get accurate stiffness factors for crankshafts of radically new designs is to obtain them from the engine itself after the new design is built. At the present time, factors obtained from formulas must be considered as only approximate.

The other point mentioned by Mr. Stone is the stiffness factor for a heavy elastic ring attached to the shaft by flexible spokes. Formula [26] applies to a heavy stiff ring attached to the shaft by flexible spokes. In certain special designs it is possible that the elasticity of the ring will be an appreciable amount that should be considered. In this case the proper formula is easily obtained from the usual theory for the bending of nearly circular rings in one plane. The resulting expression for the stiffness factor, C, of the rim relative to the hub is given below and the derivation is given at the end of the closure:

$$C = \frac{12\,NEIr}{l(1+K)}\left(r + \frac{1}{3r} - 1 + \frac{rK}{4}\right) \quad \ldots\ldots\ldots [31]$$

where

$$K = \frac{rI}{I_1}\left(\frac{\beta - \sin\beta}{1 - \cos\beta}\right)$$

The symbols are the same as those for Equation [26] with the addition of the following values:

β = $2\pi/N$, or the angle between spokes (radians)
I_1 = rectangular moment of inertia of the ring section (in.^{4})

Since the radial thickness of the ring is assumed to be small, the value of R in the expression $r = R/l$ is taken to be the same as before, namely, the outer radius of the spokes. Values of $(\beta - \sin\beta)/(1 - \cos\beta)$ for various numbers of spokes are given in Table 19.

TABLE 19

Number of spokes	$\frac{\beta - \sin\beta}{1 - \cos\beta}$
4	0.57080
6	0.36234
8	0.26730
10	0.21224
12	0.17614

Fig. 27 shows the spider of the alternator rotor that was used in the numerical example of Fig. 11.

(*a*) *Considering the rim stiff,*

$$N = 6,\ E = 15 \times 10^6 \text{ lb. per sq. in.},\ I = \frac{1}{12}\,2.75(6.5)^3$$
$$= 62.93 \text{ in.}^4,\ R = 26 \text{ in.},\ l = 16 \text{ in.},\ r = 1.625$$
$$r + \frac{1}{3r} - 1 = 0.8301$$

Therefore from Formula [26]

$$C' = 12 \times 6 \times 15 \times 10^6 \times 62.93 \times 1.625 \times 0.8301 \div 16$$
$$= 5730 \times 10^6$$

which is equivalent to 2.960 in. of 11-in.-diameter shafting. It will be noted that this was given as 3 in. in Fig. 11.

(*b*) *Considering elasticity of rim,*

N, E, I, R, l, r, and $r + \frac{1}{3r} - 1$ as before,

also

$$I_1 = 16 \times 3\left[\frac{(3)^3}{12} + (0.44)^2\right] + 2.75 \times 3\left[\frac{(3)^3}{12} + (2.56)^2\right]$$
$$= 105.55 \text{ in.}^4$$
$$K = 1.5294 \times 78.60 \times 0.3623 \div 105.55 = 0.4126$$
$$\frac{rK}{4} = 1.5294 \times 0.4126 \div 4 = 0.15776$$
$$C' = \frac{12 \times 6 \times 15 \times 10^6 \times 62.93 \times 1.625}{16 \times 1.4126}\,(0.8301 + 0.15776)$$
$$= 4827 \times 10^6$$

which is equivalent to 3.514 in. of 11-in.-diameter shafting.

The error involved in the length of step No. 2 of Fig. 11 due to assuming the rotor rim stiff is 0.554 in. or 1.9 per cent, which is much less than the variation that can occur in the effective length of the shaft under the hub of the rotor. In this case it is evident that the elasticity of the rim is of no consequence. This condition is typical of a great many actual machines. Where additional elasticity in the rim is anticipated, however, the author agrees with Mr. Stone that this condition should also be investigated.

In this discussion, several references have been made to Holzer's method of calculating natural frequencies. The author wishes to state that he had done so for the sake of comparison and with no intention of criticizing this method.

Heavy Elastic Ring Connected to Shaft by Flexible Spokes. The theory of the bending of nearly circular rings in one plane has been very completely treated by Prescott ("Applied Elasticity," Longmans, Green and Co., 1924, pp. 282–318). The rings may be closed or open; the general equations will also apply to rods whose central lines form arcs of circles or whole circles.

In summarizing the general equations given by Prescott, we shall use the symbols as he defined them.

The circle which nearly coincides with the central line of the ring is taken as a curve of reference. The radius of this circle is r. The polar coordinates of a point on the reference circle are (r, θ) with the pole at the center of the circle. The symbols M, F, T denote the bending moment, the shearing force, and the tension at any point. The external forces per unit length of the ring are p acting radially toward the center and q acting tangentially in the direction in which $d\theta$ is measured. The values of p and q for the problem at hand may be taken equal to zero. The change in the value of r for any point in the strained state is u.

The resulting equations which apply to the present problem are obtained by putting p and q equal to zero in the general equations given by Prescott. This gives

$$T = \frac{dF}{d\theta} \quad [32]$$

$$F = -\frac{dT}{d\theta} \quad [33]$$

$$F = \frac{1}{r}\frac{dM}{d\theta} \quad [34]$$

$$M = -\frac{EI}{r^2}\left(\frac{d^2u}{d\theta^2} + u\right) \quad [35]$$

The equations combine to give

$$\frac{d^3u}{d\theta^3} + 2\frac{d^2u}{d\theta^2} + \frac{du}{d\theta} = 0$$

whose general solution may be written

$$u = A + (B + C\theta)\sin\theta + (H + K\theta)\cos\theta \quad [36]$$

The problem of finding the deflection of the ring under the action of the couples acting at the ends of the spokes merely requires the evaluation of the five constants of integration satisfying the end conditions of each segment of the ring. In Fig. 28, KK' represents the segment of the ring between spokes, OK is the initial line for θ, and $\theta = 2\alpha$ is the angle between spokes, with one spoke at OK and the next at OK'.

The two couples M_0 are held in equilibrium by the couple $2R_0r\sin\alpha$. Thus $R_0 = M_0/(r\sin\alpha)$. Resolving R_0 along and perpendicular to the radius $F_0 = M_0/(r\tan\alpha)$ and $T_0 = M_0/r$.

Therefore the conditions to be satisfied at the ends of the segment are

(1) $M = M_0$	(1a) $M = -M_0$
(2) $F = \frac{M_0}{r\tan\alpha}$	(2a) $F = \frac{M_0}{r\tan\alpha}$
(3) $T = \frac{M_0}{r}$	(3a) $T = -\frac{M_0}{r}$
(4) $u = 0$	(4a) $u = 0$
at $\theta = 0$	at $\theta = 2\alpha$

(5) $\frac{du}{d\theta}$ (at $\theta = 0$) $= \frac{du}{d\theta}$ (at $\theta = 2\alpha$)

Differentiating Equation [36] gives

$$\frac{du}{d\theta} = (C - H - K\theta)\sin\theta + (B + K + C\theta)\cos\theta \quad [37]$$

$$\frac{d^2u}{d\theta^2} = -(B + 2K + C\theta)\sin\theta + (2C - H - K\theta)\cos\theta \quad [38]$$

Combining [35], [36] and [38] gives

$$M = -\frac{EI}{r^2}(A - 2K\sin\theta + 2C\cos\theta) \quad [39]$$

Differentiating [39] gives

$$\frac{dM}{d\theta} = \frac{2EI}{r^2}(K\cos\theta + C\sin\theta) \quad [40]$$

Combining [34] and [40] gives

$$F = \frac{2EI}{r^3}(K\cos\theta + C\sin\theta) \quad [41]$$

Differentiating again

$$\frac{dF}{d\theta} = \frac{2EI}{r^3}(-K\sin\theta + C\cos\theta) \quad [42]$$

Combing [32] and [42] gives

$$T = \frac{2EI}{r^3}(-K\sin\theta + C\cos\theta) \quad [43]$$

Putting condition (2) in Equation [41] gives

$$\frac{M_0}{r\tan\alpha} = \frac{2EIK}{r^3}, \text{ or } K = \frac{r^2M_0}{2EI\tan\alpha}$$

Putting condition (3) in Equation [43] gives

$$\frac{M_0}{r} = \frac{2EIC}{r^3}, \quad \text{or } C = \frac{r^2M_0}{2EI}$$

Putting condition (1) in Equation [39] gives

$$M_0 = -\frac{EI}{r^2}\left(A + \frac{r^2M_0}{EI}\right), \text{ or } A = 0$$

Putting condition (4) in Equation [36] gives $H = \theta$

Putting condition (4a) in Equation [36] gives

$$0 = \left(B + \frac{r^2M_0\alpha}{EI}\right)\sin 2\alpha + \frac{r^2M_0\alpha\cos 2\alpha}{EI\tan\alpha}$$

or

$$B = -\frac{r^2M_0\alpha}{2EI\sin 2\alpha}$$

Summarizing the constants of integration,

$$\left.\begin{aligned} A &= 0 \\ B &= -\frac{r^2M_0\alpha}{2EI\sin^2\alpha} \\ C &= \frac{r^2M_0}{2EI} \\ H &= 0 \\ K &= \frac{r^2M_0}{2EI\tan\alpha} \end{aligned}\right\} \quad [44]$$

Placing these constants in the above equations gives the desired solution of the deflection of the ring under the action of couples at the ends of the spokes.

If $\frac{du_0}{d\theta}$ is the value of $\frac{du}{d\theta}$ at $\theta = 0$ or 2α,

$$\frac{du_0}{d\theta} = B + K$$

$$= -\frac{r^2M_0\alpha}{2EI\sin^2\alpha} + \frac{r^2M_0}{2ET\tan\alpha}$$

$$= -\frac{r^2M_0}{2EI}\left(\frac{\alpha}{\sin^2\alpha} - \frac{1}{\tan\alpha}\right)$$

$$\frac{1}{r}\frac{du_0}{d\theta} = -\frac{rM_0}{2EI}\left(\frac{2\alpha - \sin 2\alpha}{1 - \cos 2\alpha}\right) \quad [45]$$

Up to this point we have carried out the solution with the symbols used by Prescott. In the remainder of the solution, Prescott's definitions for these symbols will not be retained.

Consider the equilibrium of a spoke. Let the axis of any spoke before strain be taken as a reference line with x measured along this line from the hub. Let the hub be held fixed and the ring be twisted through an angle θ; also, as shown in Fig. 29, let

P = tangential force acting on one spoke at the ring
T = couple at the ring
R = radius to ring
l = length of spoke
r = R/l
f = linear deflection of end of spoke
f_1 = linear deflection at x
E = modulus of elasticity in tension
I = rectangular moment of inertia of the spoke section
I_1 = rectangular moment of inertia of the ring section
θ_1 = angle of tangent at the end of the spoke at the ring = $\theta + \theta_0$
N = number of spokes
Q = torque transmitted by N spokes
β = angle between spokes = $2\pi/N$

Putting M_0 in [45] equal to $T/2$, $2\alpha = \beta$, $r = R$, $I = I_1$, and $\frac{1}{r} = \frac{du_0}{d\theta} = \theta_0$, we have

$$\theta_0 = -\frac{RT}{4EI_1}\left(\frac{\beta - \sin\beta}{1 - \cos\beta}\right)$$

Hence

$$\theta_1 = \theta - \frac{RT}{4EI_1}\left(\frac{\beta - \sin\beta}{1 - \cos\beta}\right) \quad [46]$$

The bending moment on the spoke at a distance x from the hub is

$$M = T + P(l - x)$$

so that

$$EI\frac{d^2f_1}{dx^2} = T + Pl - Px$$

Integrating, this gives

$$EI\frac{df_1}{dx} = (T + Pl)x - \frac{Px^2}{2}$$

$$EIf_1 = (T + Pl)\frac{x^2}{2} - \frac{Px^3}{6}$$

where the constants of integration are zero, since $\frac{df_1}{dx} = 0$ and $f_1 = 0$ at $x = 0$.

At $x = l$, the slope of the deflection curve is $\tan\theta_1$ which can be taken as θ_1 since θ_1 is small. Therefore the last two equations can be written as

$$EI\theta_1 = Tl + \frac{Pl^2}{2}$$

$$EIf = \frac{Tl^2}{2} + \frac{Pl^3}{3}$$

Solving these two equations for P and T and putting $R\theta = f$ and $r = R/l$, gives

$$P = \frac{12EI}{l^3}\left(r\theta - \frac{\theta_1}{2}\right) \quad [47]$$

$$T = -\frac{12EI}{l}\left(\frac{r\theta}{2} - \frac{\theta_1}{3}\right) \quad [48]$$

Using Equations [46], [47] and [48] we may eliminate θ_1 and obtain P and T in terms of θ. This gives

$$P = \frac{12EI\theta}{l^2(1 + K)}\left(r - \frac{1}{2} + \frac{rK}{4}\right)$$

$$T = -\frac{12EI\theta}{l(1 + K)}\left(\frac{r}{2} - \frac{1}{3}\right)$$

where

$$K = \frac{rI}{I_1}\left(\frac{\beta - \sin\beta}{1 - \cos\beta}\right)$$

The torque transmitted by N spokes is

$$Q = N(T + PR)$$

$$= -\frac{12NEI\theta}{l(1 + K)}\left(\frac{r}{2} - \frac{1}{3}\right) + \frac{12NEIr\theta}{l(1 + K)}\left(r - \frac{1}{2} + \frac{rK}{4}\right)$$

Therefore, since the stiffness factor C for the rim relative to the hub is Q/θ, we have

$$C = \frac{12NEIr}{l(1 + K)}\left(r + \frac{1}{3r} - 1 + \frac{rK}{4}\right)$$

where

$$K = \frac{rI}{I_1}\left(\frac{\beta - \sin\beta}{1 - \cos\beta}\right)$$

The Diesel-Engine Tractor and Excavator

By C. G. A. ROSEN,[1] SAN LEANDRO, CALIF.

Special emphasis is given in this paper to the combustion process for Diesel-tractor service. A variety of fuel systems were installed and tested in a rugged engine of flexible design. General results of the tests are given. Accessories are considered extremely important in tractor service, and the author discusses air cleaners, oil filters, and engine starters. The problems discussed can be applied to other traction equipment including excavators. Reliability at all loads and speeds within the engine operating range is paramount, and maintenance costs must be kept low.



IF A MECHANICAL product is to satisfy economic necessity, service demands must dictate engineering thought. The operating-field problems will then be reflected in the manufactured article. Consider, for example, the tractor. The withdrawal of tractors from any "Caterpillar" warehouse may place one machine in the hands of native labor in the torrid swamps of an African jungle; another tractor may be harnessed to a snow-plow to clear the roads of civilization in freezing weather; a third unit may be destined to travel in an enveloping cloud of dust over fields of pulverized soil under the control of a begoggled and masked operator; still a fourth tractor may be called to drag logs in the woods, careening and zooming drunkenly over rough country, sometimes up grades as steep as 75 per cent; another may be forced to consume Russian kerosene of an initial boiling point of 340 deg. and an end point of 610 deg.; a Cuban purchasing agent may require a sixth tractor to deliver its power on fuels of 104 deg. initial boiling point, but an end point of 600 deg. These six tractors from the same assembly line are expected to perform continuous heavy duty under diametrically opposite circumstances.

From such fields and others the call is heard for the Diesel-engine tractor. Two avenues of economic influence are revealed in the demand:

1 The need for an internal-combustion power unit which will burn fuels economically available at points of world distribution, which fuels are not readily combustible in carburetor engines

2 The need for an internal-combustion power unit which will have such economical consumption of cheap fuels as to cause a reduction in operating cost when compared with carburetor engines.

The Diesel engine is a prime mover designed essentially to show high economy in the consumption of cheap-grade fuels. That is its inherent reason for existence. The Diesel engine fitted to a tractor must be a satisfactory oil-burning unit. Its practical value, however, cannot be confined to those districts where specification fuels are obtainable. Desirable as specification fuels are, the use of available fuels cannot be disqualified by the manufacturer. In communities where gasoline sells for 75 cents to 80 cents a gallon, the economical consumption of cheaper grade fuels offers the high-speed Diesel engine its most lucrative field of economic reward.

The Diesel-engine excavator likewise shares this economic promise. It is also a product of world-wide distribution and, like the tractor, is encumbered with fuel-burning problems. The Diesel-driven excavator is subject to high peaks of overload and must qualify for variable-load operation. Sudden demands of power are required of the internal-combustion unit, and the questions of labor are pertinent. Although the excavator Diesel need not of necessity be light in weight, yet it has definite limitations as to space. The high-speed Diesel is therefore desirable for this type of service, particularly in the smaller units.

The tractor Diesel engine is also to be classed with high-speed engines and medium light-weight construction. Continuous power must be developed at relatively high b.m.e.p. in designs weighing 40 lb. per b.hp. or under. In substituting a Diesel engine for a gasoline engine of the same number of cylinders, the speed of the Diesel must be increased from 20 to 30 per cent for the same weight limitations. It therefore is safe to say that the tractor engine is subjected to all of the problems incumbent upon the excavator engine, but is penalized by even more severe service demands. Some commercial engines offered for excavators are reasonably successful, but are not sufficiently satisfactory for the more exacting duties required of the tractor.

The track-type tractor is a combination of many difficult problems. It must be capable of continuous operation at full load for indefinite periods of time, in all sorts of weather and environments, and in the hands of labor capable of inflicting an unusual amount of abuse. For the purpose of illustrating the severest problems involved in mobile industrial equipment, the tractor Diesel engine will be used as an example.

In view of the demands made for Diesel track-type tractors, the Caterpillar Tractor Company, about three years ago, organized a definite program of Diesel-engine investigation. The examination of the oil-engine field at that time revealed the fact that no unit was available which could be installed directly to satisfy caterpillar-tractor service. A number of commercial high-speed engines were purchased both here and abroad, to determine their possibilities for tractor use. Other units were observed under field operation and inspections were made of high-speed engines applied to extraneous services. Careful tests were made to obtain the status of commercial operating

---

[1] Diesel Engineer, Caterpillar Tractor Co., Mem. A.S.M.E. The author was graduated from Cogswell Polytechnic College and University of California with degrees of B.S. and M.E. Served in machine shop, forge shop, foundry, and drafting room on oil-engine construction and repair. Was Diesel engineer with Dow Pump & Diesel Engine Co., 1915 to 1922, terminating service as chief engineer. Engaged in practice as consulting and research engineer, 1922 to 1929, on design for Diesel-engine builders, on maintenance and installation for motorship and power-plant operators, on installation for Diesel-engine accessory and auxiliary-equipment manufacturers. Was instructor in Diesel engines for University of California Extension Division for five years. Served as Pacific Coast editorial representative for *Motorship*. Contributed articles on Diesel engines for *The Log* and *Pacific Marine Review*. Author of correspondence-course textbook "Marine Diesel Engines," published by the American Society of Marine Engineers, and co-author of University of California correspondence course on Diesel engines. From 1929 to the present, engaged in the development of mobile Diesels for Caterpillar Tractor Company.

Presented at the Third National Oil and Gas Power Meeting, State College, Pa., June 12 to 14, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

quality. The constants of design were determined and operating difficulties and shortcomings were noted.

At the conclusion of these tests it was decided that no available Diesel could be substituted for the established gasoline engine and meet the exacting service demands. Problems in combustion were not sufficiently solved to cope with the fuels which were desired for consumption. The general designs were unsuited to the environments of operation and were incapable of guaranteeing reliability under tractor-service conditions. If these Diesels could have been applied to the tractor, the tractor selling price would have increased 65 to 85 per cent. Maintenance costs in some cases were estimated to be higher than could be offset



FIG. 1 TRACTOR UNDER NATIVE OPERATION IN AFRICAN JUNGLE



FIG. 2 PLOWING SNOW IN FREEZING WEATHER WITH TRACTOR EQUIPMENT

by the fuel economy. It was therefore thought advisable to experiment with fuel systems and to organize a program of investigation for the development of an oil-burning engine of satisfactory combustion characteristics. Neumann puts it clearly when he states:

"The development which controls the high-speed prime mover today forces us to bring to the fore the dynamics of the combustion process."

The fuel system should permit the tractor Diesel to maintain its full rated load continuously with economic fuel consumption, and yet be sufficiently flexible to operate at idling speeds and varying loads without disturbing the combustion process or creating difficulties with it.

In achieving a satisfactory fuel system, the background of tractor-engine designing experience would then be available to encircle the fuel system in a suitable structure constrained by materials of proved character and selected heat treatments.

To carry out this program, a rugged engine of flexible design was chosen which would permit the installation of a variety of fuel systems. A test schedule was established which could evaluate each fuel system tested and form a basis of comparison for the ultimate judgment of the systems. Each fuel system was given preliminary trial runs to bring combustion efficiency to its highest value and to make combustion control as smooth as obtainable. Three distinct types of precombustion chamber



FIG. 3 TRACTOR HAULING ON A DUSTY ROAD



FIG. 4 USING TRACTORS FOR LOGGING OPERATIONS IN ROUGH COUNTRY

systems were tested, and three different methods of direct injection were applied to the test engine.

The first system, indicated in Table 1 as "A—Precombustion," utilized a variable injection-timing mechanism. The "B—Precombustion" chamber system was fitted with a variable cut-off control, while the "C—Precombustion" chamber system was fitted with controls on both injection timing and cut-off. The "D—Direct Injection" system was provided with an open-nozzle fuel system arranged with suitable turbulence. The "E—Direct Injection" system failed to function properly. The "F—Direct Injection" system employed a multiple-orifice constant-pressure fuel system. The schedule of the week's test run is briefly outlined as follows:

Full-Load Test From Monday, 10 A.M., Until Wednesday, 10 A.M. With the controls of the fuel system set for full-

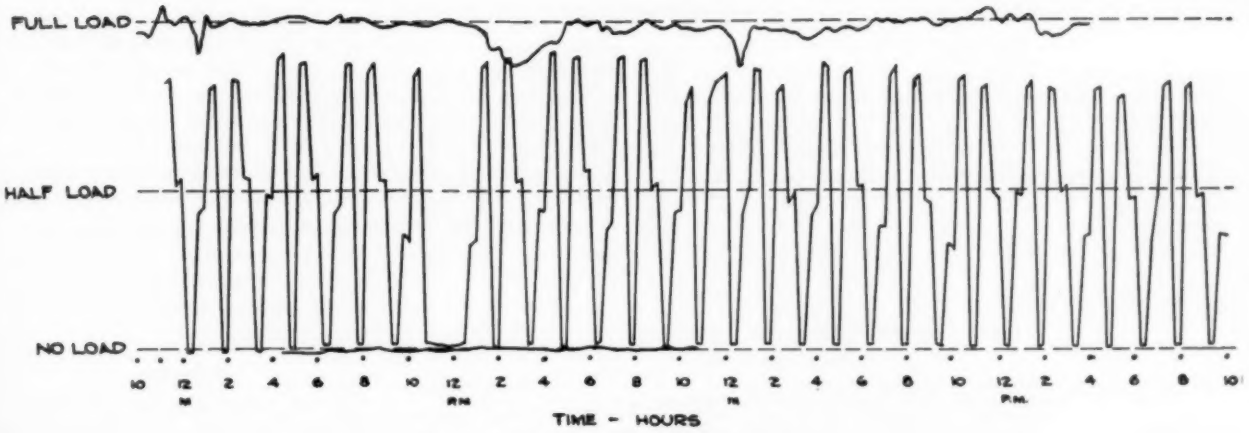


FIG. 5 LOAD CURVES FOR FUEL SYSTEM A

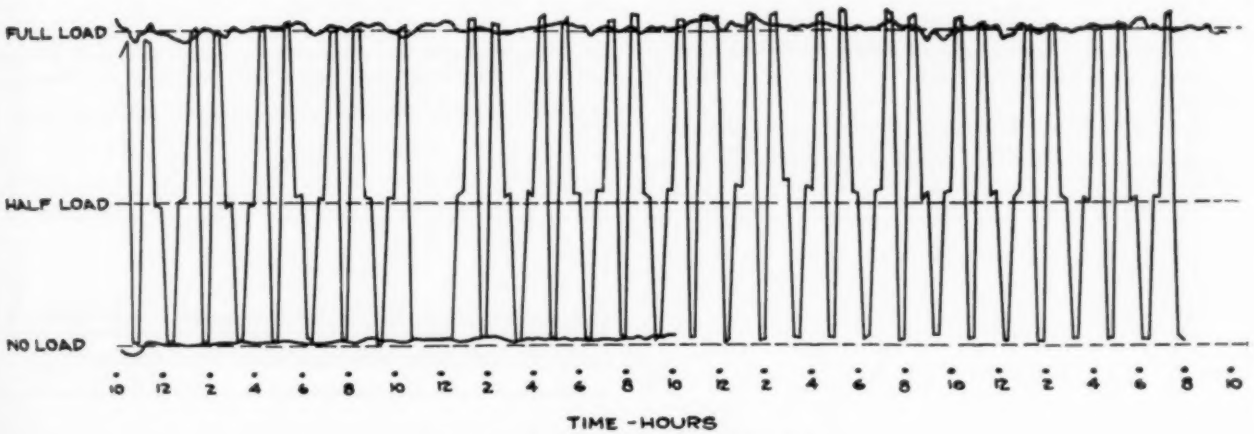


FIG. 6 LOAD CURVES FOR FUEL SYSTEM B

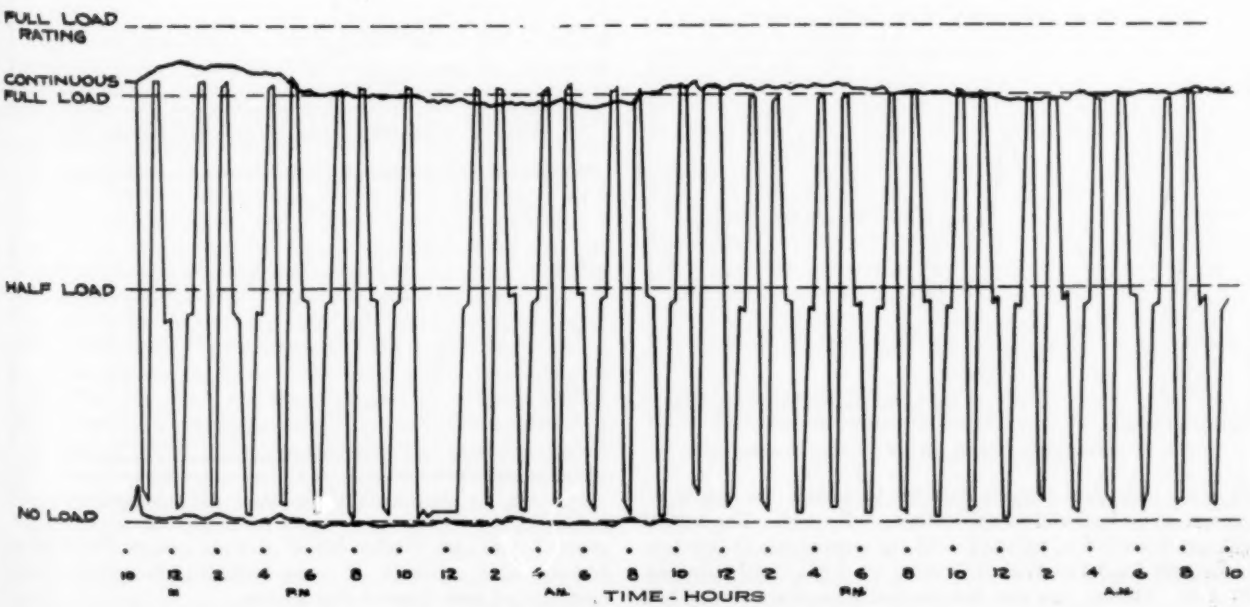


FIG. 7 LOAD CURVES FOR FUEL SYSTEM D

TABLE 1 COMPARATIVE TEST RESULTS OF PRECOMBUSTION AND DIRECT-INJECTION SYSTEMS

Fuel system terms	Fig. No.	Fuel press., lb. per sq. in.	Speed, r.p.m.	I.hp.	M.i.p., lb. per sq. in.	Fuel cons., lb. per i.hp-hr.	Exh. temp., deg. fahr.	Max. press., lb. per sq. in.	Mech. effic., per cent	Remarks
A—Precombustion	8	1250	792	20.35	116.5	0.382	955	801	73.0	Rated load
A—Precombustion	8	1250	878	23.05	120.5	0.484	1075	805	74.1	Max. load
B—Precombustion	9	900	801	18.55	106.0	0.376	820	850	78.4	Rated load
B—Precombustion	..	900	925	25.90	115.5	0.398	1005	760	77.2	Max. load
C—Precombustion	10	4350	802	21.00	119.0	0.340	875	510	[a]	Full load
C—Precombustion	..	2600	794	18.90	108.0	0.374	910	545	[a]	Rated load
D—Direct injection	11	[b]	893	26.20	103.0	0.362	935	610	77.1	Full load
D—Direct injection	..	[b]	1031	33.20	113.0	0.425	1035	815	62.8	Max. load
F—Direct injection	..	6000	797	19.60	111.5	0.377	765	875	[a]	Full load
F—Direct injection	12	6000	804	23.60	133.5	0.308	865	800	[a]	Max. load

[a]—Fuel pump, motor driven.
[b]—Open-nozzle fuel system.

TABLE 2 LUBRICATING-OIL TEST DATA

(Basis 124 hours' operation)

Fuel system	Lub. oil consumption, gal. per 1000 hp-hr.	Solid residue, per cent		Ash, per cent		Carbon residue, per cent	
		Before test	After test	Before test	After test	Before test	After test
A—Precombustion	1.70	0.947	5.40	0.03	0.095	0.92	5.30
B—Precombustion	1.59	1.05	3.08	0.014	0.386	1.04	2.69
D—Direct injection	1.11	0.996	2.093	Trace	0.088	0.996	2.005

load operation, the engine was permitted to run without further adjustment for the above-designated period. During this run data were obtained according to a printed schedule, and all variations and incidents relative to operation were noted and recorded.

No-Load Test From Wednesday, 10 A.M., Until Thursday, 10 A.M. The fuel controls were set to permit the engine to operate at no load and at approximate full speed. The control was not disturbed or adjusted until the termination of this test.

Valuable Load Test From Thursday, 10 A.M., Until Saturday, 10 A.M. During this run the controls were altered every half hour in the following manner:

1 Full load for one-half hour.
2 No load for one-half hour.
3 Full load for one-half hour.
4 One-half load for one-half hour.
5 No load for one-half hour.
6 One-half load for one-half hour.
7 Full load for one-half hour, and the cycle was repeated.

Overload Test From Saturday, 10 A.M., Until Saturday, 2 P.M. The controls for this test were set for operation at 10 per cent overload and 15 per cent overspeed.

Remarks

Before proceeding with any new experiment, the lubricating oil was flushed out and replaced with new oil. Samples of both the fuel and lubricating oil were taken in bottles on Monday morning prior to the first test and on Saturday afternoon upon completion of the test run for each experiment. The compression of each cylinder was obtained prior to and after each week's test run. The fuel consumption was obtained on Monday about 2 p.m., on Wednesday at 9 a.m., and on Saturday at 1 p.m. This schedule of tests indicated the practical performance of each type of fuel system.

Tables 1 and 2 show general results obtained on the various

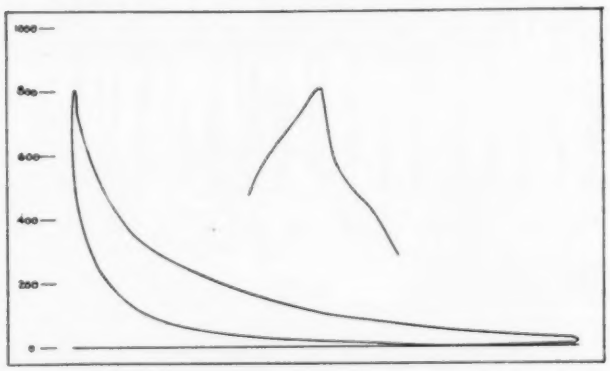


Fig. 8 Indicator Card With Fuel System A

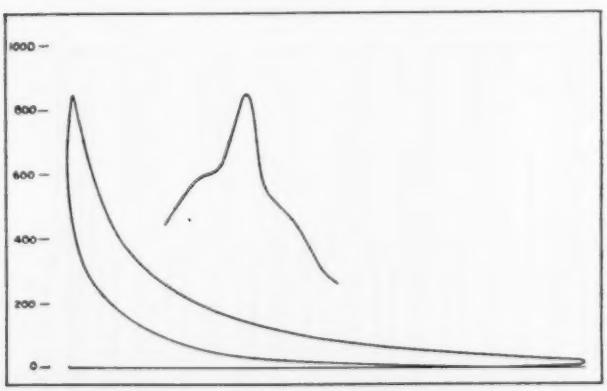


Fig. 9 Indicator Card With Fuel System B

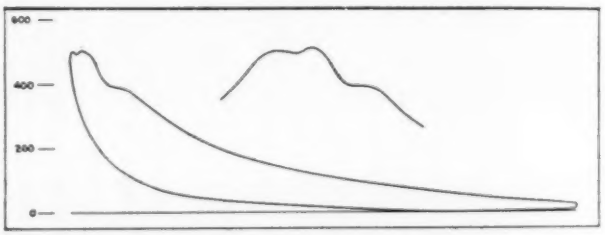


Fig. 10 Indicator Card With Fuel System C

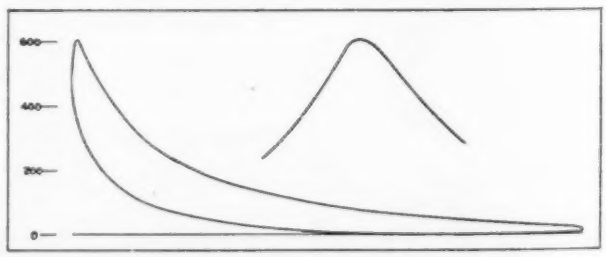


Fig. 11 Indicator Card With Fuel System D

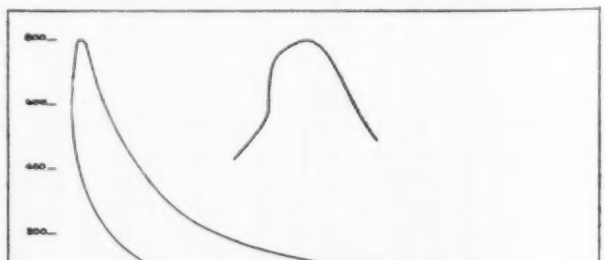


Fig. 12 Indicator Card With Fuel System F

fuel systems. The figure numbers listed in column 2, Table 1, refer to the indicator diagrams representative of loaded conditions. The 15-min. load readings taken during the week test run are graphically illustrated in Figs. 5, 6, and 7 for three of the fuel systems. These have been chosen as being representative of load variation during the runs. In Fig. 5 the full-load curve indicates considerable variation during the two days of operation. Under variable-load conditions it was impossible to reach full-load output at the throttle setting designated as

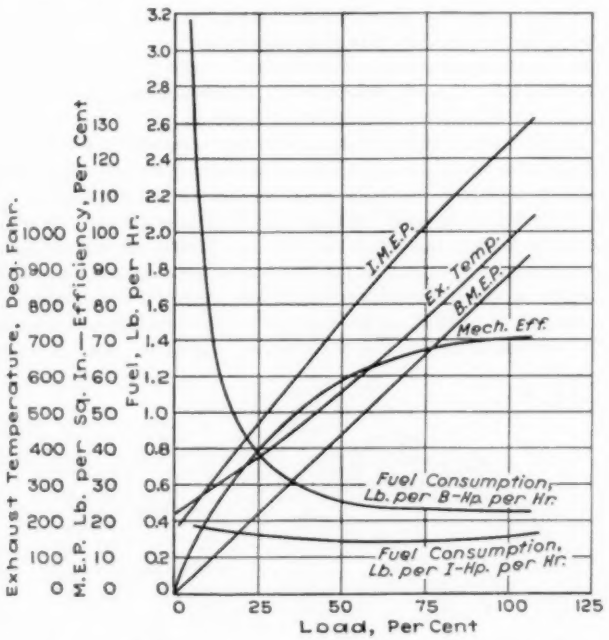


FIG. 13 PERFORMANCE CURVES WITH FUEL SYSTEM D

full load. Considerable variation is also seen in the power delivered at half-load throttle setting. The no-load curve shows fairly accurate metering of fuel. "No load" in each case was taken at full speed with the dynamometer absorbing approximately 2 hp.

Fig. 6 indicates a very uniform power output for designated throttle settings. The direct-injection system of the open-nozzle type developed a power-output curve as shown in Fig. 7. It was not deemed expedient to operate at the rated full load as combustion conditions were unsatisfactory and rough running was pronounced. It was therefore thought advisable to drop the full load so that the exhaust was fairly clean and operation fairly smooth. This procedure was followed in the case of each fuel system. It was considered imperative to determine full load at that value which would produce a clean exhaust at commercially desirable b.m.e.p. After uniform conditions were established in the preliminary tests prior to the week test runs, efforts were made to produce as smooth an operating condition as it was possible to obtain. To secure satisfactory operating quality in units which had manufacturing possibilities was the paramount ideal. Some combustion chambers are difficult to manufacture; other fuel systems require such extreme accuracy and delicate adjustments as to be prohibitive so far as manufacturing costs are concerned.

The results of these tests revealed the advantages and disadvantages of each fuel system in so far as concerns its ability or failure to meet the standards set up for oil-burning motive power by the Caterpillar Tractor Company. Some of the weaknesses may be summarized as follows:

1 Inability to maintain full load over extended periods of operation
2 Reduction of combustion efficiency at light loads
3 Disturbance of combustion process under variable-load operation
4 High maintenance cost
5 Tendency to produce rough running.

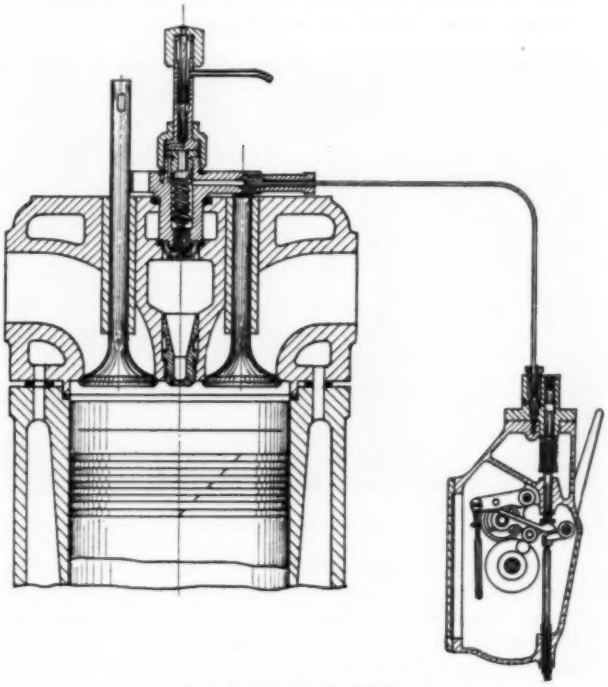


FIG. 14 A—PRECOMBUSTION

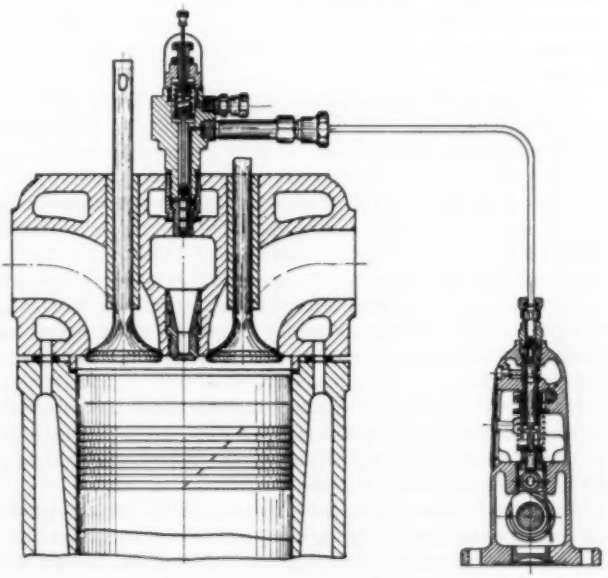


FIG. 15 B—PRECOMBUSTION

In spite of these undesirable qualities, some systems revealed sufficient promise to warrant the further development of the high-compression oil engine. A correlation of the desirable characteristics of all fuel systems indicated the path of development for a more efficient method of burning fuel, incorporating a control of practical operating quality.

It is generally understood that thermal efficiency and mean effective pressure rise as maximum pressures increase. In tractor work fuel consumption is not as pertinent as operating quality. It is therefore expedient to drop maximum pressures provided colorless exhaust and reasonable exhaust temperatures result. Experimental work has indicated a procedure for reducing maximum pressures and for controlling the increment of increase in pressure rise during the combustion period by the correlation of cam profiles with orifice design. Smooth operating

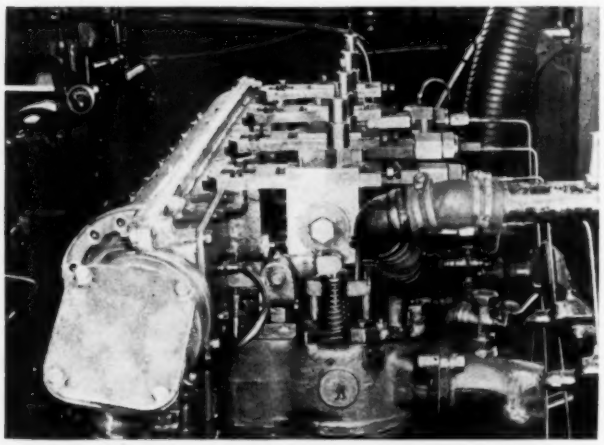


FIG. 16 C—PRECOMBUSTION

characteristics are intimately related to the rate of feed of atomized fuel, thus influencing the rate of combustion within the cylinder. The tests have also brought out inherent advantages in the precombustion chamber system. These can best be expressed by quoting Neumann from the *V.D.I.*, Sept. 8, 1928:

"In the high-speed Diesel engine the precombustion chamber process must be of advantage in the rapid heat transformation of the injected fuel by good distribution by means of high temperature and lively turbulence which may be obtained (in the engine) by simple means.

"The more rapid the formation of gas follows the inception of ignition through the breaking up of the molecule, the faster will be the ensuing combustion and of less consideration will be the heat loss to the surrounding media.

"The development of the precombustion-chamber engine to its present state took a long period of time. The beginning of its development goes back to the activities of Diesel. The top of this ascending line has not yet been reached as so often the work in actual practice advanced beyond the scientific knowledge. It is hoped, though, that the advancing knowledge into the working process of this engine will aid in giving a more efficient use of fuels in the future.

"In the use of heavy oils with high ignition points and with high carbon content, the precombustion-chamber engine is superior to the direct-injection engine; its surety of performance under severe service conditions is beyond question."

The ideal fuel system appears to contain the advantages of the common-rail and pump-injection methods of fuel injection.

Experimental work reflects the history of Diesel-engine fuel systems. The air-injection process has been in effect for many years, yet each size and type of engine requires test-stand alterations to produce the desired thermal results. Practically all solid-injection fuel systems have been developed by the "cut-and-try" method. If the high-speed engine is to enjoy manufacturing privileges, spray-research equipment is essential to guarantee uniformity of operating performance. Such factors as time distribution of spray, space distribution of spray, and pressure variation in oil lines and channels, require the use of measuring equipment for determining the control of the variables involved and also to cheapen manufacture. It is difficult to predetermine limits in finishes, tolerances, and channel courses and contours unless definite knowledge of the influence of these factors upon engine functioning is determined.

Burstall is credited with expressing the thought at the London World Power Conference that the physical and chemical questions of combustion must be solved through scientific work unaffected by commercial considerations. This statement is true and the oil-engine manufacturers laud the excellent work of Pennsylvania State College and Langley Field in spray research. The manufacturer, however, can apply only the fundamentals developed by these institutions to commercial practice by the intelligent measurement of functional values.

Combustion-chamber design must be influenced by the economics of manufacturing and the demands of service. The disposition of valves, the shape of piston crowns, and the contour of cylinder heads, should follow the dictates of scientific combustion-chamber design, but must yield in compromise to the pressure of practicality. Just how far this compromise should go to the benefit of economic construction, only research can decide.

Special emphasis has been given to the combustion process for Diesel-tractor service. Particular emphasis must also be

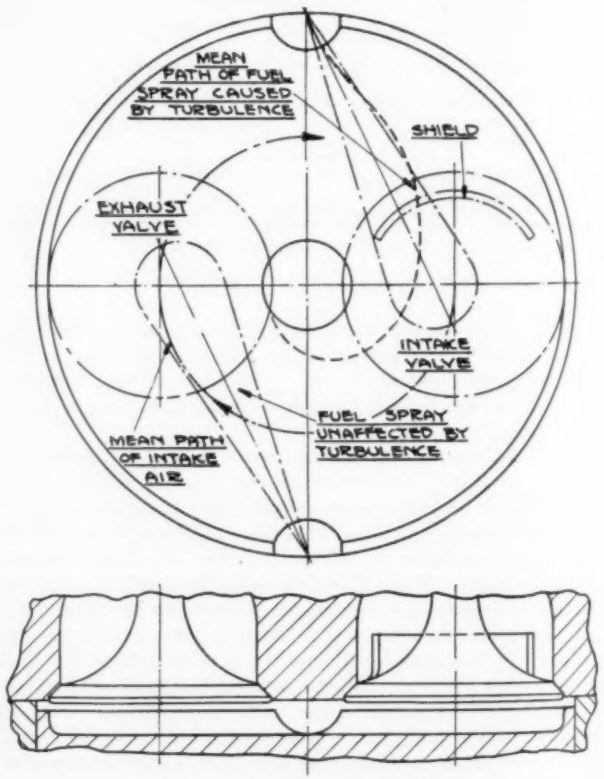


FIG. 17 D—DIRECT INJECTION

placed on the character of the design. The engine structure must have rugged strength to permit of continuous development of relatively high b.m.e.p. The total engine weight is governed by definite limits for traction equipment. The crankshaft and rotating masses must be free from severe torsional vibrations and be so balanced as to prevent structure vibrations under normal combustion processes.

In tractor-engine design, many problems are incurred in endeavoring to obtain dust exclusion and oil retention. Good

automotive practice is not sufficient to guarantee these conditions. In the gasoline type of tractor engine, special constructions are employed for magnetos, carburetors, and other accessories. The general design must permit of accessibility to vital parts and allow adjustments without risk of dirt contamination. Piping must be enclosed and control shafting and extensible parts must be protected from branches of trees and other outside agencies which might tend to tear off parts if unprotected. In the Diesel where complete enclosure is desirable, measures must be taken to prevent fuel contamination of the lubricating oil through the leakage of plungers, vents, etc.

Accessories are extremely important in tractor service inasmuch as precautionary methods of a strict character must be made to prevent dust from entering the engine structure and to minimize the amount of dust carried into the engine cylinders. Dust seals are therefore of vital importance, and all openings and through shafts must have packed seals under constant pressure. There is no more important part on the tractor than the air cleaner, for without it, the operation of the unit would fail in a very short time. Years of experience have been devoted to perfecting efficient air cleaners. In connection with the Diesel, the air cleaner presents an even greater problem. The relatively lower fuel-air ratio in the Diesel, as compared to the gasoline engine, requires every desirable means to increase the total volume of air available for combustion within the cylinders. Normally the fan and air cleaner consume 10 per cent of the engine power. The volumetric is reduced probably 2 per cent, and the best cleaners under severe operating conditions are about 90 per cent efficient in dust exclusion. The commercial necessity and the technical desirability of increasing the horsepower-weight ratio in marine Diesel engines has fostered the development of supercharging methods. Undoubtedly some commercial means for combining air-cleaner operating with supercharging methods will raise the b.m.e.p. without deleterious results. In the McLaren wheel-type of plowing-tackle tractor, superchargers are employed for Diesel engines for boosting the peak power under overload conditions. It is thus possible to obtain an increase in power of 35 per cent without increasing heat stresses or encountering valve difficulties. The urgent need for increased weight of air to support the combustion of more fuel, encourages the development of pressure-charging methods for air cleaning and cylinder filling

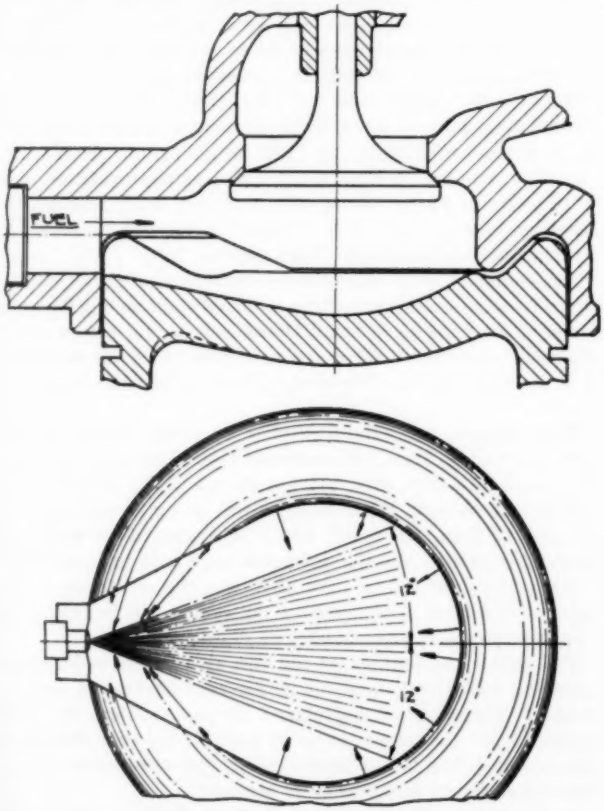
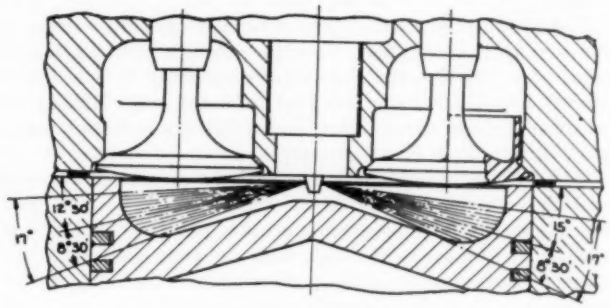


FIG. 18 E—DIRECT INJECTION



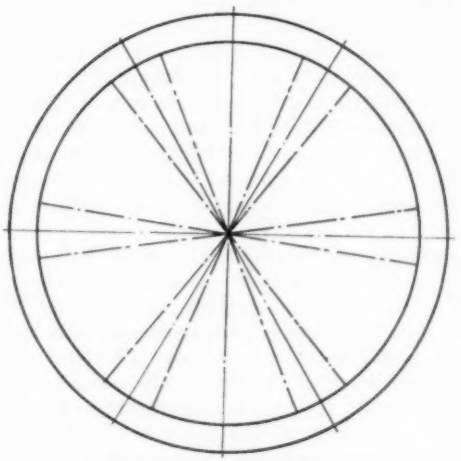


FIG. 19 F—DIRECT INJECTION

Fuel-oil filtering on mobile Diesel equipment urges investigation of fuels available at points of world distribution. Metal and cloth types of strainers each have their advantages, but have not been entirely effective in preventing foreign material from entering the fuel nozzles. The centrifuge has become an established institution on board ship and in power plants for the purification of fuel oil, but its application to tractor engines presents a difficult problem in the design of driving means to function under variable-speed operation. The opportunities for dust contamination when filling tanks in open fields and dusty country, or the risk of water absorption in moist weather, are ever present in tractor operation. Some simple method of eliminating these hazards from the engine fuel system must be developed.

The function of the lubricating-oil cleaner is sufficiently important to warrant careful study of this accessory. Under ordinary circumstances considerable surface is required in the filter-type of cleaner, as the impregnation of carbon in the minute openings reduces the effective area rapidly. Again, the centrifuge principle would be desirable, yet the added cost of such a unit may not show sufficient economy in lubricating-oil consumption to warrant its application on tractors. In the gasoline-engine tractor it is sometimes necessary, under severe service conditions, to change the lubricating oil once daily due to dirt contamination. A Diesel would not be expected to have the oil changed oftener.

Generally speaking, the Diesel can be considered as being more extravagant with lubricating oil. However, economies can be effected under clean-combustion operation. Fig. 13 indicates the influence of various fuel systems on lubricating oils. The most detrimental effect on lubricating oil is usually inflicted during variable-load and idling operation. Copper tubing for lubricating-oil conduits has also been found to accelerate the building up of acid numbers. Relatively large sump capacity permitting ample rest intervals to the oil is a helpful factor in maintaining good lubricating quality.

The problem of engine starters is rather complex and calls for continued development to reduce manufacturing cost. In most tractor service, electrical starters are out of the question as storage batteries will not stand up under the rough jostling and weaving of the tractor. Even electric-lighting systems are supplied directly from Bosch generators.

Air starting is not considered entirely satisfactory as dust and air-compressor valves do not work together harmoniously. Even in marine installations it is usual to run a suction line from the air compressor to the boat deck for pure clean air. In a tractor installation an auxiliary air compressor, gasoline engine-driven, together with air tanks, high-pressure lines, and valve controls, would be required.

A gasoline engine directly connected to the engine through a Bendix drive appears to be a far simpler arrangement. Furthermore, if starting is difficult due to cold weather or leaking plungers, considerable time is wasted waiting for the replenishment of storage tanks by the auxiliary air compressor in the air-starting system. In the direct-driven arrangement, continuous power is available to rotate the main engine at good starting speeds.

The hand-actuated impulse starter would warrant consideration if its cost could be reduced. Other methods of hand cranking have not proved dependable or given promise of reliability. The explosion of powder cartridges for starting impulse has been too erratic for unskilled labor to control. The chief objection to gasoline-engine starting methods is the necessity of carrying two grades of fuel. Its greatest asset, however, is positive starting. Improvement in starting methods is particularly desirable as a means toward reducing the cost of the starter equipment.

This paper has endeavored to present a general picture of the problems involved in the development of a Diesel engine for track-type tractors. A thorough solution of these problems has not yet been attained. Sufficient progress, however, has been made to direct the course of future development and to stimulate hope for the ultimate arrival at the desired goal.

The problems which have been discussed can also be applied to other traction equipment including excavators. However, it is believed that engineering scrutiny must be more severe in the case of track-type tractors.

In the development of the high-speed Diesel, the importance of good operating quality cannot be overemphasized. Commerical pressure demands designs capable of quantity production; the dictates of field service place a premium on simple, foolproof construction; reliability at all loads and speeds within the engine operating range is paramount; maintenance cost must be held to a minimum. The sum total of all of these factors is an equation in combustion research applied to practice.

A commercial Diesel-research organization must have its aspirations soaring in realms of scientific thought, but must also have its hand in intimate contact with economic necessity.

Discussion

A. M. Rothrock.[2] This paper is particularly noteworthy in that it gives actual experimental results and explains in detail the types of combustion chambers and injection systems used. To those who are conducting research on Diesel engines information of this type is important so that the progress may be known which is being made by the commercial organizations. There are two questions which the writer would like to ask. When using a combination common-rail and pump system, is a mechanically operated injection valve used? At what speeds will the engine idle satisfactorily when using this injection system? It is particularly interesting that the author found it necessary to use a combination common-rail and pump-injection system in order to obtain satisfactory idling characteristics. The writer also came to the conclusion that some such system is necessary to design an injection system whose operation is independent of engine speed.

Wm. K. Gregory.[3] The author states: "The relatively lower fuel-air ratio in the Diesel engine as compared to the gasoline engine requires every desirable means to increase the total volume of air available for combustion within the cylinders." Quoting further, "Normally the fan and air cleaner consume 10 per cent of the engine power. The volumetric is reduced probably 2 per cent and the best cleaners under severe operating conditions are about 90 per cent efficient in dust exclusion."

If the writer understands correctly, the air filter referred to reduces the volume of air available for combustion by 2 per cent. At an atmospheric pressure of 14.75 lb. per sq. in., the air filter would be required to have a resistance of approximately 8.28 in. of water to effect a 2 per cent reduction in air reaching the cylinders. This seems quite excessive in view of standard air-filter practice on stationary Diesel engines.

The company with which the writer is connected has concentrated its efforts on large stationary Diesel engines, but the problem of filtering the air on small Diesels suitable for use on mobile equipment does not involve any problems which are not found in large Diesel installations except greater dust concentration in the air.

The company's standard practice on large Diesel installations is to furnish filters with sufficient area so the actual pressure drop through a clean filter will not exceed 1/4 in. of water. As dust is accumulated, this resistance will go up as high as 1 in. of water in from four to six weeks' operation, after which the filter cell is cleaned, and the resistance drops back to 1/4 in. water.

When the filter cell is quite dirty and the resistance is 1 in. water, the volume of air entering the engine will be only 1/4 of 1 per cent less than when no filter at all is used.

The author also states that the best air cleaners under severe operating conditions are about 90 per cent efficient in dust exclusion. The filters manufactured by the American Air Filter Company are guaranteed to be at least 97 per cent efficient in dust removal, and actual tests in many instances have shown a cleaning efficiency in excess of 99 per cent.

In designing a filter for the protection of Diesel engines and air compressors there are three fundamental requirements:

1 The filter must have a high cleaning efficiency, at least 97 per cent
2 The initial resistance of a clean filter must be approximately 1/4 in. water, and must not exceed 3/8 in. water
3 The construction of the filter must be such that a large amount of dust can be accumulated by the time the resistance increases to as much as 1 in. of water.

What is meant by a large amount of dust may be indicated by the following test data:

A 20 in. by 20. in. by 4 in. filter cell having an initial resistance

[2] Associate Physicist, N.A.C.A. Langley Field, Va.

[3] Sales Manager, American Air Filter Co., Inc., Louisville, Ky.

of 1/4 in. water, will catch approximately 1 lb. of atmospheric dust before the resistance reaches 1 in. of water. It requires approximately 3/4 lb. of oil, or 0.85 pint, to saturate a pound of dust. It is therefore apparent that the oil-holding capacity of a filter has a definite bearing on its cleaning efficiency and dust accumulating capacity.

As mentioned previously, the company with which the writer is connected directs its attention primarily to large Diesel engines, gas engines, and air compressors, but several years ago a few contacts with manufacturers of mobile equipment were established which have led to the belief that most of them do not appreciate one fact emphasized by the author, who states: "There is no more important part on the tractor than the air cleaner, for without it the operation of the unit would fail in a very short time." To most manufacturers, however, when considering an air filter the first and most important consideration appears to be low cost; second, they want the filter to be light in weight and to occupy very little space; and third, they want a filter that will require little or no attention.

Cleaning efficiency, resistance to air flow, and dust accumulating capacity, regarded as prime requisites of a satisfactory filter, are placed in a position of secondary importance. During the past two or three years the importance of air filters has become more generally recognized, and in justice to the manufacturers their viewpoint has probably been changed.

There is no reason why a Diesel or gasoline engine installed on a tractor or excavator cannot be equipped with a filter which will not restrict the volume of air more than 1/4 of 1 per cent, and which will have a minimum cleaning efficiency of 97 per cent. Under severe dust conditions it may be necessary to clean a filter of this type as often as once per day, although in many types of service it should not be necessary to clean the filter oftener than once per week. The whole secret of efficient air cleaning and low resistance to air flow lies in providing a filter of ample size, and in view of the fact that it is an accessory which has an important bearing on the service rendered by the engine, the cost, weight, and maintenance required should be secondary considerations.

OGP-53-4

Heat Recovery From Internal-Combustion Engines

By THOS. CLARKSON,[1] LONDON, ENGLAND, AND WM. BRADFORD,[2] EDGE MOOR, DEL.



THOS. CLARKSON

Citing the heat losses of locomotives and steam turbines, the authors aver the internal-combustion engine the most efficient mechanism for converting heat into useful work; yet even in this, two-thirds of the heat is wasted in exhaust gases and in cylinder jackets. The need for further heat conservation is urgent. In considering the recovery of heat from exhaust gases of internal-combustion engines, the authors lay down seven basic conditions. Adding to these limitations the avoidance of back pressure and the provision of a simple way of decarbonizing the heating surface, the designing of the ideal waste-heat boiler is not a simple matter. The evolution of the "thimble tube" boiler after 30 years of experience is described.



WM. BRADFORD

HEAT being one of the most valuable commodities with which engineers have to deal, it is nothing less than tragic that so much of it is wasted and lost. For example, the ordinary steam locomotive wastes over 90 per cent of the B.t.u. which are released in its firebox. Modern steam-turbine plants, with high pressure, superheating, condensing, preheating the air, and improved feedwater heating, are much more efficient than the ordinary steam locomotive, and go far to equal in thermal efficiency the internal-combustion engine, which may be taken as the most efficient appliance for converting heat into useful work. Yet it has to be recognized that, even with internal combustion, about two-thirds of the heat is wasted —about one-third in exhaust gases and another third in the cylinder jackets; therefore the need for further heat conservation is urgent.

The heat carried away in the water from cylinder jackets is necessarily of a low order; as there is a practical limit to the temperature to which jacket water can be permitted to rise. The exhaust gases are at a considerably higher temperature (especially in the case of engines of the four-cycle type), and therefore the recovery of heat from exhaust gases presents greater possibilities. In some cases hot jacket water may be usefully employed as feed to supplement the heat recovered from exhaust gases.

In considering the question of heat recovery from exhaust gases of internal-combustion engines, the following basic conditions must be recognized:

1 The maximum temperatures available are too low to have useful radiant effect; therefore means had to be devised to utilize "convection" in the most effective manner, and to secure the most effective "intimacy" between hot gases and heating surface.

2 The most convenient form in which to distribute recovered heat is either as steam or as hot water; therefore the heat-recovery apparatus naturally takes the form either of a boiler or a heater.

3 A boiler or a heater to fulfil ideal conditions for this service should have *the maximum capacity for being neglected and forgotten.*

4 It should also be effective in muffling or silencing the exhaust.

5 It should have the faculty of being operated "amphibiously," by which is meant that it should be capable of operating without water, at times when heat recovery is not needed, and at the same time carry on its other useful function of silencing.

6 The ability to withstand sudden and violent changes of temperature with impunity is a desirable quality of robustness.

7 Smallness of size and weight are also important factors, especially on shipboard and in railroad service.

When to all of these conditions is added the avoidance of back pressure (especially with engines of the two-cycle type) and the provision of a simple and effective way of decarbonizing the heating surface, it will be recognized that the problem of designing an ideal waste-heat boiler for use with internal combustion engines has not been so simple a matter as might at first sight appear.

The boilers which have been used previously for waste-heat recovery have been of ordinary type; generally multi-tubular, or sometimes of the Scotch-marine type. It is not surprising, therefore, that the science of waste-heat recovery has not made

---

[1] Consulting Engineer. Mr. Clarkson commenced engineering apprenticeship in 1877. In 1885 won a first place in Whitworth Scholarships. In 1888 was graduated in metallurgy from the Royal School of Mines. Was then appointed lecturer in metallurgy at King's College, London, which appointment was resigned in 1890 to develop his inventions. In 1903 started the first Clarkson steam-bus service at Torquay in Devon, followed by services in London and elsewhere. In 1910 he founded the National Bus Company, which has continued in operation ever since and is now allied to the railroad groups. Mr. Clarkson is a Past-President of the Institution of Automobile Engineers and a Member of Council of the Newcomen Society.

[2] Director of Engineering, Edge Moor Iron Company. Mem. A.S.M.E. Mr. Bradford was graduated from the Engineering Department of the University of Wisconsin in 1904, remaining as instructor in experimental engineering for one year. Was with Henry L. Doherty & Co. in public-utility operating work at Denver, Col., and Lincoln, Neb., 1905 to 1910; chief electrical engineer, Laclede Gas Light Company, St. Louis, Mo., 1910 to 1917; general power-plant design and construction work, 1917 to 1920; joined Engineering Department of Edge Moor Iron Company, 1920. For the past three years Director of Engineering, in charge of engineering, production, and development.

Presented at the National Oil and Gas Power Meeting, University of Wisconsin, Madison, Wis., June 23–26, 1931, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

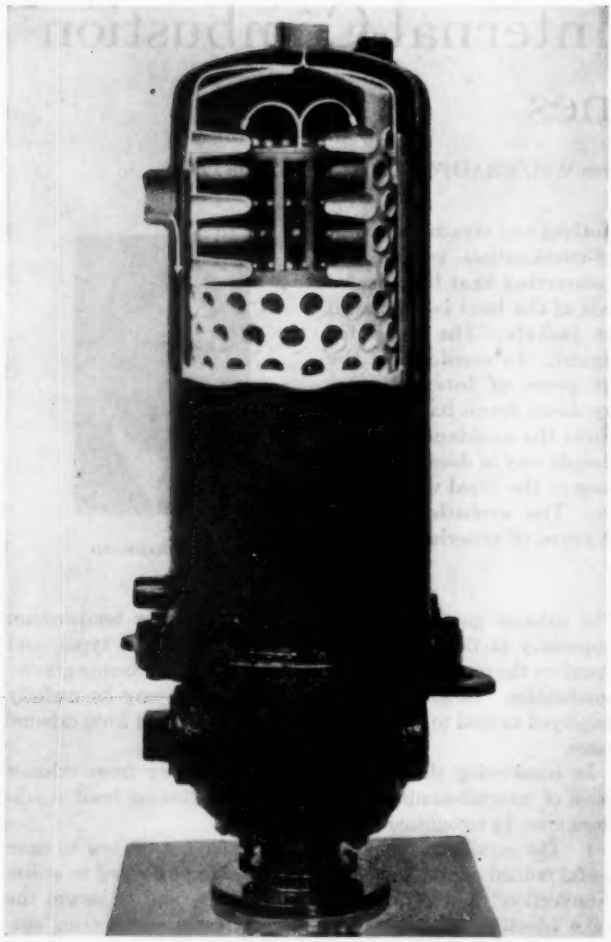


FIG. 1 "THIMBLE TUBE" HEATER, THE EXHAUST GASES ENTERING AND LEAVING AT THE BASE

more progress, as it is impossible to obtain the most effective results from such boilers. They do not function effectively as mufflers, neither do they lend themselves to decarbonizing of the heating surface by "burning off." They cannot be successfully operated "amphibiously," and they will not stand up against sudden changes of temperature.

Over 30 years' experience with boilers of many types and designs finally led to the evolution of the "thimble-tube" boiler, which in its primitive conception is an inversion of the "Porcupine" boiler. The tubes are tapered approximately radially, and in appearance resemble ordinary thimbles. Being secured at one end only, the thimbles are individually free to expand and contract, without causing strain upon the tube material or the joints. This explains the remarkable freedom, and in fact practical immunity, from tube trouble with the "thimble-tube" boiler.

The same principle of anchorage at only one end is adopted in the design of the complete boiler or heater, as in the case of individual tubes. In the case of return-flow boilers and heaters, only one end of the inner shell is fixed to the outer shell, the other end being free to expand or contract. This enables the heating surface to be decarbonized whenever necessary by "burning off," without loosening of tubes or straining of the boiler or heater.

The exhaust gases of some oil engines cause a deposit of oily soot to collect upon the heating surface of the boiler. Owing to the sticky nature of this deposit, it is not amenable to treatment by a soot blower, and the only really effective method for removal is by combustion or "burning off." In the case of four-cycle engines, with hotter exhaust gases, it usually suffices to empty the boiler of water and operate "dry" for awhile. In the case of two-cycle engines, it may also be necessary to ignite the deposit, which, as it contains about 66 per cent of oil, burns readily.

The "burning off" operation takes only a few minutes, and as soon as it is completed water can be at once readmitted without waiting for the boiler to cool; in fact, there need be no stoppage in the flow of exhaust gases over the "thimble tubes."

The most complete accessibility to the tube joints is afforded by removing the outer shell or by withdrawing the inner shell. Every square inch of heating surface is then open to direct vision. This is an important advantage also during construction. It has not been found necessary to weld the tubes in position; they are made a push-fit or a drive-fit into the tube plate, and they need only a slight amount of rolling. At the same time the open end is flared or belled to insure against the possibility of a tube being pushed through the tube plate by pressure and to give a stream-line effect to the flow of water and steam.

A few words may be devoted to the question of how circulation takes place within the "thimble tubes," as it is obvious that, with one end of the tube closed, the water and steam cannot circulate in a normal manner. Observations with glass "thimble tubes" have shown that when heat is applied to the outside of a "thimble tube" containing cold water, the water in contact with the heating surface rises vertically and displaces the water in the upper part of the tube, which is pushed out of the tube into the

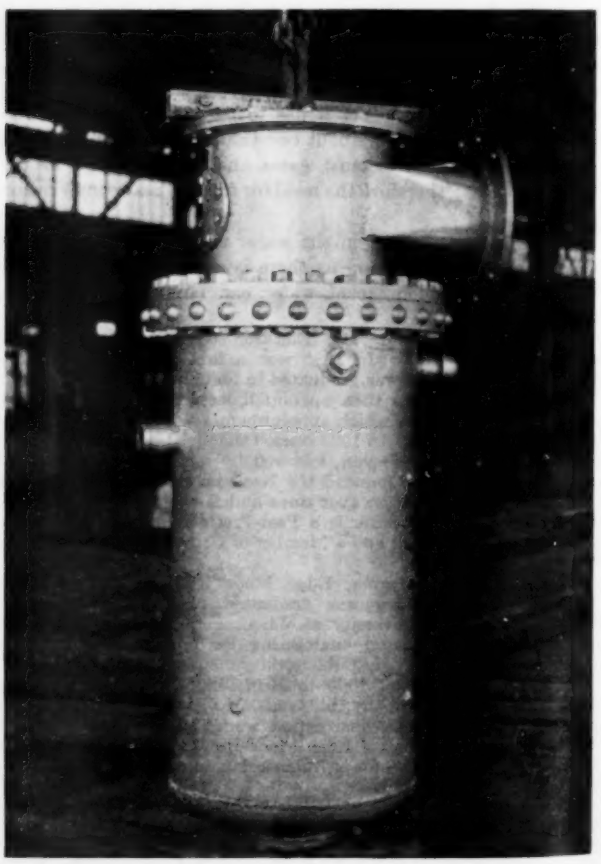


FIG. 2 "THIMBLE TUBE" HEATER INVERTED, WITH CONNECTIONS AT THE TOP

annular water space between the inner and outer shells. Another stratum of water enters the tube at the lower level, and in turn is heated, rises, and returns to the annular water space. This "convection" circulation proceeds continuously until the water reaches boiling point, when, instead of "convection" circulation, there is a copious streaming of steam bubbles from the heating surface out of the tube. When the tubes are strongly heated, this ebullition is so intense as to prevent more water entering the tube until the action has abated somewhat. The water then comes in with a rush, completely filling the tube like a water hammer, when the ebullition recommences. This pulsating action proceeds with a frequency depending upon the intensity of firing, and is helpful in preventing the formation and deposit of scale within the tubes.

The alternate rows of tubes are staggered so as to secure effective turbulence and impingement of the hot gases upon the heating surface, thus preventing the formation of inert films. The spacing of the tubes is calculated to give suitable mass velocity without causing back pressure on the engine.

Back pressure appears to be a source of anxiety to some engine builders, especially in the case of engines of the two-cycle type. It is important, therefore, to show that with a properly designed

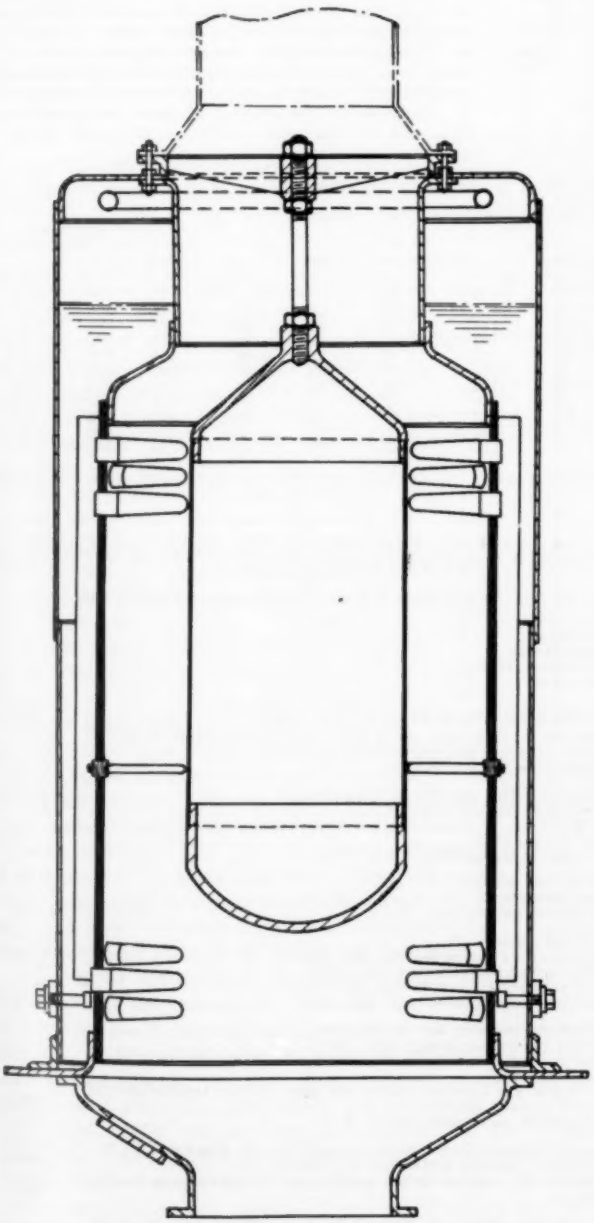


FIG. 3 "THROUGH" TYPE FOR MARINE INSTALLATIONS

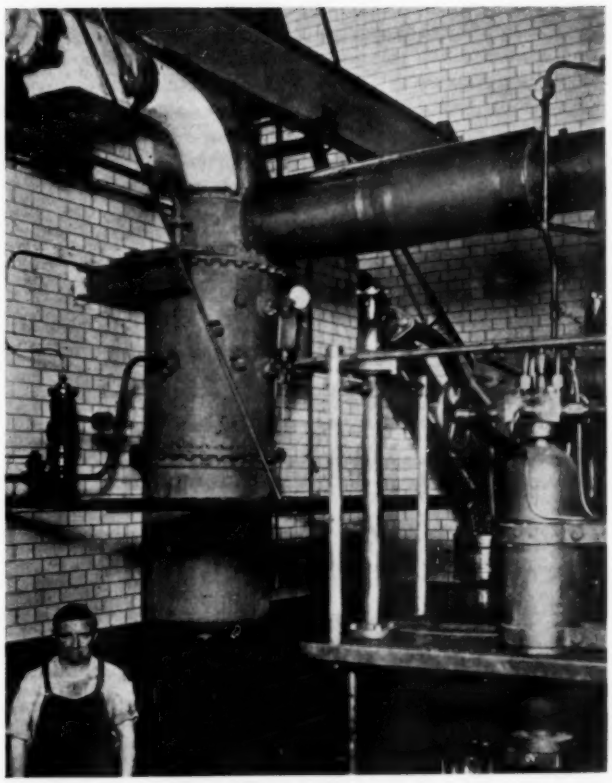


FIG. 4 EXHAUST-GAS UNIT INSTALLATION ON 300-HP., FOUR-CYCLE ENGINE

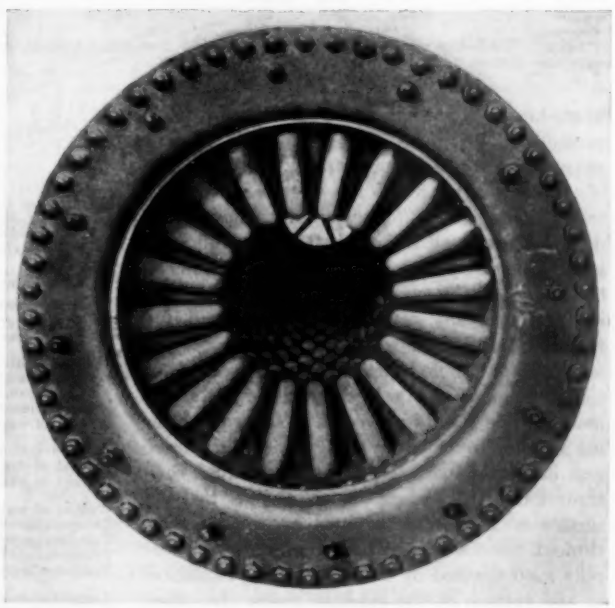


FIG. 5 LOOKING UPWARD THROUGH "THIMBLE TUBES"

TABLE 1 TEST OF CLARKSON HEATERS

(Fitted to Gardner 72-b.hp., two-cycle engine)

Time, p.m.	Temperature, deg. fahr. Gas In	Gas Out	Water In	Water Out	Water-meter readings
2:45	582	264	64	188	1640
2:55	580	270	65	197	..
3:05	583	280	65	208	..
3:15	586	275	64.5	204	..
3:25	582	270	64	162	..
3:35	587	270	64	155	..
3:45	592	270	64.5	167	..
3:55	595	280	64.5	186	1775
4:05	612	290	64.5	206	..
4:13	Thermometers checked			208	..
4:25	595	285	64.5	201	1827
4:35	593	283	64.5	184	1841
4:45	598	280	64.5	192	1848
4:55	597	275	65	186	1881

Average temperature of water entering and leaving heater, ° F.	out	188.85
	in	64.57
Average rise in temperature of water, ° F.		124.28
Average temperature of gases entering and leaving heater, ° F.	in	590.9
	out	276.3
Average fall in temperature of gases, ° F.		314.6
Water-meter readings at 4:55 p.m., gal. (imperial)		1881
Water-meter readings at 2:45 p.m., gal. (imperial)		1640
Water passed in 2 1/16 hours, gal. (imperial)		241
Water passed per hour, gal. (imperial)		111
Average B.t.u. recovered per hour (124.28 × 1110)		137,950.8

A question of some importance in connection with the use of waste-heat boilers is their effect upon the efficiency of the engine. There is some divergence of opinion on this point. On the one hand, the view is held that the introduction of the boiler in the exhaust pipe must increase the back pressure and adversely affect the efficiency; on the other hand, cases have been recorded where a definite improvement in the efficiency has resulted from the introduction of the boiler. To investigate this point two tests were carried out—first, with the boiler in use, and then with the boiler removed and a plain exhaust pipe inserted in its place. The two tests were carried out on the same day, under identical conditions of load, speed, etc. It was found that there was no measurable difference in the consumption of gas, but that the air consumption was greater by some 1 1/2 per cent when the boiler was in use than when a plain exhaust pipe was substituted. This showed at once that if any back pressure was caused it did not continue till the end of the exhaust stroke—the really vital point. On the contrary, it indicated the existence of a negative pressure over that portion of the cycle where the opening of the admission valve overlaps that of the exhaust. Light spring diagrams show this effect in a very striking manner. The fluctuations in the pressure during the exhaust stroke are the result of vibrations in the column of gases in the exhaust pipe, and the frequency of these vibrations is such

TABLE 2 RESULTS OF TESTS WITH CLARKSON WASTE-HEAT BOILER

(Working in conjunction with a Beardmore-Tosi four-cycle auxiliary Diesel engine)

Period of trial	Load	R.p.m.	Amperes	Volts	Kilowatts	Fuel consumption Lb. per hour	Fuel consumption Lb. per kw-hr.	Water evaporated Lb. per hour	Water evaporated Lb. per kw-hr.	Steam, lb. per sq. in.	Feed, deg. fahr.	Exhaust temperatures, deg. fahr. From engine No. 1 cyl.	From engine No. 2 cyl.	From boiler
2 hours	Full	244	347	225	78.1	55.6	0.71	157	2.01	25	40	643	658	390
4 hours	Full	247	369	226	83.4	58.8	0.71	143	1.72	25	40	689	684	411
4 hours	Full	243	366	223	81.6	56.6	0.69	122	1.50	25	41	675	630	396
First 2 hours	Full	245	370	220	81.4	55.4	0.68	120	1.47	25	39	668	630	402
Second 2 hours	Full	246	366	223	81.6	55.7	0.68	115	1.41	25	39	660	620	395
Third 2 hours	Full	246	365	225	82.1	55.8	0.68	125	1.52	25	39	680	635	401
Mean over 6 hours.	Full	246	367	223	81.7	55.6	0.68	120	1.47	25	39	669	628	399
1 hour	1/2	244	188	220	41.4	32.8	0.79	32	0.77	25	40	425	375	282
1 hour	3/4	244	279	220	61.4	44.3	0.72	64.6	1.05	25	40	565	500	328
1 hour	Full	243	369	220	81.2	58.9	0.72	120	1.48	25	40	750	683	391

NOTE: The boiler was connected to the engine by two exhaust pipes, each about 7 ft. long. In each pipe there were two right-angle bends. The pipes were not lagged.

waste-heat boiler there is no detrimental back pressure. This point was established several years ago in the following manner: A new engine of two-cycle type, 84–90 hp. was placed on the test bed and was run for several days against the dynamometer, with the usual muffler and exhaust-pipe connection, but without a waste-heat boiler. After the engine had been properly "run in" and constant readings were obtained for fuel consumption in relation to horsepower developed, a "thimble tube" heater was fitted close to the engine, and a fresh series of tests was made. The results proved that, far from causing any loss in efficiency, the same horsepower was developed with 3 per cent less fuel consumption. This result is explained by the sudden cooling of the exhaust gases in passing through the waste-heat boiler, which was naturally accompanied by a corresponding reduction in the volume of the exhaust gases. In other words, the boiler acted to some extent as a condenser, and actually facilitated the escape of the exhaust gases.

Careful investigations on this point have also been made by Prof. Gilbert Cook at the London University Engineering Laboratories, where several "thimble tube" boilers are installed. The results are best given in Dr. Cook's own words:

TABLE 3 TEST OF CLARKSON EXHAUST HEATER, TYPE S. I., FITTED TO CROSSLEY FOUR-CYCLE GAS ENGINE, AND HEATING WATER SUPPLIED TO IT FROM ENGINE JACKET

(Conducted by Prof. Gilbert Cook at University of London, King's College)

Date of test, April 24, 1926
Duration of test, one hour
Barometer, 30.19 in. mercury
Brake horsepower of engine, 17.61
Speed of engine, 265 r.p.m.

Calorific value of supply gas per cu. ft. at n.t.p., B.t.u.	Gross	497
	Net	452
Vapor of combustion, per cu. ft. supply gas at n.t.p., from gas calorimeter, lb.		0.045
Volume of gas supplied to engine per min. at n.t.p., cu. ft.		5.77
Weight of gas supplied to engine per min., lb.		0.22
Volume of air supplied to engine per min. at n.t.p., cu. ft.		36.4
Weight of air supplied to engine per min., lb.		2.81
Temperature of air, deg. fahr.		57
Relative humidity of air, per cent		66
Weight of vapor per min. in air supplied, lb.		0.0183
Total weight of exhaust gas per min., lb.		3.03
Weight of dry exhaust gas per min., lb.		2.752
Weight of vapor in exhaust gas per min., lb.		0.278
Weight of water supplied per min., lb.		11.37
Temperature of water entering engine jackets, deg. fahr.		55.2
Temperature of water leaving engine jackets, deg. fahr.		126.5
Temperature of water entering exhaust heater, deg. fahr.		125.5
Temperature of water leaving exhaust heater, deg. fahr.		188.0
Temperature of exhaust gas entering heater, deg. fahr.		1058.0
Temperature of exhaust gas leaving heater, deg. fahr.		218.5
Total heat in exhaust gas entering exhaust heater per min., temperature of air as base, B.t.u.		1164
Available heat in exhaust gas entering heater per min., taking temperature of water entering heater as base, B.t.u.		822
Heat given to water per min. in exhaust heater, B.t.u.		707
Area of heating surface of exhaust heater (figures supplied by Mr. Clarkson), sq. ft.		26.1
Heat transmitted per sq. ft. heating surface per hr., B.t.u.		1625
Heat transmitted per sq. ft. heating surface per hr. per lb. exhaust gas passing through heater, B.t.u.		536
Efficiency of exhaust heater, based on available heat in exhaust gas, as above, per cent		86

that the pressure happens to be positive at the end of the stroke, at the time when both valves are open, thus retarding the admission of the new charge. The effect of the boiler is to increase the capacity of the exhaust system, resulting in a lower frequency of the vibrations, so that in this particular case the pressure is negative at the end of the exhaust stroke. This condition is specially favorable for good scavenging. In a gas engine, where the fuel is already mixed with the incoming air, such a condition might not necessarily result in better efficiency, as the improved conditions of combustion might be offset by the passage of some of the fuel direct to the exhaust. That, in the opinion of the writer, is what has happened in the present case—an increase in air consumption without any appreciable effect on the efficiency. It is readily conceivable, however, that in an oil engine with fuel injection a condition such as this would entail a distinctly improved efficiency, and in those cases where such an improvement has actually been observed it is probable that the cause will be found to be similar. The writer is not prepared to say that such improvement must necessarily follow the installation of the boiler. The same effect might very well be produced either by a lengthening or a shortening of the exhaust pipe; there can be no doubt that small changes in the exhaust system have a much greater effect upon the performance of the engine than is generally recognized. It can, however, be definitely stated that there is no evidence of any sustained back pressure being caused by the boiler.

From the point of view of heat recovery it is obviously desirable not to water-cool the exhaust pipes or manifolds, just as skimmed milk is not the best for making butter. There has been a great lack of standardization among different makers of engines in the diameters of exhaust pipes in relation to the horsepower of engines. As examples of this may be cited the use of a 6-in. bore exhaust pipe for an engine of 60 hp., whereas another engine maker will use the same size of exhaust for 600 hp. The size of exhaust is greater in engines of the two-cycle type, and is naturally governed by the number of cylinders as well as the horsepower of engine. As a general rule the size of exhaust should be kept as small as possible, consistent with effective scavenging. This is facilitated by the use of multi-cylinder engines which have a more continuous exhaust, and which therefore can have a greater horsepower per square inch of exhaust-pipe area. The explanation why it is possible to discharge hot gases through an empty "thimble tube" boiler without injuring it is found in the fact that the gas temperature never reaches a point sufficiently high to burn the tubes and that there is complete freedom for expansion. The maximum temperature usually met with is in the neighborhood of 700 deg. fahr., although there are records of tests giving exhaust-gas temperatures of over 1000 deg. The usual temperature of gases from engines of the four-stroke type is from 600 to 650 deg. fahr. This of course depends upon whether the exhaust manifold of the engine is "dry" or "water-cooled." In engines of the two-stroke type the exhaust gases rarely exceed 500 deg. fahr.

There is, of course, a practical limit beyond which it is not desirable that the gases should be cooled in a waste heat boiler, in order to avoid condensation, particularly of the sulphurous contents, which might cause corrosion of the heating surface. As a general rule 300 deg. fahr. may be taken as the practical limit beyond which the gases should not be cooled.

The quantity of thermal units recoverable from the exhaust gases of an internal-combustion engine may be readily estimated by the simple formula, B.hp. $\times$ C $\times$ $D/4$, C being a constant (in the case of four-cycle engines, 12, and in the case of two-cycle engines, 20) and D being the drop in temperatures of the gases.

More heat can of course be recovered in the form of hot water than in the form of steam, as it is obvious that when generating steam the temperature of the gases leaving the boiler must be considerably above boiler temperature. The difference is usually not less than 50 deg., so that in the case of a boiler generating steam at 100 lb. per sq. in. the final temperature of exhaust gases could not be much less than 400 deg. fahr. If, on the other hand, hot water is to be produced, say up to a temperature of 200 deg. fahr., the gases could easily be cooled another

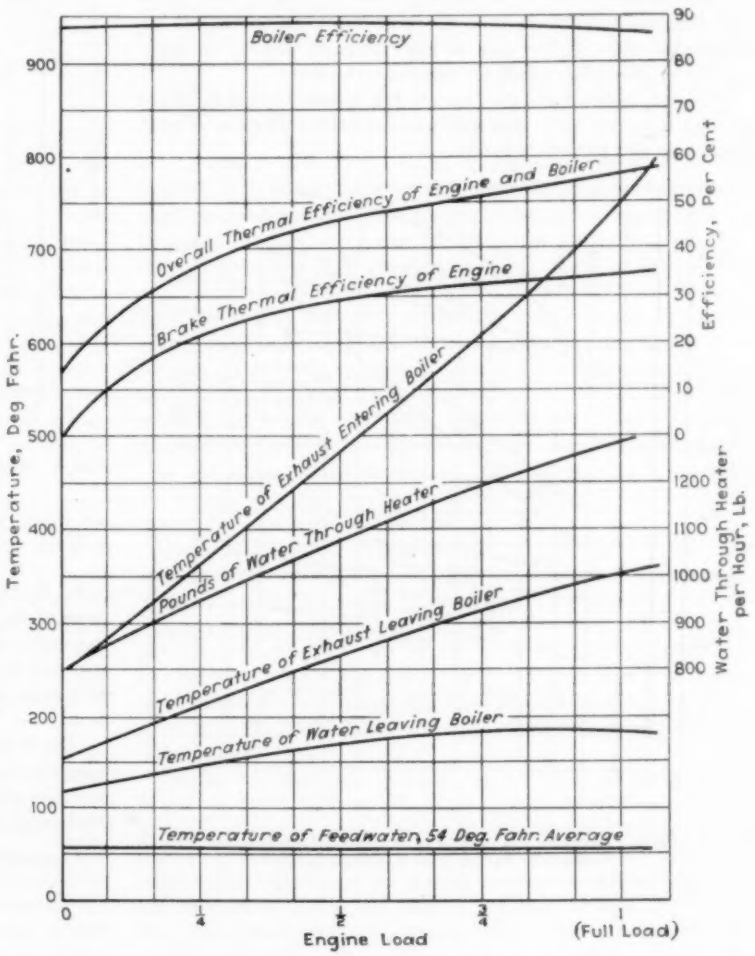


FIG. 6 CHART OF AVERAGE EFFICIENCIES AND TEMPERATURES OF FOUR-CYCLE MECHANICAL-INJECTION DIESEL ENGINE AND NELSECO CLARKSON WASTE-HEAT BOILER

100 deg., which, in the case of a two-stroke engine, may represent an increase of 100 per cent in the quantity of heat recovered.

When considering the production of hot water, advantage can usually be taken of the contra-flow principle; that is, bringing the exhaust gases in at the top of the heater where they come in contact with the hottest water, and causing them to travel in a downward direction over the heating surface, so that they ultimately come into contact with the coolest water.

The general construction and arrangement of a "thimble tube" heater are illustrated in Fig. 1, which is a heater of the erect type, the exhaust gases entering and leaving at the base of the heater. In some installations it is more convenient to invert the heater and have the inlet and outlet connections at the top. This type is shown in Fig. 2.

In large marine installations the "through" type is extensively

used. In this type the exhaust gases enter at the base and leave at the top, thus conforming to the general direction of flow of the exhaust gases upward from the engine in the hold of the vessel, to the top deck, where the funnel is located. An illustration of the "through" type is shown in Fig. 3, which is a vertical section of one of the four main waste-heat boilers dealing with the exhaust gases from 20,000-hp. Diesel engines of the motor-vessel *Britannic*.

The largest internal-combustion engines are to be found in marine installations. It is therefore not surprising that there has hitherto been more heat recovery practiced in marine installations than in the case of land services, which, however, appear to offer a large field for useful development.

Tables 1, 2, and 3 and the chart in Fig. 6 show typical examples of heat-recovery tests with internal-combustion engines of both four-cycle and two-cycle types.

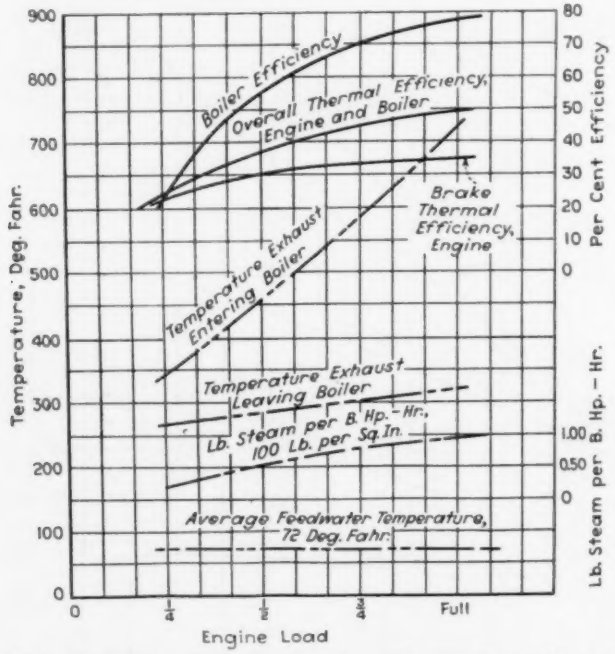


FIG. 7 SHOP TESTS OF EXHAUST-GAS STEAM BOILER, ENGINE UNDER CONSTANT SPEED AND VARYING LOAD

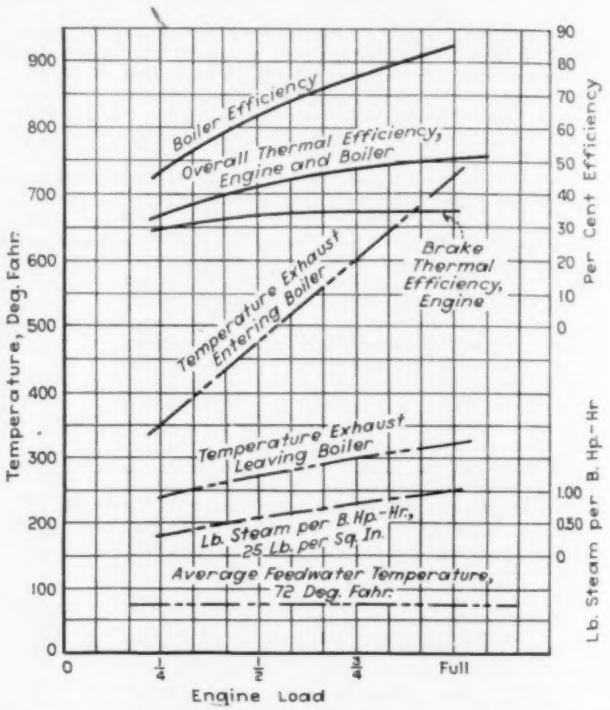


FIG. 8 SHOP TESTS OF EXHAUST-GAS STEAM BOILER, ENGINE UNDER VARYING SPEED AND VARYING LOAD

Discussion

L. B. DOANE.[3] As a supplement to the chart on the fifth page of the paper (Fig. 6), which gives results obtained by heating water with the exhaust gases from a Diesel engine, the writer presents two additional charts that give data obtained by using this same heater as a steam generator under exhaust-gas operation.

One of these charts, Fig. 7, shows results obtained when running the engine under constant-speed and varying-load conditions and the other, Fig. 8, shows results when operating the engine with both varying speed and load.

[3] Assistant Chief Engineer, Electric Boat Company, Groton, Conn.

OGP-53-5

Dispersion of Sprays in Solid-Injection Oil Engines

By KALMAN J. De JUHASZ,[1] STATE COLLEGE, PA.

In investigating the problem of how to improve atomization and dispersion in the solid-injection oil engine, the knowledge of the mechanism of spray formation was found to be scant. In the researches covered by this paper, it was not attempted to develop a spray theory. A few experiments were made for the purpose of clearing up which physical forces do or do not participate in the atomization and distribution process. A large number of charts have been made representing sprays under various conditions, these seeming to point to the following general conclusions: Dispersion becomes more even when injection pressure is increased, when the oil viscosity is decreased, and when the air density is increased; the cone angle increases with increasing oil pressure, increasing air density, and decreasing viscosity of oil; a larger percentage of oil reaches a given distance in a spray having a slenderer spray cone; no definite influence of orifice dimensions was noticeable in using cylindrical orifices of the dimensions employed in the tests discussed.

IN THE dark ages of oppression Machiavelli propounded the sinister doctrine of "Divide and rule" as the most effective policy for obtaining and holding control over a body of people. In this more enlightened and civilized era, in the peaceful art of internal-combustion engineering the same principle can be usefully applied to the fuel charge, and thereby an effective control over the combustion be obtained. Trying to put this precept into practice constitutes one of the main problems of the solid-injection oil engine today.

In an oil engine the divisions must be twofold: First, the injected fuel must be divided into sufficiently small particles. Thereby, a large surface is made available upon which the chemical reaction with the oxygen of the combustion air can take place. This process, called atomization, is the essential prerequisite of complete and rapid burning. Second, these particles must be mixed with, and be distributed evenly in, the combustion air in order that the combustion take place without waste of fuel and at high loads, also without waste of air. This process, called dispersion, is a fundamental requirement for low fuel consumption and high mean effective pressure.

These latter-named performance characteristics are highly desirable for all, and a necessity for many applications of an internal-combustion engine.

As to the first of these, low fuel consumption, the solid-injection oil engine is superior to its rivals, the carburetor engine and the air-injection Diesel engine. With respect to the second characteristic, high mean effective pressure, however, it lags sadly behind them. There is no basic reason for this disadvantage. Owing to the high thermal efficiency of its cycle and the absence of power-consuming accessories, the solid-injection oil engine possesses an inherent potentiality for high mean effective pressure. The reason for this deficiency is the lack of satisfactory combustion control, which offsets the advantage of the superior engine cycle by necessitating the use of a large percentage of excess air. It may be said that the future progress of the solid-injection engine and its entrance into new fields of application depend to a large degree on the progress in eliminating this deficiency by better combustion control, with which the processes of atomization and dispersion are closely connected.

Both these processes absorb energy. In the atomization process, by increasing the total surface of the liquid, work has to be expended to overcome surface tension. In the dispersion process, over and above the work necessary to bring a volume of oil into a space under pressure, work has to be expended against air resistance when the particles penetrate into the dense combustion air. The turbulence necessary for mixing involves overcoming the friction of the air with the chamber walls and the viscosity forces of the air motion. It is of interest to compare the energy expended for these purposes in the air-injection process and in the solid-injection process.

In the air-injection process this energy is equivalent to 6 to 14 m-kg. per gram of oil, depending on the pressure of the injecting air (900 to 1000 lb. per sq. in.) and on the weight ratio of injection air and oil (usually 1 to 2). Incidentally, this energy is equal to 0.5 to 1.2 per cent of the work derived from the oil in the engine. This comparatively large amount of energy brings about a violent turbulence and intimate mixing of the fuel particles with the air, and consequently a high degree of utilization of both. (See Appendix 1.)

In the solid-injection process the energy is contained solely in the oil due to its pressure. Storing in a gram of oil the same amount of energy as is used in the air-injection process would involve pressures of 8000 to 18,000 lb. per sq. in. There are serious objections to the use of such pressures in engineering practice on account of the high stresses and ensuing wear and tear in the pump mechanism and nozzle parts. Even if applied, the result would not be the same as with air injection owing to the different nature of the atomization process. In actually built engines much lower oil pressures (3000 to 5000 lb. per sq. in.) are used, corresponding to 2 to 4 m-kg. energy per gram of oil. This energy is equivalent to 0.2 to 0.3 per cent of the useful work derived from the same quantity of oil. There is even a tendency to use still lower injection pressures, which would be of advantage for the ease of manufacture of the pump and nozzle parts and for their useful life.

In theory, at least, there is no basic obstacle in the way of obtaining a high degree of atomization with only little expenditure of energy. The atomization energy is equivalent to the product of surface tension and total surface. The surface tension is of the order of magnitude of 0.003 kg. per m. for oils which come into consideration here. The total surface is the sum of the surfaces of the individual droplets. Imagine a quantity of oil as a compact sphere of R_1 radius and A_1 surface area. When this sphere is broken up into droplets of r_n radius, the total surface is enlarged in inverse ratio to that of the respective radii:

[1] Assistant Professor of Engineering Research, Pennsylvania State College.

Presented at the National Oil and Gas Power Meeting, Madison, Wis., June 23 to 26, 1931, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

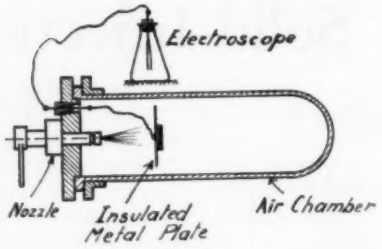


FIG. 1

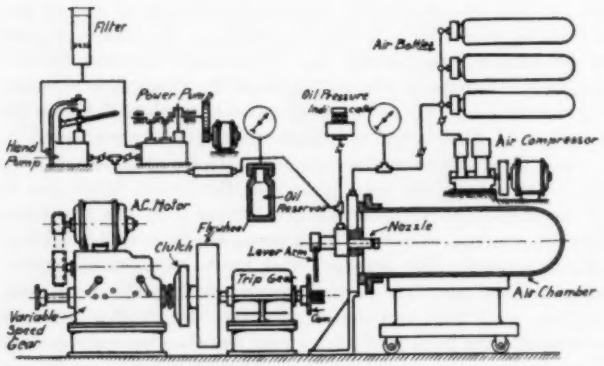


FIG. 2 SCHEMATIC ARRANGEMENT OF OIL-SPRAY APPARATUS

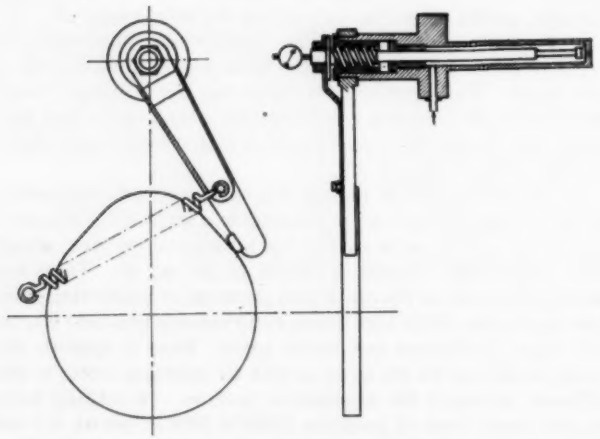


FIG. 3 SPRAY-VALVE MECHANISM

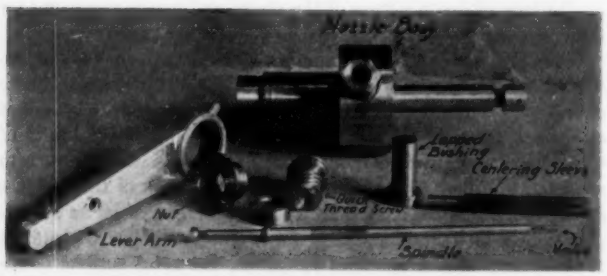


FIG. 4 COMPONENT PARTS OF THE INJECTION NOZZLE

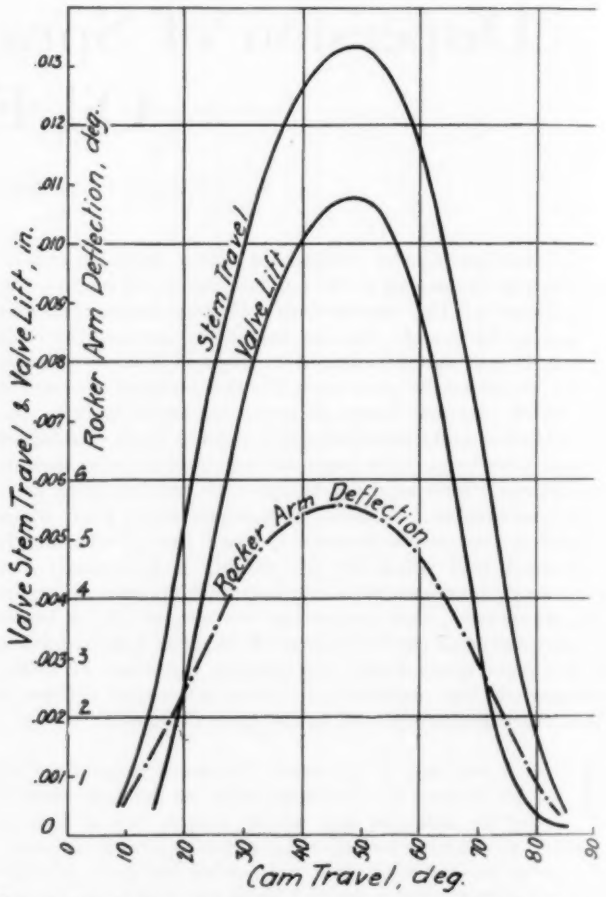


FIG. 5 CHARACTERISTIC CURVES OF CONSTANT-PRESSURE INJECTION NOZZLES

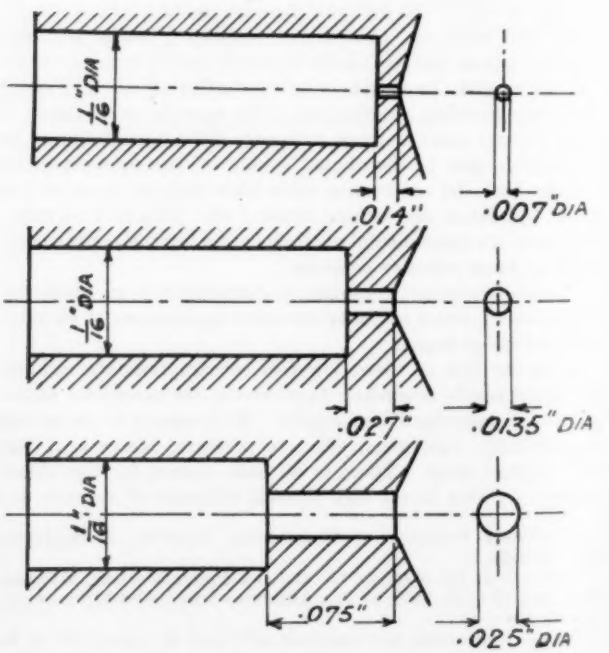


FIG. 6 ORIFICES USED IN CONSTANT-PRESSURE INJECTION EXPERIMENTS

$$\frac{A_n}{A_1} = \frac{R_1}{r_n}$$

The enlarged area A_n corresponds to the surface of one sphere, which may be called "equivalent bubble," having a radius R_n which can be expressed:

$$R_n = R_1 \sqrt{\frac{A_n}{A_1}} = R_1 \sqrt{\frac{R_1}{r_n}}$$

Putting it in another way, the atomization results in a surface increase equivalent to coating a large sphere of R_n radius with the original quantity of oil in a thickness of $r_n/3$. (See Appendix 2.)

On Fig. 17, in Appendix 2, these relationships are graphically

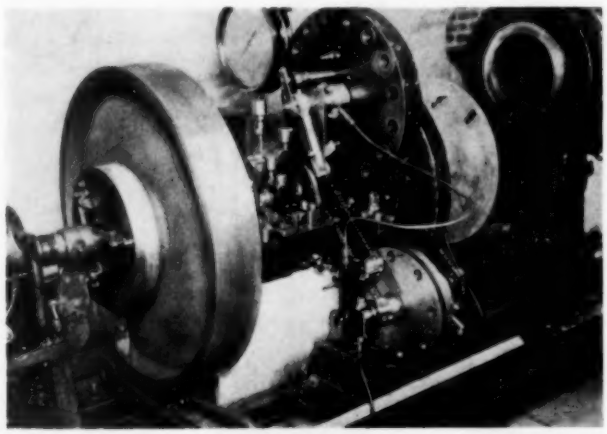


FIG. 7 ARRANGEMENT OF PUMP INJECTION SYSTEM

shown for droplet sizes comparable to those in solid-injection practice, and the surface-energy values corresponding to 1 gram of oil are given in the table therewith. Such simple calculations show that to atomize 1 gram of oil into droplets of 0.0002 cm. radius, which is an atomization comparable to that obtainable with air injection, only 0.005 m-kg. work is necessary. It is quite conceivable, however, that in some intermediary stage of the drop formation the liquid has a greater energy content than its final stage, similar to an elastic cord which is stretched until broken. Nevertheless, it seems safe to assume that the energy required for atomization is only a minute fraction of the total energy of injection, the greater part of which is expanded in the process of dispersion and turbulence and finally is converted into heat.

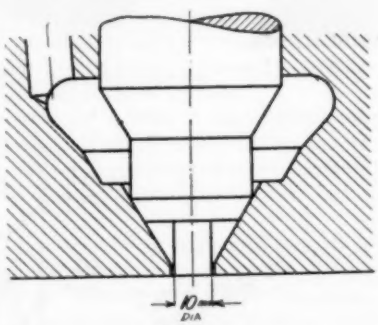


FIG. 8

In investigating the problem of how to improve the atomization and dispersion, a knowedge of the mechanism of spray formation would be of considerable help. It should be stated, however, that knowledge in this field is very scant, though there are a few theories that may provide a background for the understanding of spray phenomena. The breaking up of a liquid column issuing from an orifice under a pressure head has engaged the attention

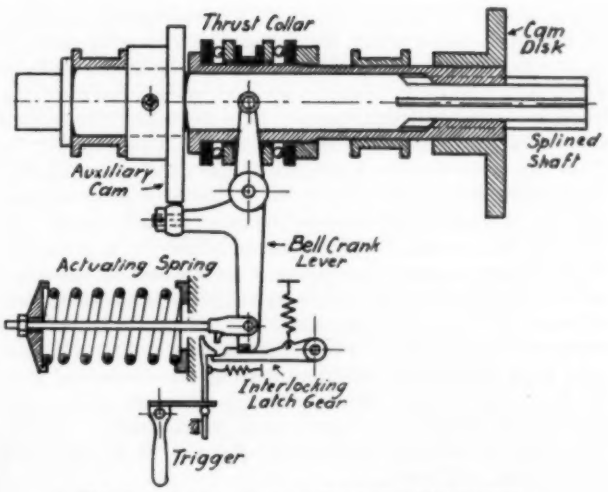


FIG. 9 ARRANGEMENT OF SELECTOR MECHANISM

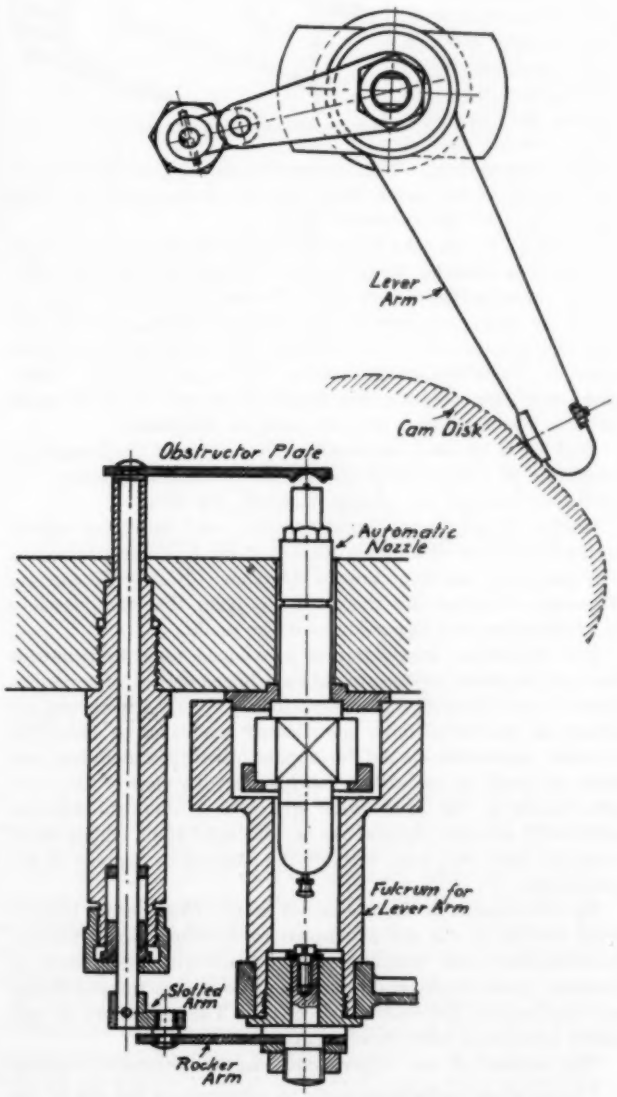


FIG. 10 METHOD OF ACTUATING OBSTRUCTOR PLATE

of numerous investigators. Plateau (1)[2] investigated the physical conditions under which the collapse of a round cylindrical liquid column takes place. He found that the column is unstable if its length exceeds its circumference and will collapse under the influence of any small disturbance. Lord Rayleigh (1) calculated the rate of collapse. More recently R. A. Castleman, Jr., (1) of the Bureau of Standards, amplified these theories and applied them specifically to the atomization of liquids in an air stream, under conditions similar to those occurring in carburetors. He describes the process of atomization as follows: "A portion of the large mass is caught up (say at a point where its surface is ruffed) by the air stream, and being anchored at the other end, is drawn out into a fine ligament. This ligament is quickly cut off by the rapid growth of a dent in its surface, and the detached mass, being quite small, is swiftly drawn up into a spherical drop. The higher the air speed, the finer the ligaments, the shorter their lives, and the smaller the drops formed, within limits." Castleman refers to the solid injection as follows: "It is found that when liquid is forced under very high pressure into still air it is finely atomized. As the high injection pressure merely gives a high initial velocity to the injected liquid, it is seen that this case also is similar to air-stream atomization. The fast-moving liquid loses ligaments to the still air in the same way that the fast-moving air drags ligaments from the quiescent liquid."

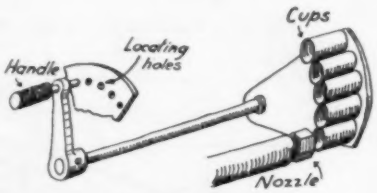


FIG. 11 DISCHARGE CUPS

Triebnigg (2) attacks the problem of atomization by solid injection from a similar point of view. In his very thorough theoretical investigation of this subject he assumes that the breaking up of the spray is caused by the combined action upon the free end of the liquid jet of two forces: air resistance and surface tension. Upon this assumption he builds up an elaborate theory from which he computes the droplet sizes and jet shape under various liquid-pressure and air-pressure conditions.

Bird (3) in his analysis considers the energy of the droplets as consisting of 1 (*a*) surface energy, (*b*) translational energy, (*c*) rotational energy, (*d*) circulation inside the drop.

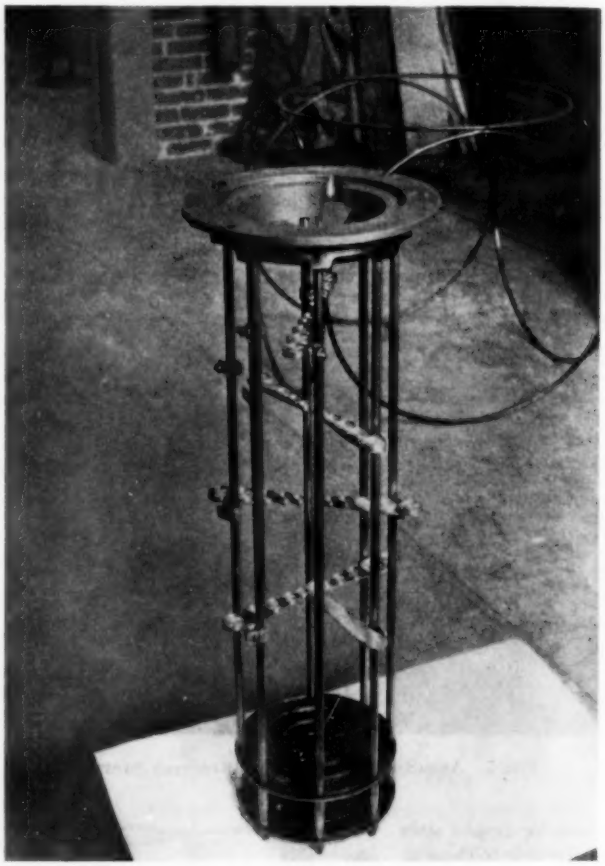
Kuehn (4) attributes the atomization and dispersion chiefly to the turbulence and vortex motion of the liquid in the nozzle. Thiemann (5) also leans toward this view. Neither Kuehn nor Thiemann develops this theory to the point that computations of atomization and dispersion could be made.

The theoretical investigations mentioned refer to primitive forms of jet under low pressure or have been evolved, of necessity, under drastic simplifying assumptions. For oil-engine use the sprays are produced under complicated conditions to which the theories mentioned cannot be applied directly and indeed are likely to result in misleading conclusions. In view of the complex nature of the oil-injection sprays, it is doubtful whether sufficiently accurate theories can be developed at all, which would stand on their own feet, without the continuous support of experiments.

For this reason in the researches at the Engineering Experiment Station it was not attempted to develop a spray theory. Nevertheless, a few experiments were made with the purpose of clearing up which physical forces do or do not participate in the atomization and distribution processes. The results are of sufficient interest to merit brief mention.

The purpose of one experiment was to determine whether atomization is due to the friction of jet with the air. Sprays were injected into an evacuated chamber, the air pressure being one-fiftieth atmospheric pressure (about 0.6 in. Hg. abs.), and thereby the air resistance was reduced to a very low value. In spite of this it was found that the spray was atomized and dispersed to approximately the same extent as in air of atmospheric

[2] Numbers in parantheses apply to references at the end of the paper.


FIG. 12 DISPERSION RACK

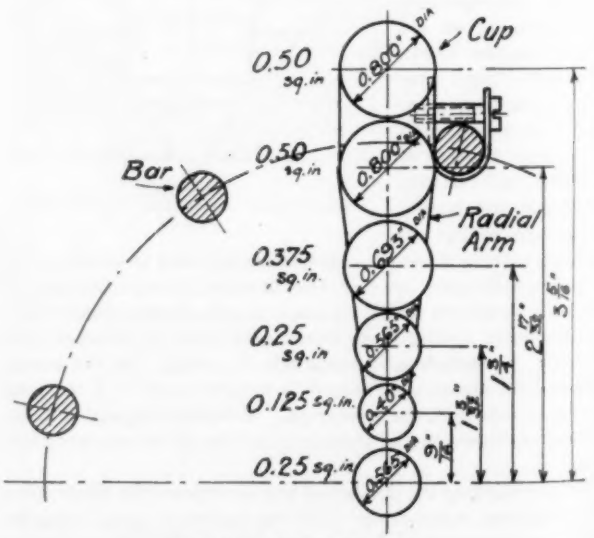


FIG. 13 DISPERSION RACK

density. Further increasing air density to multiples of the atmospheric density, however, had the effect of widening the spray angle, making the distribution pattern more even and reducing the penetration. This result seems to be strong evidence against the supposition that the atomization and dispersion are due solely to air-friction forces.

Another experiment was made to determine whether the oil particles, during their passage through the nozzle, become charged with static electricity owing to friction, which may contribute to their dispersion as a consequence of electric repulsion. The spray was injected against an insulated metal plate connected to an electroscope as shown on Fig. 1. The indication was entirely negative. On no occasion could the existence of static charge be demonstrated.

These evidences lend support to the view that (under the conditions of oil-engine injection) the liquid emerges from the nozzle in an already atomized, or at least disgregated state, that it already possesses the ability to disperse, and therefore that in the atomization and dispersion process the flow phenomena in the nozzle are the main influencing factors.

Experiments Relating to Spray Dispersion

Lacking a suitable theory for the explanation of spray phenomena, recourse has to be taken to experimental methods in order to determine those spray characteristics that are of interest from the engine-performance point of view. Even though the law of it is as yet undiscovered, yet spray formation is none the less a process obeying physical laws, and as a consequence the sprays assume the same properties under similar conditions, i.e., with the same values of the governing factors. The spray is formed by forcing the required quantity of fuel through a nozzle into the air contained in the combustion air. The characteristics of the spray, and among these its dispersion in space, depend therefore on three sets of factors, as follows:

(1) Fuel factors, comprising oil pressure, specific gravity, viscosity, and surface tension.

(2) Nozzle factors, comprising size and shape of the channels and apertures through which the fuel passes.

(3) Combustion-air factors, comprising its density (pressure temperature and its state of stagnation or agitation.

At the Engineering Experiment Station of the Pennsylvania State College a general research is being carried out on the dependence of spray properties on the various governing factors. One branch of this research deals with the distribution of fuel in space.

The equipment for the investigation of dispersion consists of the following groups: (1) spray-producing equipment, (2) air chamber, (3) cup and rack for catching the spray. The equipment is known probably to many from previous visits at Penn State. It has been described also in a number of publications. (6, 7, 8, 9) Therefore only a brief description will be given.

The layout of the spray-producing equipment and of the air chamber is shown in the schematic drawing Fig. 2. An electric motor drives a flywheel through a variable-speed gearing, the extension shaft of the flywheel actuating the spray-producing equipment which it is desired to investigate.

Two injection systems were investigated:

(1) Constant-pressure injection with mechanically operated valve.

(2) Pump injection with hydraulically operated valve (Robert Bosch type). In the constant-pressure injection system the oil can be put under pressure by means of a hand-operated or of a power-operated pump. The nozzle proper is shown schematically in Fig. 3, and the dismantled parts are shown in the photograph, Fig. 4. The nozzle contains a co-axial spindle, the lapped and polished flat tip of which serves for a valve. It is lifted off from or pressed on to its seat whenever the lever arm is actuated by the cam. The characteristics of this nozzle are shown in Fig. 5, and the orifices used in connection with it are shown in Fig. 6.

The pump-injection system was driven from the variable-speed shaft by means of a chain-and-sprocket drive, as shown in Fig. 7. The nozzle used is the so-called pintle type, having a cylindrical guide pin, as shown in Fig. 8.

In experiments on sprays it is necessary to separate one single injection, but without altering thereby the conditions prevailing when continuously repeated injections are made. There are two

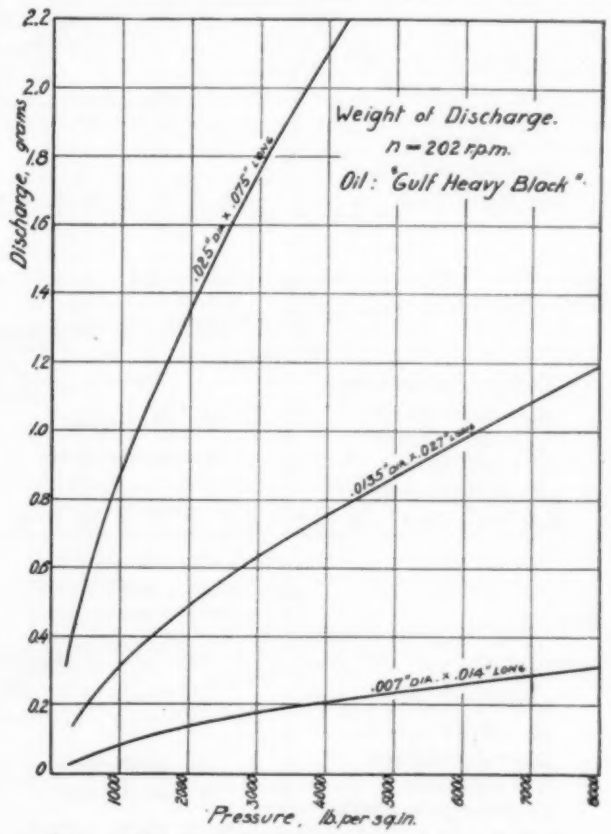


Fig. 14 Weight of Discharge With Various Orifices

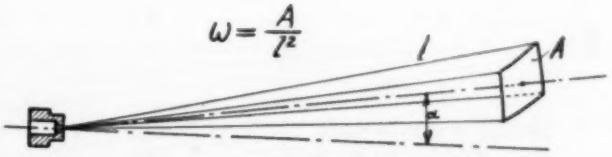


Fig. 15 Definition of Solid Angle

ways of accomplishing this: First, by actuating the flywheel, camshaft, and cam continuously at the specified speed, but preventing it from lifting the cam follower and the valve, except for a single injection. This method was used with the constant-pressure injection system. The second method is by actually producing repeated injections and shielding the measuring apparatus from all sprays except one. This method was used with the pump-injection system.

In both instances a so-called "selector mechanism" is used which moves the cam axially and thus brings it into engagement with its follower whenever a trigger is pulled. This mechanism is shown schematically in Fig. 9, which is largely self-explanatory. In the pump-injection system the shielding plate or "obstructor

plate" is actuated by this same cam mechanism, as shown in Fig. 10.

The nozzle is fixed, by means of a circular flange, to the air-chamber cover, which is a 2 in. thick steel plate of 20 in. diameter. It is bolted to a large cast-iron platform forming the foundation of the whole equipment.

The injection is made into the air chamber, which is a 36 in.

The apparatus used in dispersion experiments consists of two parts:

(1) The discharge cup for catching the total discharge of the spray.

(2) The dispersion rack for catching portions of the spray at definite locations in space.

The discharge cup is illustrated in Fig. 11. It is in reality a

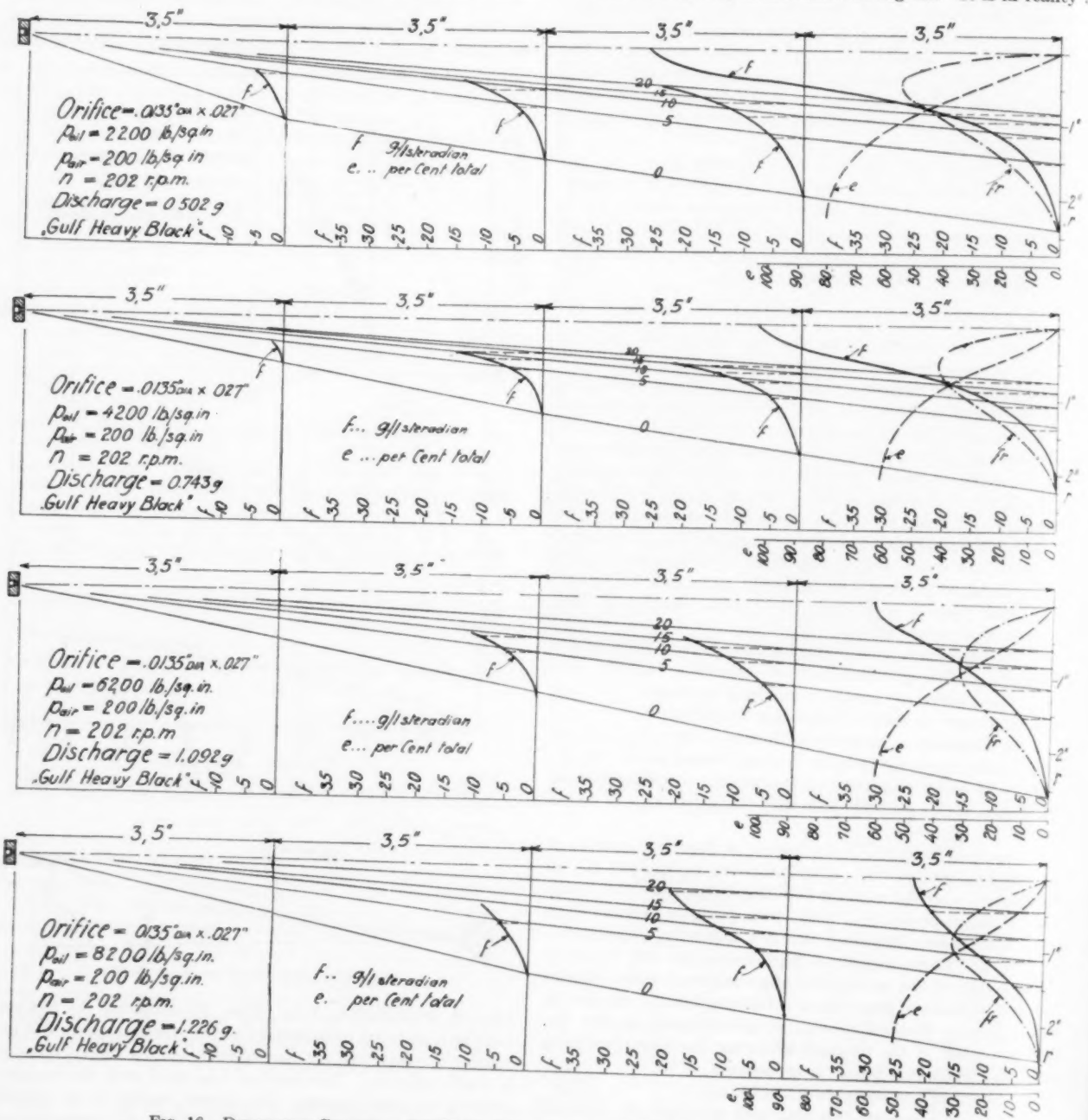


FIG. 16 DISPERSION CHARTS OF 0.0135 IN. DIAMETER BY 0.027 IN. ORIFICE IN COMPRESSED AIR

long piece of 10 in. diameter seamless-steel tube closed hemispherically on one end and arranged so as to be bolted hermetically to the chamber cover mentioned. The air chamber can be filled with air at any required pressure. The air is compressed by a motor-driven compressor and stored conveniently in a battery of steel bottles.

group of five cups, mounted on a sector. Each cup can be brought in front of the nozzle. The cups are filled with cotton waste or with some other absorbent material, and from their increment of weight after injection the weight of the injected fuel can be calculated.

The dispersion rack is a skeleton upon which absorbent pads

Orifice = .0135" dia. x .027" p_{oil} = 2200 lb/sq. in. p_{air} = 200 lb/sq. in. n = 202 r.p.m. Gulf Heavy Black.

Field		l in.	A sq. in.	$\omega = A/l^2$ steradian	Q grams	$z = Q/A$	$f_\omega = Q\frac{l^2}{A}$	$f = \frac{1}{Q_{total}} \cdot f_\omega$
a	1	14	.125	.00064	.0120	.0960	18,75	37,20
	2	10,5	"	.00114	.0135	.1080	11,85	23,50
	3	7	"	.00255	.0090	.0720	3,525	7,00
	4	3,5	"	.01022	.0010	.0080	.098	0,19
	5	14	"	.00064	.0115	.0920	17,96	35,60
	6	10,5	"	.00114	.0120	.0960	10,53	21,00
	7	7	"	.00255	.0130	.1040	5,10	10,15
	8	3,5	.125	.01022	.0020	.0160	1,955	3,90
b	1	14	.250	.00128	.0065	.0260	5,080	10,10
	2	10,5	"	.00227	.0030	.0120	1,323	2,64
	3	7	"	.00510	.0025	.0100	,490	0,98
	4	3,5	"	.02044	traces	0	0	0,0
	5	14	"	.00128	.0075	.0300	5,860	11,70
	6	10,5	"	.00227	.0035	.0140	1,541	3,08
	7	7	"	.00510	.0020	.0080	,393	0,78
	8	3,5	.250	.02044	traces	0	0	0
c	1	14	.375	.00192	.0020	.0053	1,041	2,07
	2	10,5	"	.00341	.0005	.0013	.147	0,30
	3	7	"	.00765	traces	0	0	0
	4	3,5	"	.03066	traces	0	0	0
	5	14	"	.00192	.0030	.0080	1,561	3,11
	6	10,5	"	.00341	.0010	.0026	,293	0,58
	7	7	"	.00765	traces	0	0	0
	8	3,5	.375	.03066	traces	0	0	0
d	1	14	.500	.00255	.0005	.0010	.196	0,39
	2	10,5	"	.00454	traces	0	0	0
	3	7	"	.01020	traces	0	0	0
	4	3,5	"	.04080	traces	0	0	0
	5	14	"	.00255	.0005	.0010	.196	0,39
	6	10,5	"	.00454	traces	0	0	0
	7	7	"	.01020	traces	0	0	0
	8	3,5	.500	.04080	traces	0	0	0
e	1	14	.500	.00255	traces	0	0	0
	2	10,5	"	.00454	"	0	0	0
	3	7	"	.01020	"	0	0	0
	4	3,5	"	.04080	0	0	0	0
	5	14	"	.00255	traces	0	0	0
	6	10,5	"	.00454	"	0	0	0
	7	7	"	.01020	"	0	0	0
	8	3,5	.500	.04080	0	0	0	0
o		14	.250	.00128	.0455	.182	35,65	71,0

Q_{total} = 0.502 g.

FIG. 17 DATA SHEET ON TEST

can be fastened for intercepting the spray at a number of predetermined locations in space (Fig. 12). The dimensions of the pads are given in Fig. 13. In the experiments here to be described 41 pads were used on the rack. In carrying out the experiments, first of all discharge tests were taken. A chart of the discharge weights with various oil pressures and orifices is shown in Fig. 14. Then the dispersion rack is put in place and carefully lined up co-axially with the spray. The chamber is closed and filled with air at the required pressure, and then one or two injections are made.

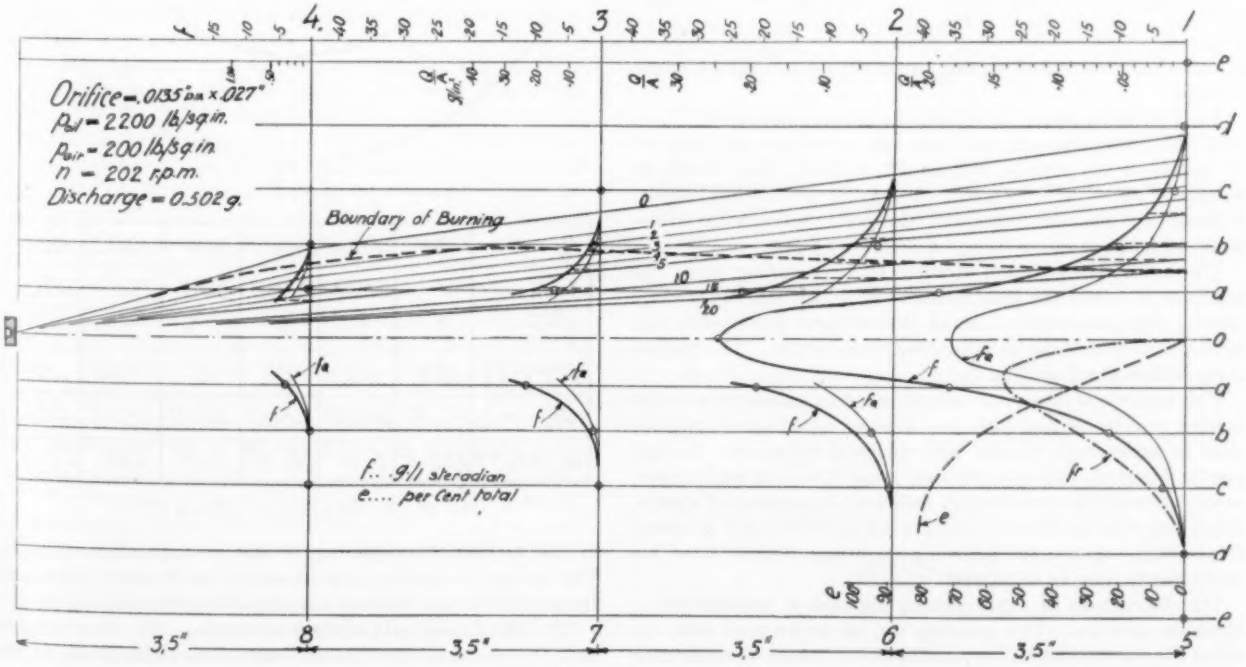
After this the chamber is opened up, and the weight increment of the pads is determined. In view of the large number of weighings a sensitive spring balance (Paulin, Roller-Smith type) was used which greatly reduced the weighing time and maintained satisfactory accuracy. Much attention was given to developing a method of representation for the spray pattern. The weight increment of the pad is equivalent to the quantity of oil passing, during the duration of the spray, through that area in space occupied by the pad.

Dividing the amount of oil by the area of the pad, the value "spray per unit area" $z = Q/A$ is obtained. It is preferable, however, to refer the spray weight instead of the area, which is a dimensioned quantity, to the enclosed solid angle, which is a non-dimensioned quantity. It is to be recalled that the solid angle is the ratio of a portion of a spherical surface to the square of the radius $\omega = A/l^2$, which concept is illustrated in Fig. 15. The unit of solid angle is "one steradian," this being $1/4\pi$ part of the complete sphere. Using this as reference, the "spray weight per unit solid angle" $f_\omega = Q(l^2/A)$ is obtained, which shall be called by the name flux, for the sake of brevity.

In an ideal spray which has no loss and no resistance, and with its particles traveling in a straight-line trajectory, the "spray weight per unit area" value decreases with the square of the distance, while the "spray weight per unit solid angle" is constant along a given trajectory.

All these values refer to a given total weight of spray, which latter is of no particular interest, since evidently a given quan-

FIG. 18 (BELOW) CHART OF A DISPERSION TEST



tity of oil can be dispersed by both a large orifice in a short time (or in a lesser number of injections) or by a small orifice in a longer time (or in a larger number of injections). In order to arrive at a representation of spray which would be a measure of the dispersing ability of a given orifice, oil pressure, and air pressure combination, it is of interest to refer the discharge to unit weight, i.e., to divide all the f_Q values by the total weight of discharge. This quantity is called "specific flux" $f = \frac{1}{Q_{total}} \frac{Ql^2}{A}$ and it can be defined as the dispersion of one gram of oil in a given direction.

The table shown in Fig. 17 gives the measured and computed data from a dispersion test with varying oil pressures, and Fig. 18 shows the chart corresponding to this spray.

In the experiments with round orifices it can be assumed that the dispersion is the same in all meridional planes. On this assumption the total amount of spray reaching a given cross-section can be determined from the dispersion chart. The amount is always less than the total discharge because, owing to air resistance, parts on the outer fringe of the spray lose their velocity



FIG. 19

at certain distances and do not advance to more distant cross-sections.

The flux being equal, $f_Q = l^2(Q/A)$, or more accurately, $f_Q = l^2(dQ/dA)$, and assuming that the flux values are the same at all points of the annular surface $dA = 2\pi r dr$, then it can be written $\int dQ = (2\pi/l^2) \int f_Q r dr$. Using the specific flux values, Q becomes e, i.e., the fraction of one gram reaching the cross-sectional surfaces in question: $e = (2\pi/l^2) \int f r dr$.

The actual computation consists in constructing, first, the auxiliary fr curve, the integral curve of which represents the e curve. The maximum value of this integral curve gives the total quantity reaching the respective cross-section. The method is applied on the chart Fig. 18.

This analysis of dispersion is applicable to, and may be a useful helping tool in dealing with, any kind of liquid spray, such as used in agriculture, textile, and chemical industries. In Appendix 3 is given a list of industries using sprays for various purposes, showing the considerable technical importance of sprays. Regarding now, specifically, a fuel spray to be burned in internal-combustion engines, the following quantities derived from the spray charts may be of interest:

(1) Percentage of spray passing through a precombustion-chamber aperture. This quantity can be determined, with the aid of the f or the e curve, by drawing the ordinates corresponding

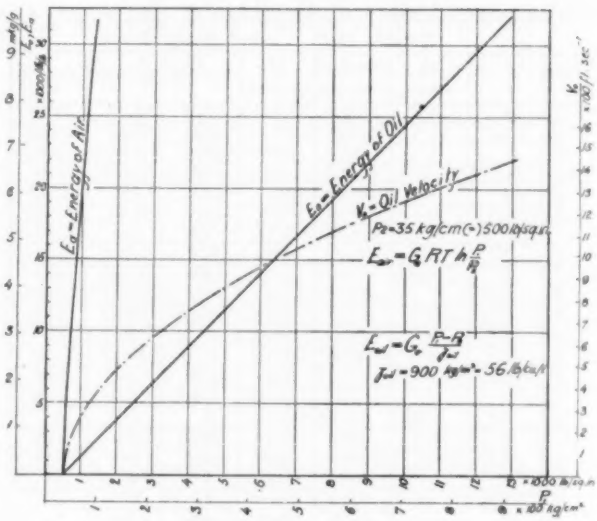


FIG. 20 ENERGY OF INJECTION

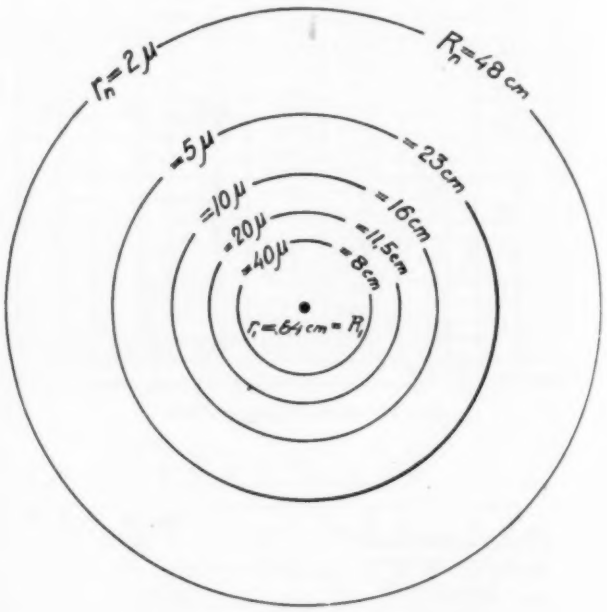


FIG. 21

r_n cm	.64	.0002	.0005	.001	.002	.004
r_1/r_n	1	3200	1280	640	320	160
$\sqrt{r_1/r_n}$	1	56	36	25	18	13
$R_n = R_1\sqrt{\frac{r_1}{r_n}}$	.64	48	23	16	11,5	8
A_n cm²	5,15	16400	6600	3300	1650	825
$W_n = \sigma A_n$ mkg	1,56.10⁻⁶	5.10⁻³	2.10⁻³	1.10⁻³	5.10⁻⁴	25.10⁻⁵

FIG. 22 ATOMIZATION OF 1 GRAM OF OIL

to the boundary of aperture of the precombustion chamber. This method of application is shown on Fig. 19, which represents the precombustion chamber used in a Venn Severin engine.

(2) The theoretical boundary of burning. Let it be assumed that the chemical combination commences immediately at the

orifice and lasts until all the oil in the spray is consumed. Then the spray will be completely burned at such distance that the corresponding cone element contains the air necessary for combustion. The weight f of fuel requires a weight of air αf for its combustion (α = 12–15, depending on the composition of the fuel). The volume of this air is $V = (\alpha f/\rho)$; ρ being the weight of unit volume of air at the given density. Denoting by y the distance to which the spray can travel before its complete combustion, then the volume of the corresponding spherical sector having one steradian inclosed angle can be expressed by $V = \frac{1}{3} y^3$. From which $y = \sqrt[3]{(3\alpha f/\rho)}$.

Measuring up this distance on the corresponding radii vectors, a surface is obtained which is characteristic of the spray in so far as it represents the shape of that minimum container in which the spray can burn completely. The meridional curve of such a body is shown in Fig. 18.

A large number of charts representing sprays under various conditions will be published in a bulletin of the Pennsylvania State College which is now in preparation. (See Fig. 16.) These seem to point to the following general conclusions:

(1) The dispersion becomes more even when the injection pressure is increased
(2) The dispersion becomes more even when the oil viscosity is decreased
(3) The dispersion becomes more even when the air density is increased
(4) The cone angle increases with increasing oil pressure, increasing air density, and decreasing viscosity of oil
(5) A larger percentage of oil reaches a given distance in a spray having a slenderer spray cone
(6) No definite influence of orifice dimensions was noticeable in using cylindrical orifices of the dimensions employed in the tests discussed.

Finally, one conclusion has to be emphasized: that the experimental method described and the procedure of evaluation developed are easy to apply, and the results give a ready means of judging the spray from the point of view of its applicability in an engine. It is recommended as a useful tool in development work in oil-injection engines, and its general adoption is likely to lead to a better understanding of the influence of spray pattern on the performance of the engine.

These investigations were carried out under the general direction of R. L. Sackett, dean of the School of Engineering, and Prof. F. G. Hechler, head of the Engineering Experiment Station. In both the theoretical and experimental work Dr. P. H. Schweitzer, associate professor of engineering research, contributed several suggestions. O. F. Zahn, Jr., graduate assistant, made the greater part of the experiments and worked out the large number of numerical computations with painstaking care.

References

(1) R. A. Castleman, Jr., "The Mechanism of the Atomization of Liquids," Bureau of Standards Journal of Research, March, 1931, pp. 369–376. In this publication reference is made to Lord Rayleigh, Proc. London Math. Soc., 1879; "Theory of Sound," chapter 20; Plateau, "Statique experimentale et theorique," Paris, 1873; F. N. Scheubel, "Ueber die Zerstaeubung in Vergasern," Jahrbuch der W.G.L., 1927.

(2) H. Triebnigg, "Der Einblase und Einspritzvorgang bei Dieselmaschinen," Julius Springer, Wien, 1925.

(3) A. L. Bird, "Experiments on Oil Jets and Their Ignition," Proc. Inst. Mech. Eng., Nov., 1926.

(4) R. Kuehn, "Atomization of Liquid Fuels," *Der Motorwagen*, Dec. 10, 1924, Jan. 20 and Feb. 10, 1925; translation N.A.C.A. Tech. Mem. 329, 330, 331.

(5) A. E. Thiemann, "Fahrzeugdieselmotoren," R. C. Schmidt & Co., Berlin, 1929.

(6) P. H. Schweitzer, "Oil Spray Research at the Pennsylvania State College," paper presented at the First Oil Power Conference at Pennsylvania State College, 1927, Pennsylvania State College Technical Bulletin No. 5, pp. 65–70.

(7) P. H. Schweitzer, "Oil Spray Research at the Pennsylvania State College," paper presented at the Second Oil Power Conference at Pennsylvania State College, 1928, Pennsylvania State College Technical Bulletin No. 5, pp. 175–187.

(8) K. J. De Juhasz, "Some Results of the Oil-Spray Research at Pennsylvania State College," paper presented at the Third Oil Power Conference at Pennsylvania State College, 1929, Pennsylvania State College Technical Bulletin No. 8, pp. 119–131.

(9) P. H. Schweitzer, "Factors in Diesel Spray-Nozzle Design," paper presented at the Fourth Oil Power Conference at Pennsylvania State College, 1930, Pennsylvania State College Technical Bulletin No. 12.

Appendix 1

ENERGY OF INJECTION

(1) *Solid Injection.* Pressure energy of oil:

$$L_o = G_o \frac{P_1 - P_2}{\gamma_o}$$

Example: Assumed values in metric units:

G_o = 1 gram = 0.001 kg.
γ_o = 900 kg. per cu. m.
P_2 = 35 at. = 350,000 kg. per sq. m.
P_1 = 250 at. = 2,500,000 kg. per sq. m.
$L_o = \frac{2150}{900}$ = 2.39 m-kg. per g. = 7840 ft-lb. per lb.

(2) *Air Injection.* Expansion energy of air assuming isothermic expansion:

$$L = G_a RT \log_e \frac{P_1}{P_2}$$

Example: Assumed values in metric units:

G_a = 1 gram = 0.001 kg.
R = 29.97
T = 323 deg. abs. = 50 deg. cent.
P_2 = 35 at. $\frac{P_1}{P_2} = 2$; $1 \log_e \frac{P_1}{P_2} = 0.6931$
P_1 = 70 at.
L_a = 6.56 m-kg. per g. = 21,500 ft-lb. per lb.

Using 2 gram of air for the injection of 1 gram of oil, the total energy is:

$$L_{total} = 2 L_a + L_o = 2 \times 6.56 + \frac{10{,}000\,(70 - 35)}{900 \times 1000}$$
$$= 13.32 + 0.39 = 13.71 \text{ m-kg. per g.} = 45{,}000 \text{ ft-lb. per lb.}$$

In order to supply the same amount of energy with solid injection, P_1 pressure has to be employed.

$$L_{total} = 13.71 = \frac{P_1 - 350{,}000}{900 \times 1000}$$

P_1 = 1270 kg. per sq. cm. = 18,100 lb. per sq. in.

Appendix 2

TOTAL SURFACE OF ATOMIZED DROPLETS

One gram of oil has a volume V_1 = 1/0.900 cu. cm. = 1.11

cu. cm., equivalent to a sphere $V_1 = \frac{4}{3} R_1^3 \pi$ having a radius $R_1 = \sqrt[3]{3V_1/4\pi} = 0.64$ cm.

The surface of this sphere is $A_1 = 4\pi R_1^2 = 5.15$ cm.

When this compact sphere is broken up into n droplets of r_n radius, the following relations hold:

$$V = \frac{4}{3} R_1^3 \pi = n \frac{4}{3} r_n^3 \pi; \quad r_n = R_1 \frac{1}{\sqrt[3]{n}}; \quad \frac{R_1}{r_n} = \sqrt[3]{n}$$

$$A_n = n 4\pi r_n^2 = \sqrt[3]{n} A_1 = \frac{R_1}{r_n} A$$

i.e., the area is increased $\frac{R_1}{r_n}$ times.

This area is equivalent to the surface of *one* sphere of R_n radius: $A_n = 4\pi R_n^2$; $R_n = \sqrt{\frac{A_n}{4\pi}} = \sqrt[6]{n} R_1 = R_1 \sqrt{\frac{R_1}{r_n}}$

The increased area is equivalent to spreading out the oil in a δ thickness

$$\delta = \frac{V_1}{A_n} = \frac{r_n}{3}$$

The energy due to surface tension is:

$$W_n = \sigma A_n$$

Assuming a surface tension $\sigma = 0.003$ kg. per m.,

$$r_n = 0.0002 \text{ cm.}; \quad A_n = \frac{R_1}{r_n} A_1 = 16{,}400 \text{ sq. cm.}$$

then

$$W_n = 0.005 \text{ kg-m.}$$

In a sphere the surface tension σ causes a pressure p for which the following equation holds:

$$\sigma 2r\pi = p r^2 \pi$$

$$p = \frac{2\sigma}{r}$$

In a droplet of $r_n = 0.0002$ cm. radius, the internal pressure due to surface tension becomes: $p = 0.3$ kg. per sq. cm. = 4.2 lb. per sq. in.

Appendix 3

APPLICATION OF SPRAYS

I Combustion
- 1 Engines, carburetor type
- 2 Engines, oil-injection type
- 3 Oil furnaces for industrial and household use

II Textile and Related Industries. Humidification of air in the manufacture of
- 1 Textiles
- 2 Rayon
- 3 Paper
- 4 Photo films and sensitized paper
- 5 Hides

III Agriculture and Food Industries
- 1 Artificial rain
- 2 Spraying insecticides (tree fruits, vines, potatoes, corn, etc.)
- 3 Spraying for destroying weeds
- 4 Manufacture of evaporated milk

IV Chemical Industries
- 1 Oxidation and reduction
- 2 Washing of gas
- 3 Dust and smoke precipitation
- 4 Acceleration of chemical reactions

V Deposition of a Thin Layer of Material
- 1 Painting, varnishing
- 2 Metals (Schoop's process)
- 3 Quartz (for large astronomical mirrors)
- 4 Concrete (Gunite process)

VI Medical and Sanitary Applications
- 1 Inhalators, throat sprayers
- 2 Disinfectors, perfume atomizers
- 3 Air conditioning
- 4 Dust precipitation

VII Miscellaneous
- 1 Fire extinguishers
- 2 Car washers

Discussion

K. J. E. Hesselman.[3] The researches carried out on oil-spray dispersion are extremely valuable and interesting. It was of great interest to see that researches have been made which the writer during practical work with oil motors has found desirable and necessary and which he also partly carried through in a similar way. There is no doubt that it is of the greatest importance for development that such researches have been made and published for engineers working in this field.

F. Sass.[4] The writer has followed the researches of the Pennsylvania State College with great interest from the very beginning and has derived valuable information, especially from the ingenious experiments on spray formation carried out by Mr. De Juhasz and Dr. Schweitzer. In the development of the solid-injection oil engine much credit is due to these investigators for their contribution to the knowledge of the characteristics of oil sprays. This knowledge is of great value to the engine designer, providing him with a surer basis for his design. In fact, more is known today about solid injection than about air injection, and this is greatly due to the investigations carried out at Pennsylvania State College.

None of the statements contained in the paper is in disagreement with the writers own experiments. Among the findings of the author it is particularly noteworthy that evacuating the space into which the spray is injected does not alter materially the characteristics of the spray. This is an important evidence against the theory which attributes the spray formation chiefly to air friction. In another place the author has already given expression to his doubt about this hypothesis.

One may look forward with particular interest to the publication on the dispersion of oil sprays, as stated by the author. In case further investigations should be carried out at the Pennsylvania State College on the penetration of sprays—which is also important for the design of solid-injection oil engines—then in the writer's opinion direct observation by means of spark photography would be highly desirable. To the writer's knowledge, at Pennsylvania State up till now only the indirect method of the ballistic pendulum has been used, similarly to the one used previously by Riehm. In this method it is unavoidable—even if only one single injection is made—to impart a momentum also to an unknown mass of air. Thereby to the impacting mass of oil the unknown mass of air is added, which introduces an error

[3] Stockholm, Sweden.

[4] Allgemeine Electricitaets-Gesellschaft, Berlin, Germany.

into the calculation of the spray velocity from the impact-force equation $P = mv$. For this reason the ballistic pendulum cannot supply as accurate data as can be obtained by means of the spark photography method.

R. A. CASTLEMAN, JR.[5] In considering the mechanism of the atomization that accompanies solid injection, it seems hardly necessary to point out again that "air stream atomization" only was specifically treated in the paper to which reference (1) applies and that solid injection was merely included as a case evidently physically similar as regards the relative motion of gas and liquid at their interface. The necessity for this method of treatment arose from the fact that Rayleigh's analysis could only be applied when the ligament size was known, and this had to be computed from that of the drops formed when a liquid is atomized; and the latter appeared to have been satisfactorily determined only for "air stream atomization." However, Sass has recently made preliminary reports (F. Sass, "Kompressorlose Dieselmaschinen," Springer, 1929, p. 45) on results of drop-size measurements in solid injection performed at the A.E.G., in which a method apparently initiated by Woltjen (Woltjen, Tech. Hochschule at Darmstadt Dissertation, 1925) was used. In this the spray is injected into a high-pressure chamber, where it is caught in a "dispersing liquid" and the drops are examined microphotographically. While these results show many perhaps surprisingly large drops and considerable distribution of drop sizes, they also show many drops small enough to account for a similar mechanism in this case. It will appear later that both the larger drops and the size distribution are to be expected.

Now there are evidently other forces acting simultaneously in this case (say those due to fluid friction in the nozzle), so it seems important to consider whether, and in how far, experiments really indicate that this simple theory cannot be applied to solid injection.

Considering first the results of the Pennsylvania State tests which are claimed to disprove the influence of surface forces:

(*a*) The observed effect on drop size is not very clear, but it is gathered that this did not appear to change much with air density, nor is the method of measuring this drop size apparent. The familiar N.A.C.A. pictures indicate the absolute impossibility of estimating the drop size in such a spray, even when aided by the electric spark. Sass (loc. cit., p. 46), however, reports considerable and consistent increase in drop size with decrease in air density, thus indicating dependence of drop formation on surface forces.

(*b*) The reported observations of the effect of air density on cone angle are to be expected. Joachim and Beardsley found (N.A.C.A. Report No. 268, 1927) (1) a decrease of cone angle, for a "non-centrifugal" spray, from about 22 deg. at 200 lb. per sq. in. chamber pressure to about 8 deg. at atmospheric; (2) a simultaneous increase in cone angle for a spray deliberately made "centrifugal." Recognizing that any liquid injected at such high speeds must have some radial velocity on account of turbulence, it seems that (2) may equal (1), and one would not expect much further change in cone angle when the air density is reduced to a very low value; in other words, the effect reported in this paper seems to be due to the mechanical action of the medium on the drops after their formation.

Considering next the probable bearing of some other observations:

(1) It has been recognized for some time that there is marked drop-size inhomogeneity in sprays formed by solid injection, and the importance of this as regards penetration was noted. The degree of this inhomogeneity was only recently revealed, however, by some experiments at Langley Field, when Rothrock (N.A.C.A. Tech. Note No. 376, May, 1931), tearing away the enshrouding veil of small drops with a high-speed air stream, revealed a very coarsely "atomized" core. Now this is to be expected if the "atomization" is due to surface forces: The velocity difference between air and liquid would be greatest at the outer cone element; as the interior is entered this relative speed drops to a value probably too low for atomization. This, however, seems to account for both the very large drops (which cannot be completely gotten rid of even by the use of very high injection pressures) and the size distribution reported, for instance, by Sass (loc. cit.).

(2) The familiar appearance of these jets, that the atomization seems to occur very close to or even within the nozzle, is also to be expected. The velocity difference is greatest at this point (both the air is accelerated and the liquid decelerated in further progress); so that the interfacial forces must act then if at all. Ligaments fine enough to break up into the smaller drops observed in the spray would have life periods of the order of only 10^{-5} sec.; so that they would collapse within 1 or 2 mm. of the nozzle, which is consistent with N.A.C.A. observations.

It thus appears that close examination of the experimental results so far reported shows that none really seems to prove that surface forces do not influence atomization in solid injection, while much can be found to support this view. Remembering (1) how easy it is to ruffle the surface of a liquid, (2) the very brief life period of ligaments fine enough to break up into the smaller drops found in the spray (order of 10^{-5} sec.), and (3) the manner in which the surface forces are constrained to act, it seems legitimate, at least until real evidence to the contrary is produced, to adopt the pleasing simplicity of the explanation that seems to suffice for air-stream atomization, and thus get just as clear a mental picture in the case of solid injection.

Credit is due the author for recognizing the need for further argument in the case of solid injection.

P. H. SCHWEITZER.[6] Attention has been called to the advantage of the square-edged orifices in producing a better dispersion than those obtained by rounded orifices.

It has to be admitted that as a rule the square-edged orifices do produce better atomization and dispersion than those used in our experiments. In spite of that, the use of square-edged orifices is discouraged in engines, for two reasons. First, it is very hard to produce a geometrically correct, clean square-edged orifice; and still more so, several of them with an equal hydraulic resistance and discharge. Second, square-edged orifices that are so in the beginning do not stay that way. The scouring action of the oil passing through rounds off the edge of the orifices. The discharge coefficient of a short square-edged orifice is about 0.65 and that of a slightly chamfered orifice is 0.80. If one places a square-edged orifice in the oil line, the discharge coefficient rises in a few days from 0.65 to 0.80. Microscopic examination reveals a slight rounding of the edges by use.

We started out using square-edged orifices for scientific experiments for reproducible results. We abandoned that in favor of chamfered orifices, which we obtain by rubbing emery powder against the edges of the orifice, the chamfer amounting to 0.004 to 0.005 in. We found that small differences in the amount and angle of the chamfer do not make as much difference as unavoidable burrs and irreguiarities in square-edged orifices.

A. M. ROTHROCK.[7] The work that is being conducted at the Pennsylvania State College on the dispersion and energy of the fuel spray in conjunction with the work that is being conducted

[5] Assistant Physicist, Bureau of Standards, Washington, D. C.

[6] Associate Professor of Engineering Research, Pennsylvania State College, State College, Pa.

[7] Associate Physicist, Langley Field, Va.

at the Langley Memorial Aeronautical Laboratory on the spray penetration, formation, and atomization are giving us a better concept of the laws which control the actions of the fuel spray in the compression-ignition engine. The writer is particularly interested in Professor De Juhasz's statements regarding fuel sprayed into a vacuum. The laboratory has also done some work along this line. With the N.A.C.A. spray photography equipment we have taken high-speed motion pictures of sprays injected into an air pressure of 0.1 in. of mercury. The results were quite interesting. Of course the rate of spray-tip penetration was extremely high. The spray itself had a cone angle of approximately 2 deg. The photographs showed that there was no fuel torn away from the edge of the spray, as is the case when the fuel is sprayed into air at a pressure of 1 atmosphere. We have also photographed fuel sprayed into water. The photographs appeared similar to those of fuel sprayed into dense air, although the penetration was decreased. In regard to the dispersion of the spray into air at densities corresponding to those in the combustion chambers of compression-ignition engines, we made a very simple test of spraying against a piece of modeling clay mounted 5 in. from the nozzle. The spray made a small hole approximately $^1/_8$ in. in diameter in the clay, indicating that the greater part of the spray was contained in a cone with an angle of approximately 2 deg. This is in accordance with the results presented by the author. The writer agrees that the resistance to air is not the sole cause of the atomization of the spray. If the flow through the nozzle is turbulent, the jet is in an unstable condition as it issues from the discharge orifice.

Recent tests that were conducted at this laboratory, however, on atomization by Mr. D. W. Lee have shown that the atomization is not changed to any great extent by any of the means available for atomizing the fuel in a hydraulic injection system. We have measured the atomization of sprays from single round-hole orifices, of sprays from injection valves employing helical grooves, of sprays impinging against each other close to the nozzle, and of sprays injected against a metal plate close to the nozzle. Certainly if the atomization does not take place within the nozzle, it does take place within a fraction of an inch after leaving the nozzle. As a result of these tests the writer has come to the conclusion that higher mean effective pressures in the compression-ignition engine will be obtained only when there has been improvement in the uniformity of the dispersion of the spray, either by the methods listed or by the intelligent use of air flow. Dispersion tests made under these conditions would be extremely interesting. Of course, there is the possibility that the period of after-burning in the combustion chamber is partially caused by the ignition lag of the larger drops in the spray. In this case it will be extremely difficult to eliminate it.

The data that the author has presented are for injection periods of approximately 0.05 sec. His results show that under this condition, at an injection pressure of 2200 lb. per sq. in., 80 per cent of the fuel penetrated to a distance of 14 in. When the injection pressure was raised to 8200 lb. per sq. in., the quantity penetrating to 14 in. decreased to 26 per cent of the total fuel quantity. The writer would like to ask whether this is attributed to the decrease in the mean drop diameter caused by the increase in the injection pressure. (Incidentally should not the f curves be labeled "per cent of total per steradian" instead of "grams per steradian?") He would be interested in seeing the dispersion tests continued for injection periods of approximately 0.003 sec., since this time interval is approximately that used for the injection of the fuel in compression-ignition engines running at speeds of 1500 r.p.m. and over. Has the region from the nozzle to a distance of 3.5 in. from the nozzle been explored? The writer realizes the difficulties to be encountered in this region, but it is very necessary for those who are working with high-speed engines to know what the dispersion is in combustion chambers having a maximum length of not more than 4 in.

A. L. Bird.[8] The subject of spray formation is a fascinating one and has tempted many to try to go further than Lord Rayleigh, who proved the tendency to ripple formation at a boundary for low speeds, and to determine analytically the factors which decide that drops of a certain size shall be formed rather than others very much larger or smaller.

Probably the author is wise in concentrating on the investigation of the laws of behavior rather than those of formation, but his reference to injection into a partial vacuum is of particular interest in view of some similar experiments at Cambridge, England. Using a chamber 20 ft. long and 1 ft. diameter, exhausted to $^1/_{10}$ atm., some dispersion was still found, though not as much apparently as by the author; in fact, the splash scarcely covered the end cover of the chamber; whereas in the open the same jet did not reach 20 ft., the center of gravity of the deposited drops being about 10 ft. distant from the nozzle and the lateral spread some 2 ft. there. This result is preliminary only, and no special trouble was taken to select a good nozzle. Possibly with a smooth entrance and large length-to-diameter ratio, the dispersion might have been even less. It would appear to be necessary to distinguish also between nozzles with a small-lift value above them (in which the drops appear to emerge ready formed) and true orifices unaffected by any valve, the latter being the type mainly tried at Cambridge.

The results from the ingenious dispersion apparatus should be of great interest and usefulness. Conceivably in the future the author may be able to elaborate it so as to measure the impact on each of the platforms.

Author's Closure

It is gratifying and encouraging to note that the investigations into the basic phenomena of the fuel spray carried out at the Pennsylvania State College find interest among the ranks of the research fraternity as well as meet the endorsement of distinguished exponents of the practical development of the solid-injection oil engine. Thanks are due to all the discussors for their suggestions and constructive criticism.

Dr. Sass rightly recognizes the error inherent in the pendulum method of determining the mean velocity of the spray. It is to be borne in mind, however, that all velocity determinations have their errors and limitations. For example, the spark-photography method as used in the N.A.C.A. investigations and by the A.E.G. under the direction of Dr. Sass give only the velocity of the spray tip, which itself is changing, and is replaced by new oil every moment. The double-disk (stroboscopic and target disks) method (Ref. 8) gives the instantaneous values as an average over the distance of the disks. Therefore the various methods need the support of each other, and the most probable velocity values can be determined by checking and correlating the results obtained by each.

Regarding Mr. Castleman's discussion, the author wishes to emphasize the statement in his paper that the atomization and dispersion phenomena cannot be accounted for solely by the air friction forces. These forces, however, do have an important influence on the spray characteristics.

It is of interest to note that both Mr. Rothrock with the N.A.C.A. and Professor Bird in Cambridge carried out injection tests in rarefied air. Many of us would like to see published the spark photograph of such a spray, especially in conjunction with sprays injected into air at various densities, but otherwise under identical conditions, as spark photography is able to record finer details than are obtainable by any other means.

[8] Cambridge, England.

In experiments conducted at the Pennsylvania State College by Lieut. J. P. Thew, U.S.N., it was found that the fineness of atomization largely depended on the oil pressure, the drop size decreasing with the increase in pressure. The resulting increase in air resistance accounts for the decrease in the percentage of spray reaching a given cross-section, when the oil pressure is increased.

The *f* curves on the charts refer to the flux due to 1 gram of total discharge. Therefore the expressions "per cent of total per steradian" or "grams per steradian" are equally justified.

The exploration of smaller sprays would only involve making a dispersion rack of smaller dimensions all over, and a refinement in the weighing technique on account of the smaller weights to be measured. At Pennsylvania State the diameter of the spray cone at different distances from the nozzle was measured by means of a caliper fixed on the traveling platform of the spray pendulum. This method proved satisfactory and accurate.

It would be difficult to combine satisfactorily the function of a dispersion rack and the spray pendulum in one instrument. It is planned, however, to measure with the spray pendulum the momentum of any portion of a spray by simply placing in front of the pendulum plate a stationary plate which permits the passing of the spray portion to be investigated and shields the rest.

Owing to the small momentum of the outer portions of the spray, it is necessary to build the pendulum extremely sensitive. Such a modification has been tried out on our spray pendulum and may be further developed.

OGP-53-6

Experiments on the Flow of Air Through Engine Valves

By E. S. DENNISON,[1] T. C. KUCHLER,[2] AND D. W. SMITH,[3] PHILADELPHIA, PA.

The resistance of air flow through poppet valves has been tested experimentally by the authors. The experiments which they carried out and describe include flow in both inlet and exhaust directions, and the ranges of valve opening and of pressure head extend to or beyond the limits which normally may be utilized in practice. Individual valves were tested variously mounted, and pairs of valves were tested as grouped in an actual design.

The experiments have been extended in an effort to improve the flow characteristics of the poppet valve by modification of its shape and that of the valve approaches.



E. S. DENNISON



T. C. KUCHLER



D. W. SMITH

THE choice of valves and gear for a high-speed internal-combustion engine is limited by space, by permissible mass and acceleration of the parts, and by the necessity for reducing throttling losses. Features which must be determined include the choice of number, size, and type of valve, the disposition of valves and passages, and the determination of valve-operating-gear details.

So far as the problem is purely mechanical in nature, as it is when dealing with the valve-operating gear, it may be rationally dealt with upon the basis of known properties of materials, and of familiar physical laws. But a rational treatment of the whole problem requires a similar accurate knowledge, either analytical or empirical, of laws governing the flow of air through valves and adjacent passages.

There is perhaps no existing knowledge which would permit an analytical treatment of the flow characteristics of an ordinary poppet valve as mounted in a practicable engine design. Data suitable for design purposes must be found experimentally, and several investigators have given their attention to securing such information.

An examination of the existing data brought out several conditions which seemed to justify further work in the field. In the first place, the information was found to be incomplete and to apply to only a few forms of valve assembly. Furthermore, it showed that considerable differences of behavior might result from even a minor change in the valve or port design. Secondly, no published data exist on the effect of a curved approach passage as usually designed, nor of grouping valves in a pair in conjunction with a single passage; certainty in this respect could be had only by a test of the valves in place in the actual cylinder head. Thirdly, it seemed that there was a possibility of improving upon the usual type of poppet valve as an element in a fluid conduit system. A part of the program was therefore directed to the latter object, without a view to any immediate application in practice.

Questions which appeared to have been adequately answered by the work of previous investigators were not reopened. A review of a few published results is given here, in order that the reasons for limiting the direction of the present work may be apparent. A mere check on results already at hand was not sought after.

Summary of Published Experimental Results

Nutting and Lewis.[4] A limited series of experiments was made on the flow of air in the intake direction through various valve combinations into a model combustion chamber of 5.75 in. diameter joined to a cylinder of 5 in. diameter. The combinations included a single $2^1/_2$-in. valve, two $1^3/_4$-in. valves in parallel, and two $1^1/_4$-in. valves in parallel. Each valve of a pair was

---

[1] Section Engineer, Four-Cycle Oil-Engine Section, Westinghouse Elec. & Mfg. Co., South Philadelphia Works. Jun. A.S.M.E. Mr. Dennison is a graduate of the Massachusetts Institute of Technology, class of 1921, with the degree of S.B. in Mechanical Engineering. His first position was with the Carbondale Machine Company, Carbondale, Pa. In 1922 and 1923 he was employed by the U. S. Board of Helium Engineers, Washington, D. C., in the calculation of gas-liquefaction processes used in the extraction of helium gas, and in low-temperature research. From 1923 to 1927 he was employed by the Curtis Oil Engine Company, New York City, in the development of a supercharged high-speed Diesel engine. After a brief connection with the Doherty Research Corporation, New York, he joined the Westinghouse Company as development engineer. He has subsequently taken charge of design and other engineering activities relating to high-speed oil engines of the four-cycle type.

[2] Oil-Engine Division, Westinghouse Elec. & Mfg. Co., South Philadelphia Works. Jun. A.S.M.E. Mr. Kuchler was graduated in 1926 from the Swiss Federal College of Science, as a pupil of Dr. A. Stodola. From 1926 to 1928 he was employed by Sulzer Bros., Winterthur, Switzerland, as a draftsman in the marine Diesel engine department. He has been connected with the Westinghouse Company since 1928, and has been engaged in research and development in connection with high-speed oil engines.

[3] Oil-Engine Division, Westinghouse Elec. & Mfg. Co., South Philadelphia Works. Jun. A.S.M.E. Mr. Smith was graduated from Pennsylvania State College in 1929, with the degree of B.S. in M.E. Previous to this, he was employed as a draftsman for four years by the Union Switch and Signal Company and the Westinghouse Company. During this time he took engineering work in the night school of Carnegie Institute of Technology. After receiving his degree from Pennsylvania State College, he returned to the Westinghouse Company at East Pittsburgh, where he was engaged for a year and a half as a member of the student engineering course, the mechanical design school, and in the mechanical engineer's office. He was then transferred to his present location.

Presented at the National Oil and Gas Power Meeting, University of Wisconsin, Madison, Wis., June 23–26, 1931, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[4] E. M. Nutting and G. W. Lewis, "Air Flow Through Poppet Valves," National Advisory Committee for Aeronautics, no. 24, Washington, 1918.

connected to an individual curved air passage. Details of valves, passages, etc., are not given. It is not stated that all valves were geometrically similar, although this seems to be assumed in their interpretation of the results. The tests covered a full range of valve lifts, and pressure heads up to 28 in. of water at small openings.

Their results showed a decreasing coefficient of discharge with increasing lift. The coefficient of discharge, when plotted on a non-dimensional basis against Lift/Port Diameter showed substantial identity for the $1^3/_4$-in. and $2^1/_2$-in. valves. Results for the $1^1/_4$-in. valves were at variance; the latter results were discounted by the original authors because of a special condition in the air passage.

Experiments of this character usually involve the assumption that results obtained with one size of valve may be applied to similar valves of other sizes, provided that geometrical proportions of the adjacent parts are also similar. This should be

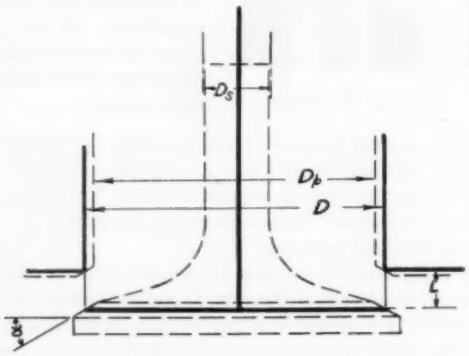


FIG. 1 ELEMENTARY DISK VALVE

correct except for the part played by surface friction, which presumably is small. The same assumption is used in the interpretation of most wind-tunnel results. The experiments of Nutting and Lewis partially confirm the assumption as it applies to valve flow, and the experiments described in the present paper have been limited to one size of valve.

Their report remarks (p. 19): "Venturi effects in the passage immediately above the valve might be productive of excellent results." Seemingly no attempt was made to follow this suggestion further.

Tanaka.[5] The extensive work by this author is notable for thoroughness and for refinement of method. A very lengthy series of tests was made with air flow through both inlet and exhaust valves. These cover the full range of valve lifts with various pressure drops. The effect of variation in valve details was systematically studied. Features which were progressively varied included the angle of cone seat; projected width of seat, or overlap; valve-fillet radius, and valve-head angle. Model valves were tested in a concentric apparatus in connection with a simple cylindrical port with the approach rounded at the inlet side. The result of these experiments is largely to confirm accepted valve practice, since only minor differences are found by variations of the above-mentioned features within practicable limits. For example, cone seat angles of 30 deg. and 45 deg. are roughly equivalent in result, whereas with a 60-deg. angle the resistance becomes excessive.

Sharp discontinuities of flow were found, occurring at definite valve lifts, and in some degree modified by changes in pressure head. Four distinct modes of flow are noted, and a discontinuity corresponds to a shift from one mode to another. The differences of flow are found to be associated with sharp corners at the edges of the valve and valve seat. The existence of four types of flow was proved by photographs and confirmed analytically. By rounding all corners to a radius of 4 mm. (0.158 in.), the sharp discontinuities are nearly obliterated, and resistance of the valve is decreased. Tanaka finds the rounding of these edges to be the most advantageous means of improving valve flow. He remarks that rounding of the corners in effect makes a convergent-divergent nozzle of the crevice between valve and seat, but it is surprising to note that he does not attempt to extend the application of the principle.

Schrenk.[6] Experiments were made upon the resistance to flow of water in the inlet direction through plate valves with square and conical seats. With flat plate valves, two principal types of flow were found. Valves having cone seats showed at least six types with possible minor variations.

A type of valve was developed employing a rounded nozzle inlet and a diffuser-type discharge. Resistance by this means was reduced to a small fraction as compared with the worst case (flat-seated valve with all edges sharp).

The valves tested were not of the poppet type, and the improved valve, unfortunately, is not applicable to engine purposes.

The subject was treated analytically by Eck.[7] He applies his work to the experiments of Schrenk and finds that mathematical explanations are possible of the types of flow which were observed. As yet no attempt has been made to apply his methods to the results here presented.

FACTORS CONTROLLING AIR FLOW THROUGH VALVES

NOMENCLATURE

D = mean diameter of valve seat, in.
D_s = valve-stem diameter, in.
D_p = valve-port diameter, in.
L = valve lift, in.
A_l = valve "limit area," sq. in.
= πDL when $L < D/4$
= $\pi D^2/4$ when $L > D/4$
A_a = actual minimum flow section at valve, sq. in.
A_n = "unity orifice" area, namely, the area of an imaginary orifice with unity coefficient of flow, which is the calculated equivalent of an actual valve under given conditions, sq. in.
μ = A_n/A_a = orifice coefficient of the valve, based upon actual minimum section
ψ = A_n/A_l = ratio of equivalent orifice to "limit area"
γ = ratio of absolute pressures before and after valve.

ASPECTS OF VALVE OPENING

"Limit Area" A_l. The port diameter D_p is frequently used as a measure of valve size. For purposes of the present discussion the mean seat diameter D is preferred, and is generally used throughout. This preference is due firstly to the fact that the mean diameter corresponds more closely to the elementary disk valve which might replace the actual valve and which determines the mass of the valve and its gear. (See Fig. 1.) Secondly, the

[5] K. Tanaka, "Experimental Research on Poppet Valves," Journal of the Aeronautical Research Institute, Tokyo Imperial University, nos. 22 and 27, 1926.
"Air Flow Through Suction Valve of Conical Seat," Report of the Aeronautical Research Institute, Tokyo Imperial University, nos. 50 and 51, 1929.
"Air Flow Through Exhaust Valve of Conical Seat," Proc. 3rd International Congress for Applied Mechanics, vol. 1, p. 287, Stockholm, 1930.

[6] Prof. Dr. Ing. E. Schrenk, "Versuche über Strömungsarten, Ventilwiderstand und Ventilbelastung," Berlin, 1925.

[7] Bruno Eck, "Potential Flow in Engine Valves," Translation Tech. Memo. no. 343, N.A.C.A., Washington, 1925.

use of D_p carries with it the assumption that the port is to be of simple cylindrical form down to the seat of the valve. This was not the case in all tests, and was found to be in some respects undesirable. These considerations lead to the definition, limit area $= \pi DL$, with a maximum of $\pi D^2/4$, as a measure of the capacity provided in the valve gear.

Actual Area A_a. The actual area is determined from the detail dimensions of the valve and its seat, according to the following equation:

$$A_a = 2\left(R_0 + \frac{L}{2}\sin\alpha\cos\alpha\right)L\cos\alpha \quad \ldots\ldots [1]$$

in which R_0 = inner radius of valve seat $= D_p/2$, and α = valve-seat angle. With large lifts a maximum opening equal to

$$A_a \text{ max.} = \frac{\pi}{4}(D_p^2 - D_s^2)$$

is attained. Where a test is made with the valve removed, A_a becomes $\pi D_p^2/4$. The calculation of area for the special form of valve had to be done graphically for some openings.

Table 1 summarizes the numerical values applying to valves used in the various tests.

TABLE 1 NUMERICAL CONSTANTS FOR TEST VALVES

Type of valve			Standard	"Drop" and "Tulip"
D, in.			2.82	2.82
α, deg.			30	30
D_s, in.			0.625	0.625
D_p, in.			2.625	2.375[1]
L	L/D	A_l	A_a	A_a
0.010	0.00355	0.0886	0.0715	0.0762
0.025	0.00887	0.2215	0.1793	0.1912
0.050	0.01774	0.443	0.360	0.385
0.075	0.0266	0.665	0.542	0.578
0.100	0.0355	0.886	0.726	0.769
0.150	0.0532	1.329	1.098	1.130
0.200	0.0710	1.771	1.476	1.480
0.300	0.1064	2.66	2.249	2.150
0.400	0.1419	3.54	3.045	2.770
0.500	0.1774	4.43	3.87	3.330
0.600	0.213	5.32	4.55	3.710
0.750	0.266	6.23	5.11	3.950
0.900	0.319	6.23	5.11	4.075
1.050	0.372	6.23	5.11	4.123
1.200	0.426	6.23	5.11	4.123
Valve removed	.....	6.23	5.41	4.430

[1] Diameter of throat above valve.

"Unity Orifice" Area. The third aspect of valve opening, namely, the area of the equivalent "unity orifice" is defined as $A_u = \mu A_a$. The overall efficiency of the valve set is given by

$$\Psi = \frac{A_u}{A_l} \quad \ldots\ldots [2]$$

particularly for design purposes. The importance of actual area A_a and of the true coefficient μ lies in their combined effect in the product μA_a.

Valve as Orifice or Nozzle

The familiar treatment of a valve is as a more or less restricted orifice opening into the cylinder. In this case the flow may be calculated by any of the usual formulas for air flow under a moderate head, such as that of Moss for "Normal Air,"[8]

$$G = 157.6 A\mu\sqrt{\left(P_1 - P_2\right)\frac{P_2}{T_1}} \quad \ldots\ldots [3]$$

where G = air flow, lb. per sec.
A = nozzle area, sq. ft.
P_1 = total pressure (static plus velocity pressure) before nozzle, lb. per sq. in. abs.
P_2 = static pressure after nozzle, lb. per sq. in. abs.
T_1 = temperature before nozzle, deg. fahr. abs.
μ = coefficient of discharge.

[8] S. A. Moss, "Measurement of Flow of Air and Gas With Nozzles," Trans. A.S.M.E., vol. 50 (1928), no. 2.

Taking this view of the functioning of the valve, various steps are immediately suggested which may be expected to improve its performance. These are implied in the form of the equation, and in the known means of raising the discharge coefficient of a mouthpiece toward its limiting value of unity. They may be summarized as follows:

1 Employ large passages for approach and discharge. By this means friction will be minimized and the full pressure drop $(P_1 - P_2)$ will be available across the valve itself
2 Design the valve to give a large actual opening for a given lift. That is, a small seat angle is to be preferred to a large one
3 Round the edges of the valve and its seat, particularly on the approach side.

That this approach to the problem is capable of producing valuable results is shown particularly by the experiments of

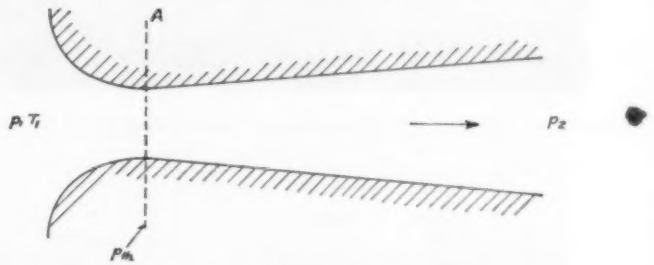


Fig. 2 Elementary Diffuser

Tanaka, already mentioned. The ideal limit is set approximately by the value $\mu = 0.98$ for a rounded mouthpiece; the practical limit is much lower, being imposed by construction requirements and by the unavoidable abrupt change in direction of flow in a poppet-type valve.

Valve as Diffuser

The limit $\mu = 1$ which applies to a rounded mouthpiece is removed if the valve and its passages be constructed to act as a diffuser. The phenomenon of a high throat velocity, with nevertheless, slight overall resistance or pressure drop, is common in the operation of venturi tubes. It seems that this form of construction has not had the application in the design of engine valves which its advantages merit. These advantages appear to be considerable, even after they are discounted—as they must be—on account of the practical design limitations which exist.

Theoretically, the flow of air or gas which will pass through an ideally constructed diffuser is limited only by approach to the velocity of sound at the throat. This in turn is established by the existence in the throat of the critical pressure, namely:

$$P_{th} = \left[\left(\frac{2}{K+1}\right)^{\frac{K}{K-1}}\right]P_1 = 0.53P_1 \text{ (for air)} \quad \ldots\ldots [4]$$

in which P_{th} = static pressure in throat, lb. per sq. in. abs. Thus the flow through the diffuser is the same as the flow through a rounded orifice when $P_2/P_1 \leqq 0.53$, namely, in the form given by Moss[8]

$$G = \frac{76.367 A \mu P_1}{\sqrt{T_1}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [5]$$

in which, in this instance, A is the throat area in square feet.

The ideal diffuser (see Fig. 2) will be capable of attaining this flow even when the overall pressure drop ($P_1 - P_2$) approaches zero as a limit.

The actual diffuser falls far short of this capacity even when conditions are favorable. Its losses are of two kinds: (1) Friction loss, both internal friction, due to turbulence of flow, and friction at the wall surface. (2) Inefficiency in the conversion of velocity head (at the throat) into static pressure head (at the point of discharge). Of the two, the loss by friction is comparatively small.

Referring to the sketch (Fig. 2), the diffuser is divided by the throat section (indicated by the broken line) into two parts. The upstream portion is no more than a rounded mouthpiece.

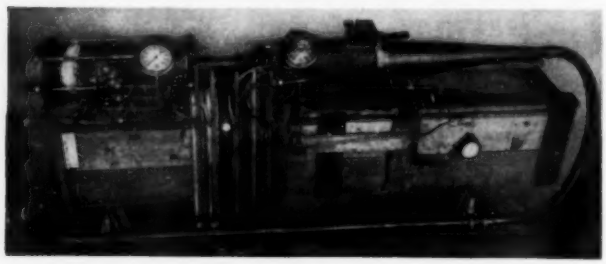


FIG. 3 PHOTOGRAPH OF TEST APPARATUS

true either in the ideal or actual cases. The actual diffuser case is irreversible and is accompanied by an increase of entropy. The increase of entropy may be traced to the two distinct sources of loss, that is, friction and inefficiency of diffuser action.

Difficulty in obtaining a good diffuser form in valve arrangement, particularly for the intake direction of flow, greatly narrows the possibilities of the principle. Its application may nevertheless be looked on as a possible means of going beyond the unity discharge coefficient.

DESCRIPTION OF APPARATUS

The test apparatus consists essentially of a flow meter followed by a suitable holding device for the model under test. Air entering the equipment flows first through the meter, then to the test valve, whence it is discharged to the atmosphere.

The general arrangement is shown by Figs. 3 and 4. The course of the air through the apparatus may be followed by reference to Fig. 4. The items referred to are readily discernible in the photograph, Fig. 3.

Air for the tests was drawn from the shop compressed-air line through a throttling valve. A tank about 30 in. in diameter and 8 ft. in height was secured and filled to a point near the top with coke. Air from the throttle valve entered near the bottom of this tank and flowed upward through the coke. Water was drained out at the bottom. Air after passing through this drier was free of water drops. Provision for heating by means of a steam coil was found unnecessary, and all tests were run at atmospheric temperature.

Entering the test apparatus, the air flows through a diffuser of 10 deg. included angle. This serves the double purpose of

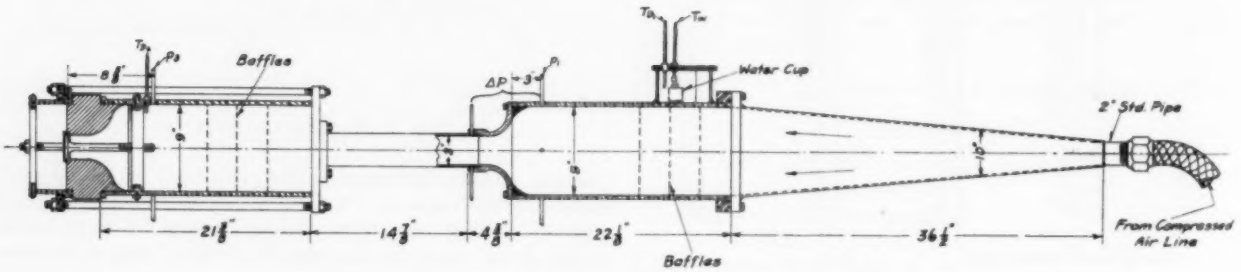


FIG. 4 GENERAL ARRANGEMENT OF TEST APPARATUS

Here the static pressure of air before the valve is converted into velocity head at the throat. In other terms, the internal energy of the gas is converted by adiabatic expansion into kinetic energy. The process is easily accomplished with little loss, the efficiency being perhaps 98 per cent. Downstream from the throat, the reverse process of restoring static head and internal energy proceeds in the expanding nozzle. This is less efficient. Under favorable conditions the efficiency may be 80 or 85 per cent. When the diffuser shape is not favorable it will be much lower. In any case where the included angle of the cone exceeds 8 to 10 deg., the diffuser action will be poor.

It may be noted that thermodynamically the ideal processes of flow, in the nozzle and in the diffuser, are fundamentally different. In the simple nozzle a finite flow requires a finite difference of pressure. The process is adiabatic and irreversible, and is accompanied by an increase of entropy. In the ideal diffuser, a finite flow may take place with only an infinitesimal difference of pressure. The case is then adiabatic and fully reversible. Internal energy is converted to kinetic energy, and is then fully reconverted to internal energy, and the process is isentropic. Both processes, when comparing initial and final states, are isothermal for air in continuous flow. The latter is enabling somewhat higher test pressures and of helping to quiet the flow of air before it enters the flow meter. Velocity distribution over the diffuser discharge area was checked and found uniform.

The diffuser discharges into a 9-in. cylinder fitted with three baffles through which the air flows toward the meter. Above this cylinder is mounted a box, opening into the cylinder and carrying dry-bulb and wet-bulb thermometers (T_{D1} and T_W). This equipment at first included also a small motor-driven fan. The latter was found unnecessary, since the disturbance of air movement in the cylinder was sufficient and the fan operation had no noticeable additional effect on the thermometer readings.

At the end of the 9-in. cylinder, the air passes to a 3-in. tube through a smoothly rounded mouthpiece. This tube is seamless drawn steel, accurate and uniform in diameter and smooth.

A pressure reading (p_1) is taken in the 9-in. cylinder, and a manometer to read pressure drop (ΔP) is connected from the cylinder to a point on the 3-in. tube. Both pressures are static pressures, the connecting tubes leading to small holes smoothly finished at right angles to the inner walls. At each point of measurement, three such holes are used, spaced around the circumference in one plane, and leading into a common pressure tube.

The 3-in. steel tube discharges into another 9-in. cylinder which also is fitted with baffles made of plate with progressively smaller perforations. After the baffles, the air passes to the model or cylinder head, through which it is discharged to the atmosphere. Readings of pressure (p_3) and temperature (T_{D2}) are made immediately before the test model.

sembly. A dial indicator is used in that case for setting the lift.

Pressures p_1 and p_2 are read with alcohol manometers up to about 27 in., then with mercury manometers to about the same height. Above this point, pressures are read with gages, these being checked at frequent intervals in the course of the work. The pressure drop ΔP is read with an inclined alcohol manometer

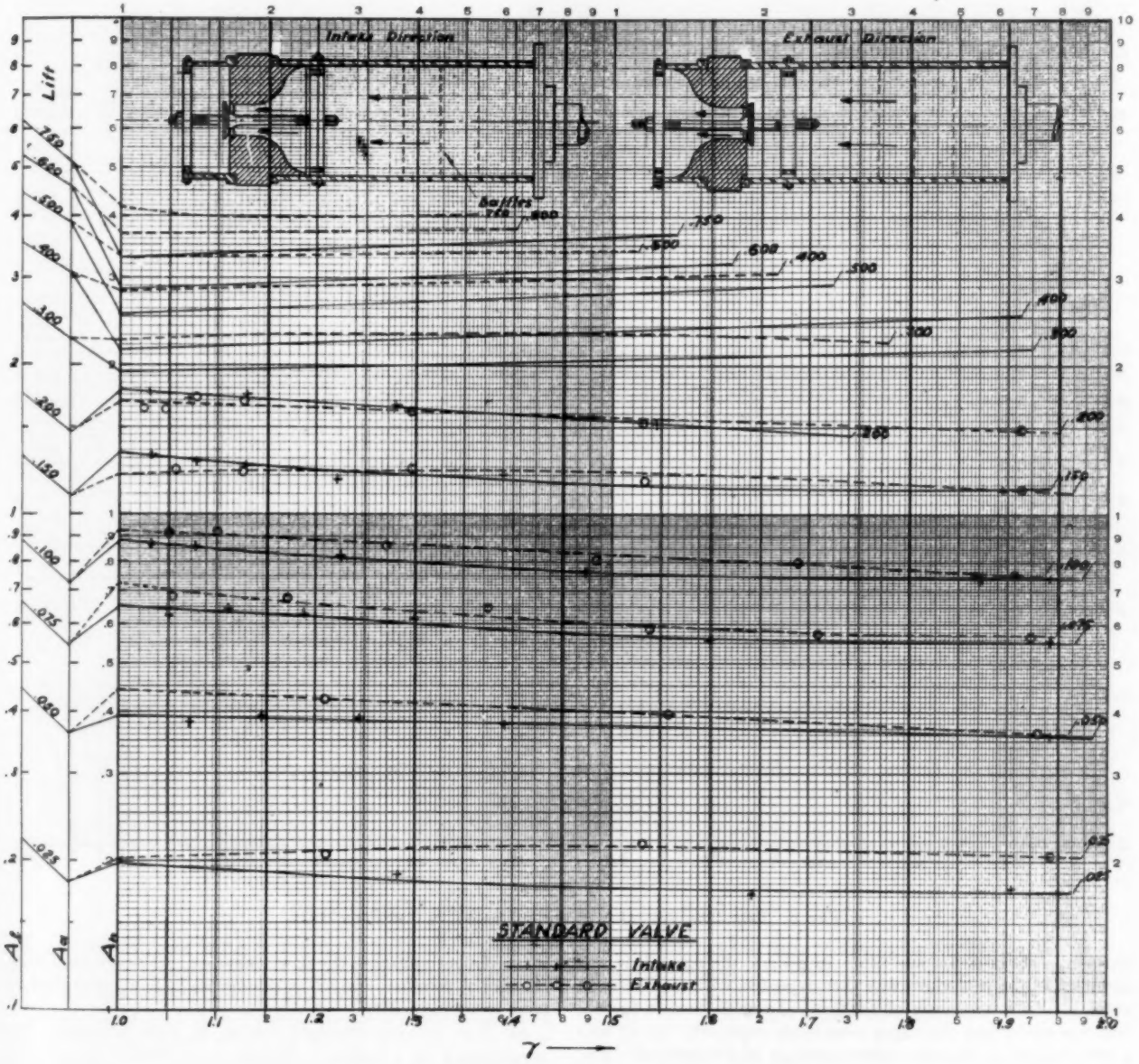


FIG. 5 TEST RESULTS—STANDARD VALVE IN MODEL

The general arrangement of the model for various tests is apparent in the individual sketches (Figs. 5 to 11, incl.). Fig. 4 shows the standard-type valve in place for test of flow in the intake direction. The valve is carried in two light spiders, which are accurately machined to hold it central in the model. These spiders are so shaped as to have slight effect on the flow of air. The lift of the valve is set by means of a thread (20 per inch) on an extension stem. This thread having been checked for accuracy, it is used as a micrometer.

When a cylinder head is being used in test, this is bolted to the 9-in. cylinder as is done when making an actual engine as-

having a slope of 1:10 to a measured length of 19 in., then with vertical alcohol and mercury manometers. Alcohol was preferred to water for use in the manometers because of its superior meniscus.

CONDUCT OF TESTS

The test procedure consisted in establishing a valve lift, opening the throttle valve to a definite position, and allowing air to flow until steady conditions were established. Then a reading was taken and the throttle-valve opening altered. When a complete series of readings with various throttle openings was

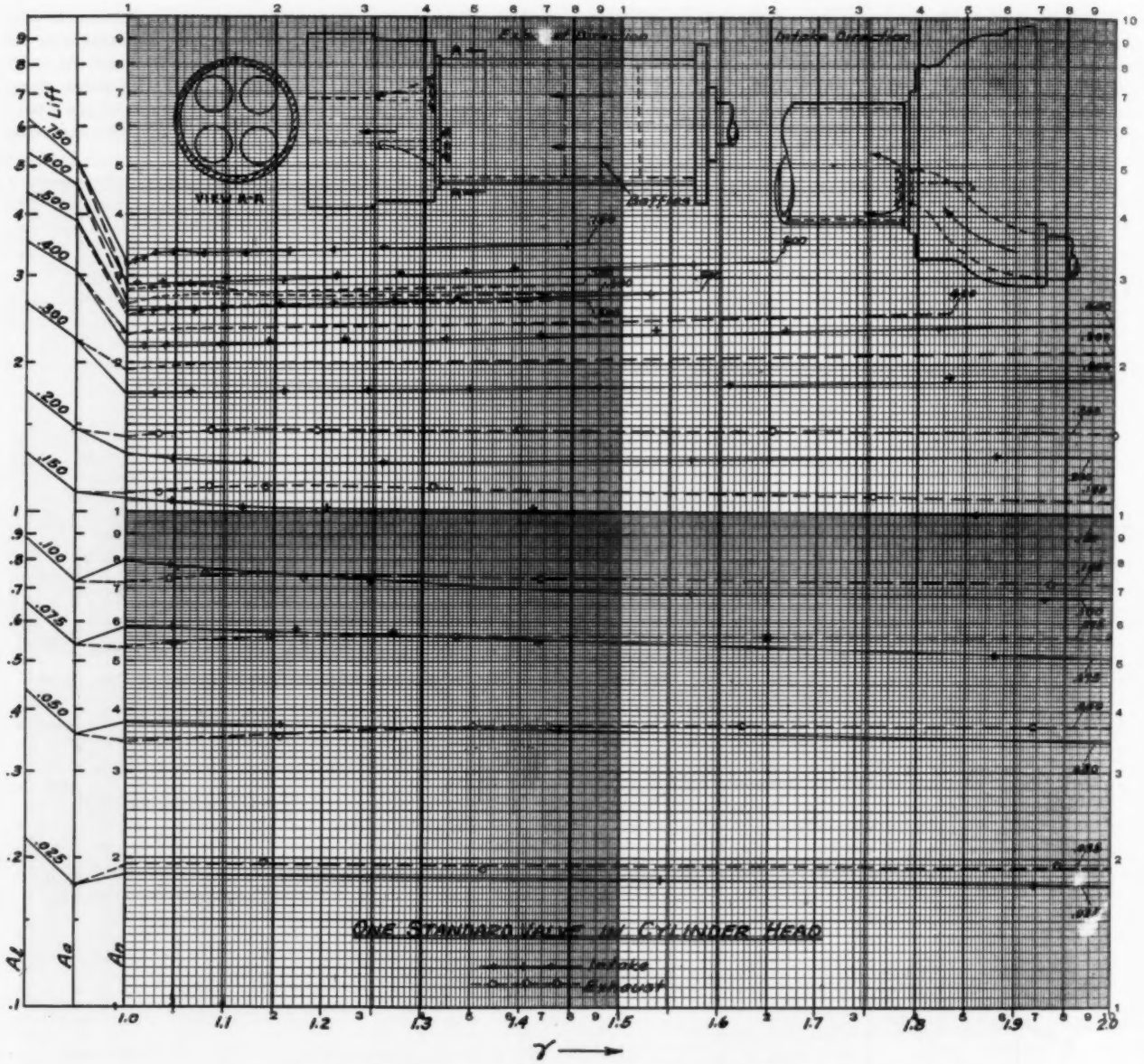


FIG. 6 TEST RESULTS—ONE STANDARD VALVE IN CYLINDER HEAD

finished, a check series was run. In general the agreement was excellent. The procedure was then repeated with an increased valve lift, to a maximum of 1.200 in. In some cases a further test was made with the valve completely removed. At all lifts a wide range of throttle valve openings was employed. The result of this was to extend the pressure ratios of tests with small lifts much beyond the range of ordinary application. Tests with large lifts were restricted by the capacity of the supply-pipe system. Nevertheless, these in all cases cover the generally useful range of pressures.

Readings of barometric pressure and of room temperature were made, in addition to the full series provided for in the test arrangement.

GENERAL OBSERVATIONS ON TESTS

Noteworthy differences were found in the behavior of the special "tulip" and "drop" types of valve on test, as compared with that of the standard valves. These took the form (1) of sharp discontinuities in the flow resistance, with only very slight changes of pressure, and (2) of differences in flow manifested by noise or by instability.

Air flow through standard valves, in both directions, was accompanied by a hissing or rushing sound which changed only in degree as the amount of air flowing and the pressure head increased. No abrupt changes of resistance or of flow characteristics were noticeable at any time.

The special valves in the majority of tests acted similarly, that is, no unusual sound or other behavior was noticeable. However, at points where instability of flow appeared, the other peculiarities in general developed also. The change in resistance was very abrupt and amounted to as much as 4 in. of mercury with an extremely slight difference of flow. The change of resistance was in the direction of a decrease with increasing flow. Hence sometimes actually less pressure drop was required for a large flow than for a smaller one. This effect was most noticeable with intermediate valve openings and in the exhaust direc-

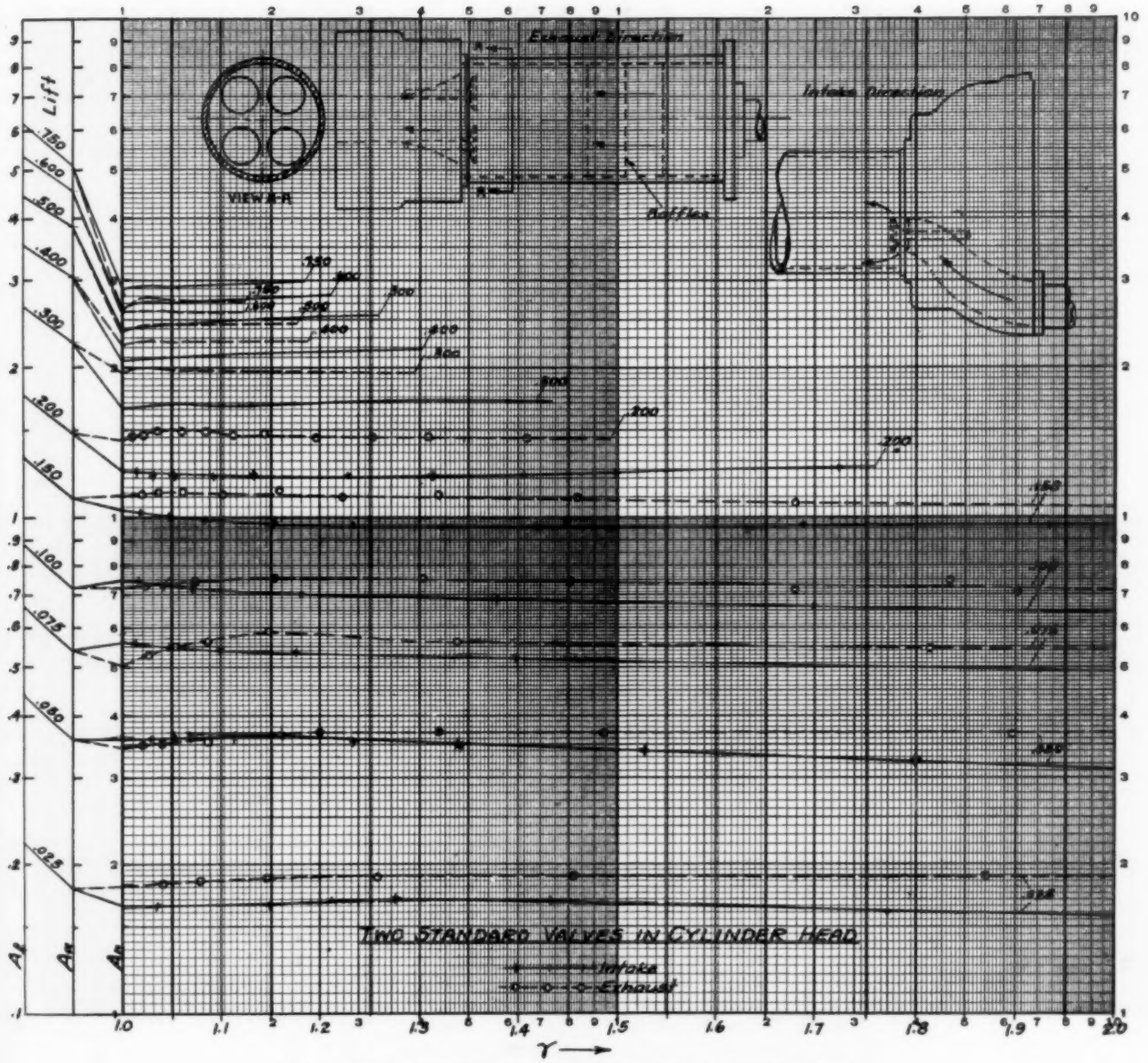


FIG. 7 TEST RESULTS—TWO STANDARD VALVES IN CYLINDER HEAD

tion of flow. In one instance the flow appeared to fluctuate back and forth between the two types in an interval of about 1.5 sec., thus rendering readings impossible.

An extremely loud, high-pitched whistle was associated with the discontinuity above described, though it did not appear in every instance. This whistle occurred only in the intake direction of flow, and was accompanied by only slight pressure fluctuation; often no noticeable fluctuation at all. The whistle occurred only at some pressures in a series with a fixed valve opening, starting and stopping once or twice during the series. Sometimes it would begin and continue for three or four consecutive readings, and then stop with no noticeable change in resistance or flow.

Another type of instability was seen in some cases with small openings. The flow of air instead of being uniform around the periphery of the valve was concentrated at one side. It might also be subject to fluttering, the flow shifting about the sides of the valve several times in a second. The pressure changes associated with this action were slight.

Further mention of these observations will be made in reference to particular tests.

CALCULATION OF RESULTS

The calculation of a test consists essentially of three steps; namely,

1 Computation of air weight from flow readings = G
2 Computation of theoretical discharge through the known area of valve opening under test conditions = (G_{th})
3 Computation of $\mu = G/G_{th}$.

The type of flow formula given by Equation [3] is suited to the flow-meter calculation. For this case the following constants are to be entered:

$$A = \frac{\pi}{4}\frac{(3)^2}{144} = 0.0492 \text{ sq. ft.}$$

$$\mu = 0.985.$$

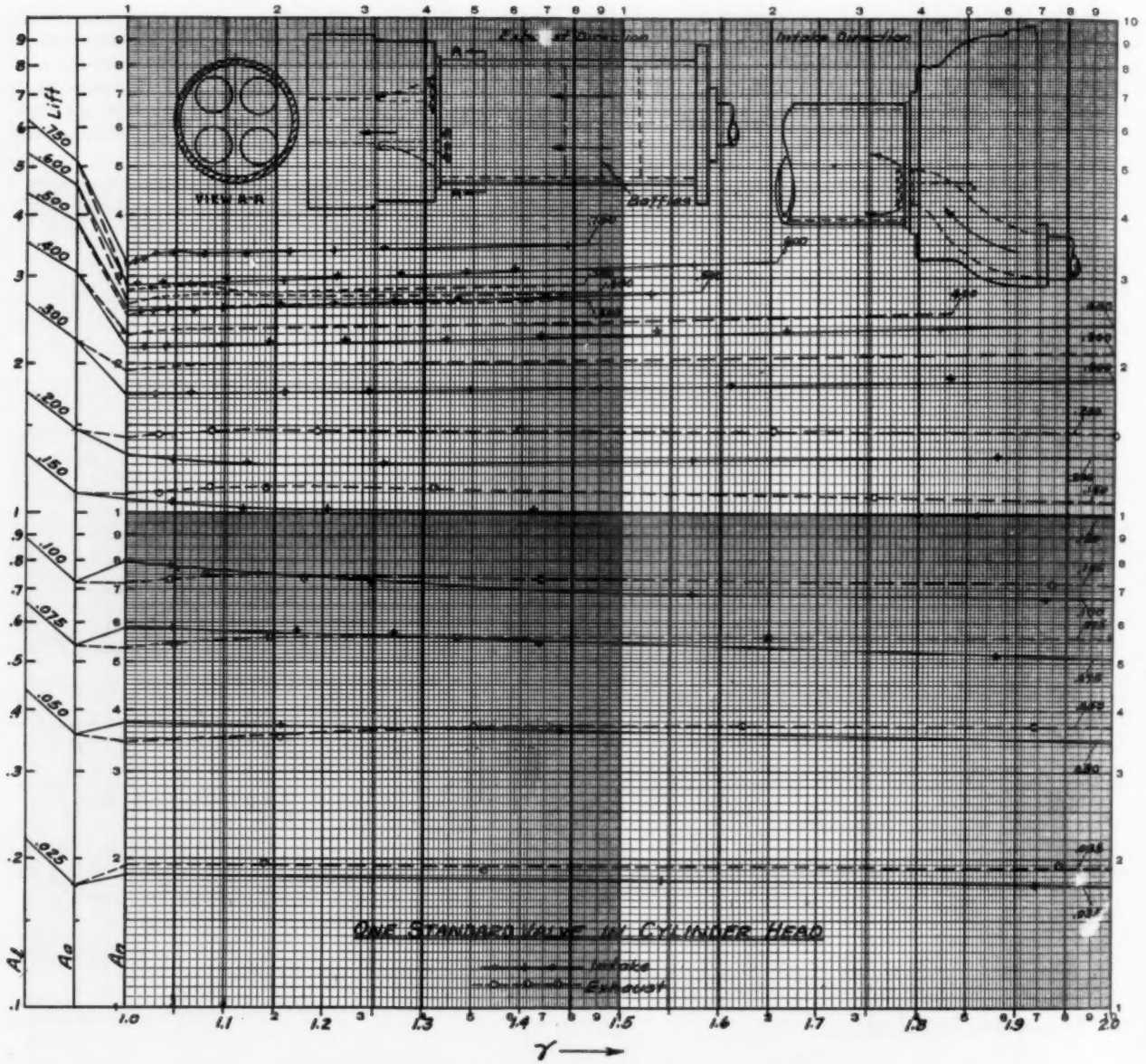


FIG. 6 TEST RESULTS—ONE STANDARD VALVE IN CYLINDER HEAD

finished, a check series was run. In general the agreement was excellent. The procedure was then repeated with an increased valve lift, to a maximum of 1.200 in. In some cases a further test was made with the valve completely removed. At all lifts a wide range of throttle valve openings was employed. The result of this was to extend the pressure ratios of tests with small lifts much beyond the range of ordinary application. Tests with large lifts were restricted by the capacity of the supply-pipe system. Nevertheless, these in all cases cover the generally useful range of pressures.

Readings of barometric pressure and of room temperature were made, in addition to the full series provided for in the test arrangement.

GENERAL OBSERVATIONS ON TESTS

Noteworthy differences were found in the behavior of the special "tulip" and "drop" types of valve on test, as compared with that of the standard valves. These took the form (1) of sharp discontinuities in the flow resistance, with only very slight changes of pressure, and (2) of differences in flow manifested by noise or by instability.

Air flow through standard valves, in both directions, was accompanied by a hissing or rushing sound which changed only in degree as the amount of air flowing and the pressure head increased. No abrupt changes of resistance or of flow characteristics were noticeable at any time.

The special valves in the majority of tests acted similarly, that is, no unusual sound or other behavior was noticeable. However, at points where instability of flow appeared, the other peculiarities in general developed also. The change in resistance was very abrupt and amounted to as much as 4 in. of mercury with an extremely slight difference of flow. The change of resistance was in the direction of a decrease with increasing flow. Hence sometimes actually less pressure drop was required for a large flow than for a smaller one. This effect was most noticeable with intermediate valve openings and in the exhaust direc-

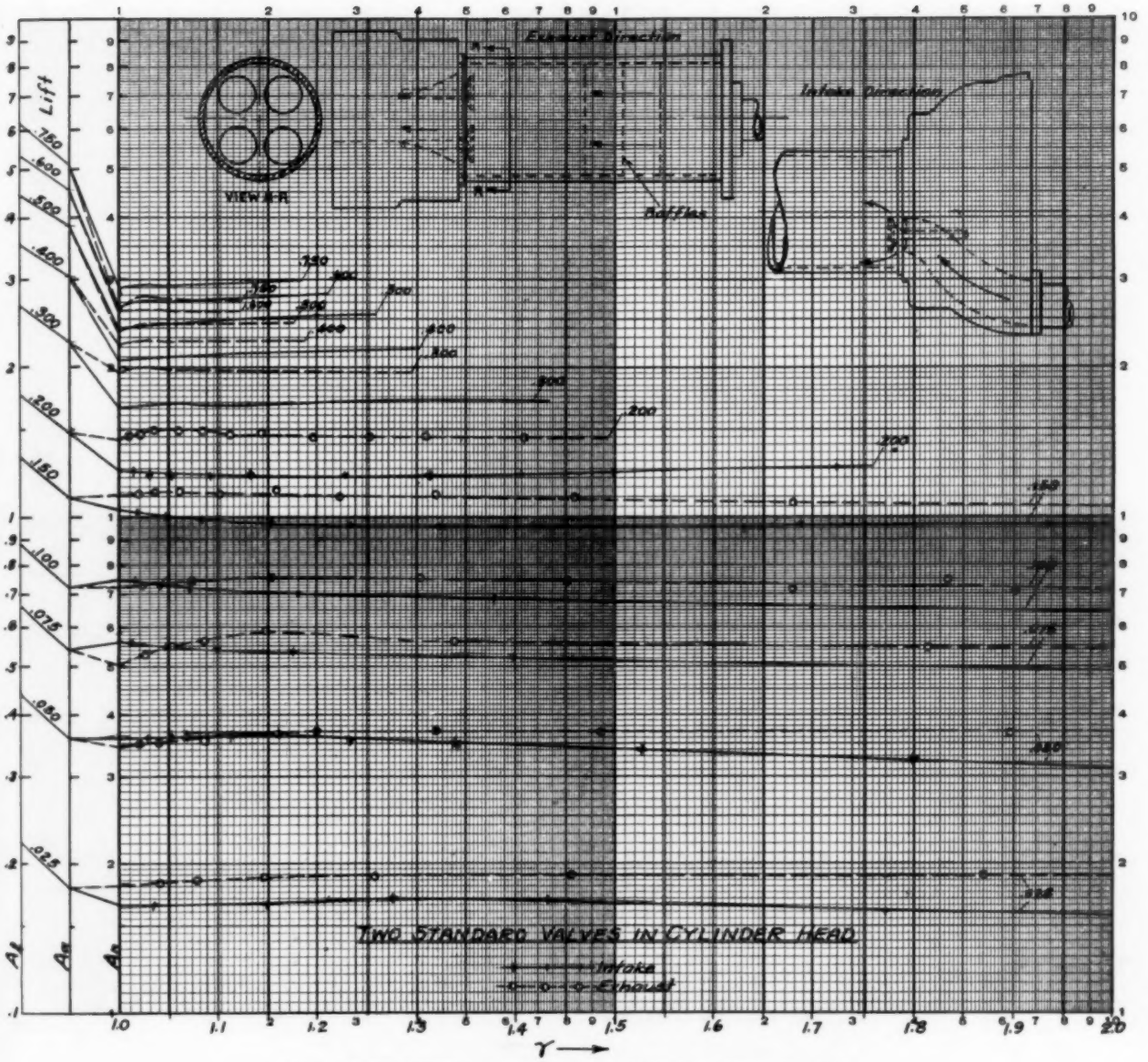


FIG. 7 TEST RESULTS—TWO STANDARD VALVES IN CYLINDER HEAD

tion of flow. In one instance the flow appeared to fluctuate back and forth between the two types in an interval of about 1.5 sec., thus rendering readings impossible.

An extremely loud, high-pitched whistle was associated with the discontinuity above described, though it did not appear in every instance. This whistle occurred only in the intake direction of flow, and was accompanied by only slight pressure fluctuation; often no noticeable fluctuation at all. The whistle occurred only at some pressures in a series with a fixed valve opening, starting and stopping once or twice during the series. Sometimes it would begin and continue for three or four consecutive readings, and then stop with no noticeable change in resistance or flow.

Another type of instability was seen in some cases with small openings. The flow of air instead of being uniform around the periphery of the valve was concentrated at one side. It might also be subject to fluttering, the flow shifting about the sides of the valve several times in a second. The pressure changes associated with this action were slight.

Further mention of these observations will be made in reference to particular tests.

CALCULATION OF RESULTS

The calculation of a test consists essentially of three steps; namely,

1 Computation of air weight from flow readings = G
2 Computation of theoretical discharge through the known area of valve opening under test conditions = (G_{th})
3 Computation of $\mu = G/G_{th}$.

The type of flow formula given by Equation [3] is suited to the flow-meter calculation. For this case the following constants are to be entered:

$$A = \frac{\pi (3)^2}{4\ 144} = 0.0492 \text{ sq. ft.}$$

$$\mu = 0.985.$$

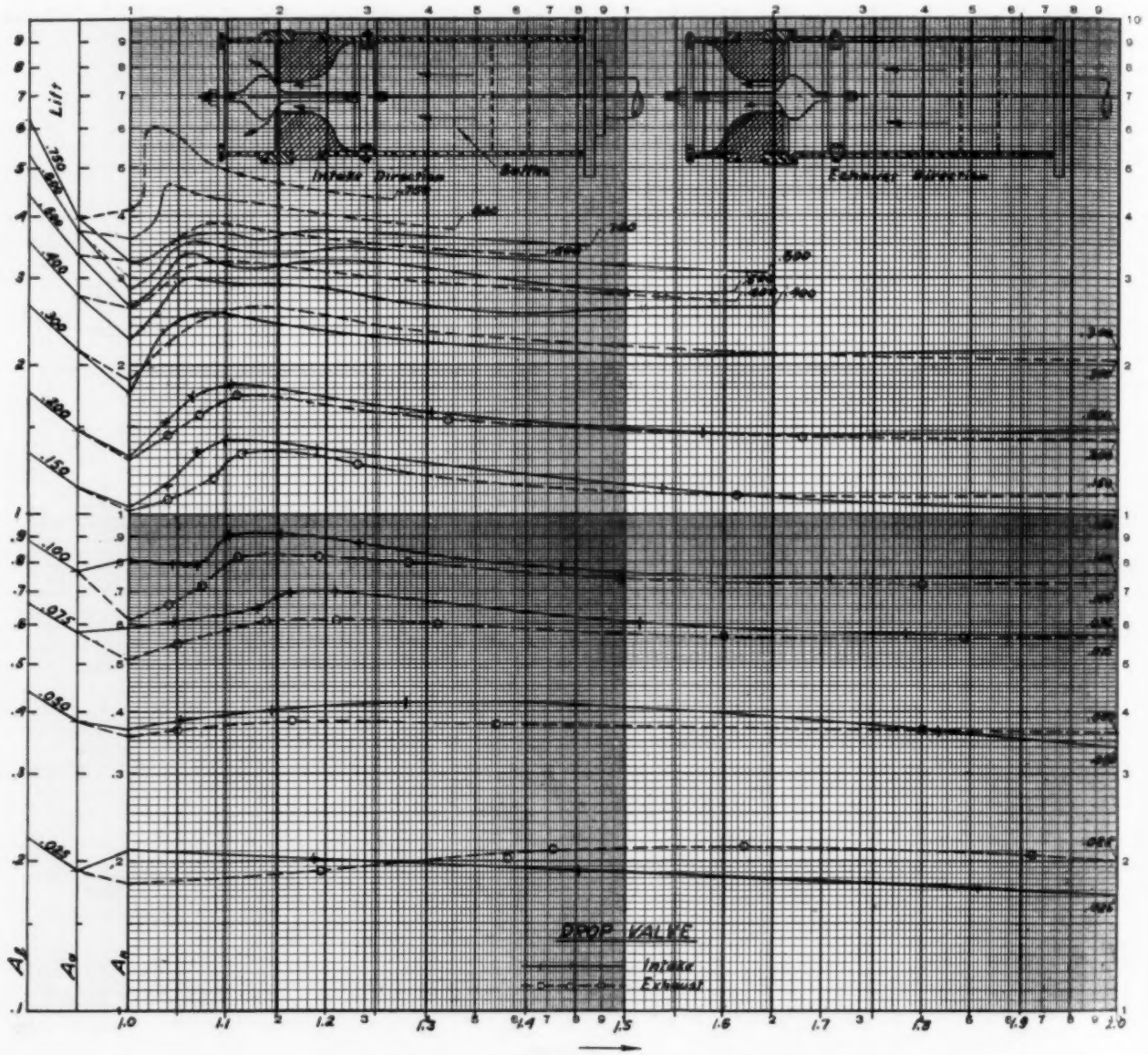


FIG. 8 TEST RESULTS—"DROP" VALVE IN MODEL

Z = correction factor for velocity of approach

$$= \sqrt{1 - \left(\frac{9}{3}\right)^4 \cdot \left(\frac{P_2}{P_1}\right)^{2/K}}$$

= 1.006 approximately.

The flow-meter equation then takes the form:

$$G = 7.65 \sqrt{\frac{\Delta P \cdot P_2}{T_{D1}}} \text{ lb. per sec.} \quad [6]$$

in which ΔP and P_2 are in lb. per sq. in.

Following some early difficulties with moisture, attention was given to the possible effects of humidity. A considerable amount of condensed water having been removed from the air after compression, its humidity was somewhat below normal, averaging less than 50 per cent. Check calculations showed that Moss's equations for "normal air" could be used with a maximum error of about 0.3 per cent, and this was considered satisfactory. Equation [6], based on constants for normal air, was used without further correction.

Computation of theoretical flow through the valve was made, using one or the other of the following equations:

When $P_a > 0.53 P_3$,

$$G_{th} = 5.37 A_a \sqrt{\rho_3 H \left[1 - \frac{3}{2K}\frac{H}{P_3}\right]} \text{ lb. per sec.} \quad [7]$$

When $P_a < 0.53 P_3$,

$$G_{th} = 0.325 A_a \sqrt{P_3 \rho_3} \text{ lb. per sec.} \quad [8]$$

in which:

P_3 = pressure before valve, lb. per sq. in. abs.
ρ_3 = density of air before valve, lb. per cu. ft.
P_a = atmospheric pressure, lb. per sq. in. abs.
$H = P_3 - P_a$ = gage pressure before valve, lb. per sq. in.

Equation [7] is derived from the expanded form of the thermodynamic equation of flow, given for normal air as follows:[3]

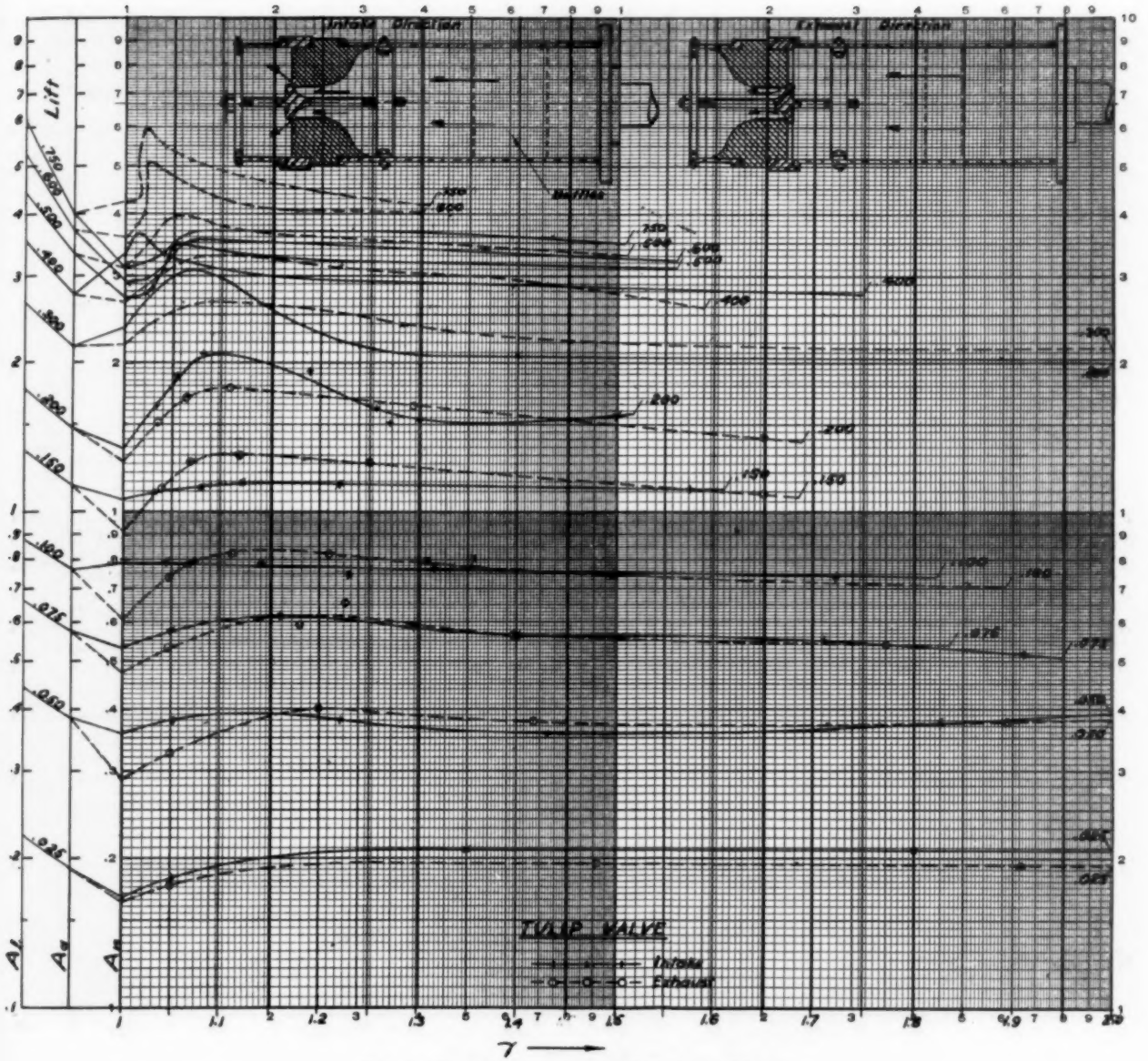


FIG. 9 TEST RESULTS—"TULIP" VALVE IN MODEL

$$G = 158.17A\sqrt{\frac{(P_1 - P_2)[P_2 - 0.0755(P_1 - P_2)]}{T_{D1}}} \quad \ldots [9]$$

while Equation [8], in which flow depends upon initial conditions only, has the form

$$G = 76.367\,\frac{A\mu P_1}{\sqrt{T_{D1}}} \quad \ldots [10]$$

The result of each test is expressible in the term $\mu = G/G\text{th}$. It was found that much labor could be saved by combining Equation [6] with Equations [7] and [8], respectively, in the forms

$$\mu = \frac{G}{G_{th}} = \frac{6.97}{A}\sqrt{\frac{\Delta P}{H\left[1 - \frac{3}{2K}\frac{H}{P_3}\right]}\,\frac{T_{D2}}{T_{D1}}\,\frac{P_2}{P_3}} \quad \ldots [11]$$

when $P_a > 0.53P_3$, and

$$\mu = \frac{G}{G_{th}} = \frac{14.31}{A}\sqrt{\frac{\Delta P}{P_3}\cdot\frac{T_{D2}}{T_{D1}}\cdot\frac{P_2}{P_3}} \quad \ldots [12]$$

when $P_a < 0.53P_3$.

Further simplification was attained by the approximations:

$$\frac{T_{D2}}{T_{D1}} = 1 \text{ (error negligible)}$$

and

$$\frac{P_2}{P_3} = \text{(error in } \mu \leqq 1 \text{ per cent)}$$

The equations then reduce to the forms:

When $P_a > 0.53P_3$

$$\mu = \frac{6.97}{A}\sqrt{\frac{\Delta P}{H\left[1 - \frac{3}{2K}\frac{H}{P_3}\right]}} \quad \ldots [13]$$

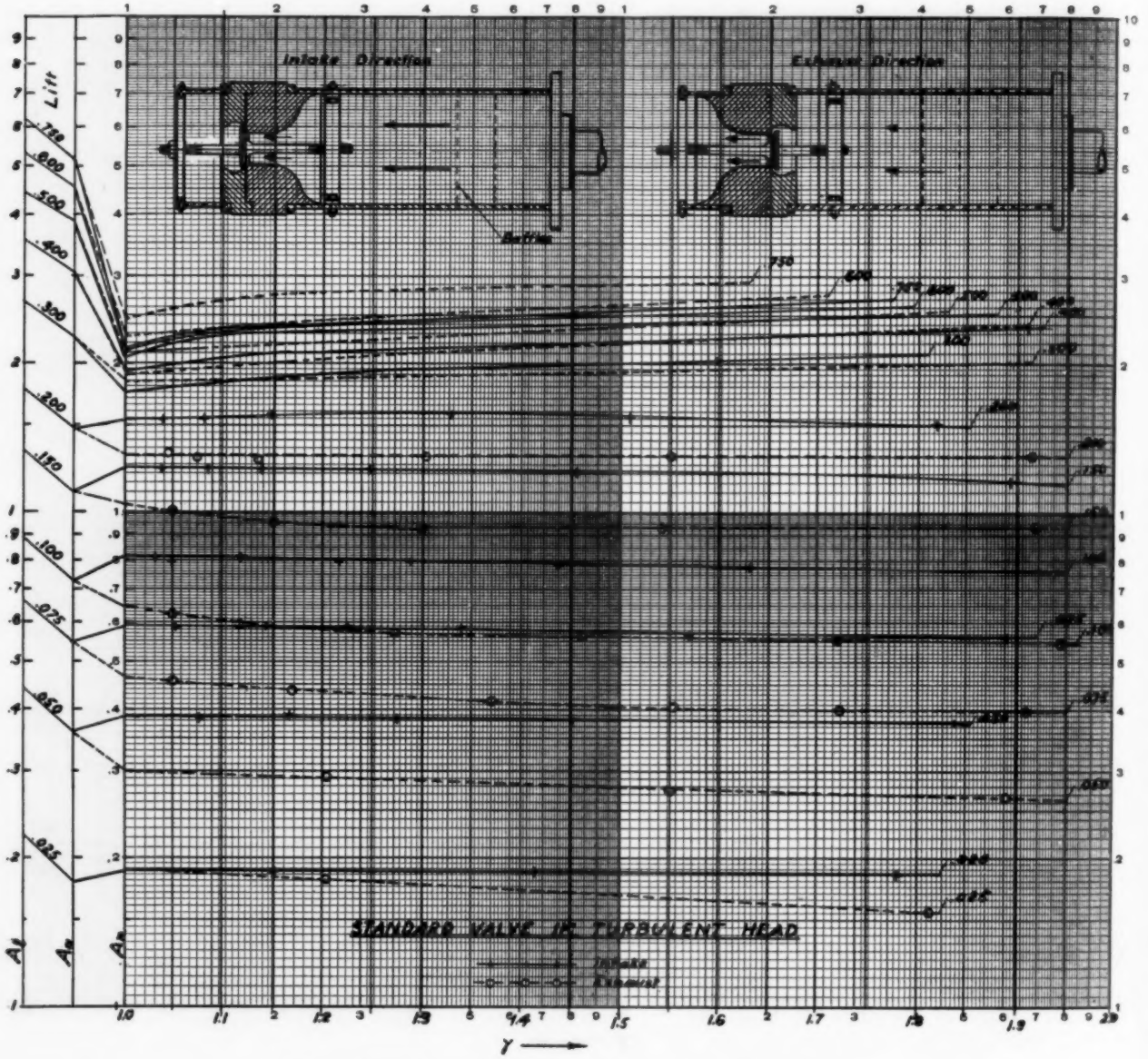


FIG. 10 TEST RESULTS—STANDARD VALVE IN TURBULENT HEAD

when $P_a < 0.53P_1$,

$$\mu = \frac{14.31}{A}\sqrt{\frac{\Delta P}{P_1}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [14]$$

These forms were accepted as being sufficiently close for the purposes of this work.

ESTIMATE OF ERRORS AND PRECISION OF RESULTS

Errors contributing to inaccuracy of the results are of two kinds: those due to approximations in the formulas, and those due to inaccuracies of observations.

Errors in Formulas and Method of Calculation

In this class are to be considered:

1 Flow-meter calculation, Equation [3]. Moss's curves indicate maximum error within the range of application = ±0.2 per cent
2 Neglect of humidity. Check calculations showed maximum error in density about ±0.6 per cent; in μ, ±0.3 per cent
3 Equations [7] and [8]. Negligible errors
4 Neglect of velocity of approach to valve. Calculated 0.35 per cent, error for worst case
5 Use of simplified formulas, Equations [13] and [14]. Maximum about 1.0 per cent.

The total effect of these possible errors in calculation methods is probably not over 1.5 per cent and averages perhaps 1 per cent.

Errors of Observation

1 Setting of Valves. Accuracy of valve lift is affected by the following conditions:
   (*a*) Setting to correct opening
   (*b*) Movement of valve after setting
   (*c*) Eccentricity of valves with respect to seat.
2 Reading gages and manometers.
3 Changing of air pressure while readings were being taken.

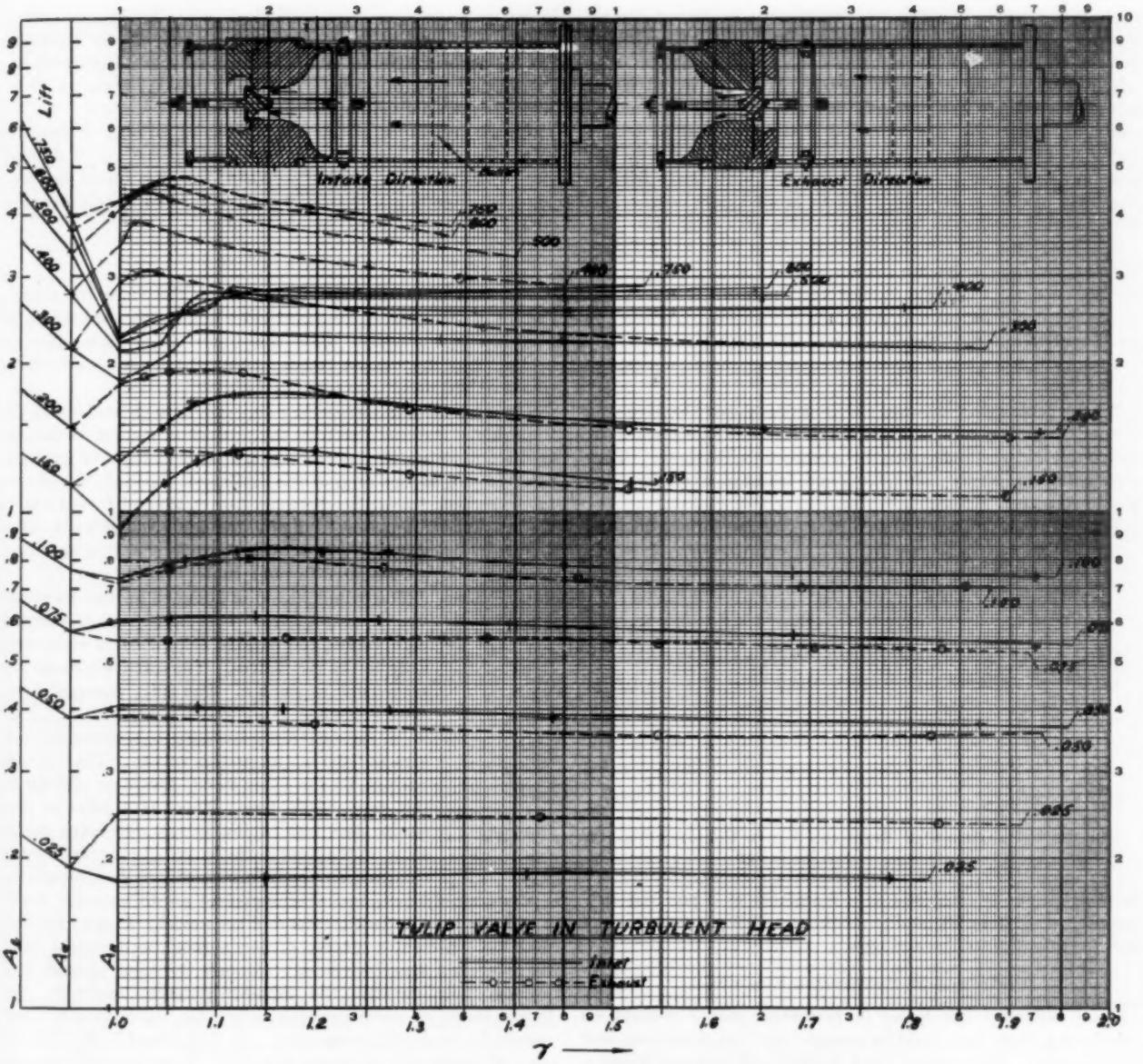


FIG. 11 TEST RESULTS—"TULIP" VALVE IN TURBULENT HEAD

1 *Setting of Valves.* In the test of the valves in the cylinder head the valves could be set accurately with a dial indicator placed against the end of the stem. The settings were checked frequently during each run, as pressure against the valve increased, and any longitudinal movement was corrected by moving the valve back to its correct position by means of a screw against the end of the valve stem. For openings above 0.200 in. the opening was obtained by setting this screw against the stem and giving it the proper number of turns. Then the indicator was placed in position to register any longitudinal movement of the valve. The screws were threaded with 20 threads per inch so that one turn of the screw opened the valve 0.050 in.

The valves were observed to move longitudinally from 0.001 to 0.003 in., but this movement was corrected by resetting the valve each time the pressure was changed. Movement of the valve occurred only at small openings (0.025 to 0.300 in.) where the pressure went up to 20 lb. per sq. in. or more.

Under these conditions, the error was not over 0.001 in. at the time of observation. At an opening of 0.025 in. this error is 4 per cent; at lifts of 0.100 in. and over, 1 per cent or less.

When valves were tested in the wooden models, the valve lift was obtained by giving the valve a certain number of turns, after it had been closed on its seat, to give the desired opening. The valve stems were threaded with 20 threads per inch and screwed into the spider which held the valve in place.

A slight looseness in the fit of the thread in the spider permitted a longitudinal movement of 0.003 to 0.005 in. when pressure came against the valve, that is, at small valve openings where the pressure went up to 25 lb. per sq. in. or more. The movement of the valve in this case could not be measured conveniently by the dial indicator because the airstream came out all around the stem and the indicator would be moved by it.

Inaccuracy of the seat in the wooden model and perhaps a slight eccentricity of the valve with respect to the seat could be a source of error in the valve setting and in affecting the direction of the air stream. The effects would be negligible for large

valve lifts. Some difficulty was experienced in getting the air to flow out uniformly around the entire circumference of the valve at small valve lifts.

2 *Reading Gages and Manometers.* Pressures in all cases, except where there was unsteady flow and fluctuation, could be easily and accurately read by means of the alcohol and mercury manometers. In many cases where the flow was steady the pressures would remain quite constant and were easily read to $^1/_{32}$ in. of alcohol or mercury. Leaks around gage connections were guarded against by frequently testing with soap suds.

3 *Changing of Air Pressure While Readings Were Being Taken.* There were times when the pressures would slowly change and one pressure, say P_1, might change 1 or 2 per cent before P_2 and $(P_1 - P_2)$ could be taken. Attempts were made, however, to get these readings simultaneously or as quickly as possible.

Errors due to incorrect manometer readings should not exceed 1.5 per cent.

Where pressures were read on Bourdon gages the errors in readings may be greater. These gages were not highly sensitive, and only fair readings could be obtained with them. They were calibrated every other day, and readings were corrected when necessary.

Total of Errors. The combined effect of errors in formulas and in observations is estimated at a maximum of 5 per cent to 7 per cent at 0.025 in lift, and a maximum of 2.5 per cent to 3 per cent for lifts of 0.100 in. and over.

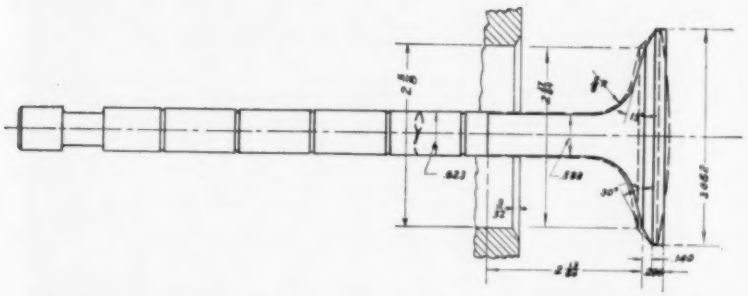


FIG. 12 COMPARISON OF STANDARD VALVE USED FOR TEST WITH $2^3/_8$-IN. S.A.E. VALVE

METHOD OF PRESENTING RESULTS

Some difficulty was found in presenting rather voluminous results in a form that could be compact and yet would not lead to confusion. The method used is not in all respects satisfactory, but it does permit a fair comparison of valve performance to be gained by inspection, and also to enable numerical data for individual cases to be read by simple graphical means. Figs. 5 to 11, inclusive, give the results of tests of various arrangements, the inlet and exhaust directions of flow being shown on the same diagram.

The vertical scale is logarithmic and refers to valve area (sq. in.) in various aspects. The abscissas give the ratio of pressures $\gamma = P_2/P_a$, before and after the valve, and cover values of this ratio between 1 and 2.

Three equal scales of area are given at the left. On the first, the value of A_l is marked for each valve lift tested. The second scale gives the corresponding actual minimum area A_a. These quantities are both established by geometrical relations, and hence are identical for either direction of flow.

Equivalent orifice areas $A_n = \mu A_a$ are plotted on the diagram proper. The variations in μ are reflected in the vertical displacement of individual points, where plotted, and of the curves drawn through them. The latter are in all cases extrapolated to the limit $\gamma = 1$, and tied in by tracers to the A_a-scale. It was found inexpedient to show individual points for lifts exceeding 0.200 in. on account of crowding. Probably without exception the fitting of curves to points is progressively better with increasing lift over 0.200 in. Valve lifts over 0.750 in. ($L/D = 0.266$) are not plotted here.

Certain properties of the diagram are apparent. Where the tracer between A_a- and A_n-scales slopes upward, $\mu > 1$, and vice versa. The value of $\mu = A_n/A_a$ may be found for any point by extending a horizontal line from A_a-scale and stepping off log μ from the figure; similarly for $\Psi = A_n/A_l$.

RESULTS OF TESTS

Fig. 5 shows a test of a standard valve in the concentric model. Details of the so-called standard valve and its port are shown in Fig. 12. For reference only, this latter figure shows also in broken lines the nearest corresponding S.A.E. standard valve. (None of the latter type were used in test.)

This test is characterized by superior flow characteristics in the exhaust direction, and by the absence of sudden variations or discontinuities in the coefficient μ with changes of pressure ratio γ. An abrupt drop of the coefficient, especially for the intake direction, takes place between $L = 0.200$ and $L = 0.300$.

Figs. 6 and 7 represent tests of valves identical with the valve shown in Fig. 5, with the difference that here they are mounted in the cylinder head.

The cylinder head is designed for two inlet and two exhaust valves. One inlet passage supplies both valves, the passage being located symmetrically to the two valves. The exhaust-passage arrangement is similar. The inlet passage, however, is shaped as a diffuser. The area is 7.1 sq. in. at the entrance, expanding to a combined port area of 10.8 sq. in. at the valves. The exhaust passage, on the other hand, contracts slightly in the direction of flow. There is a total of 10.8 sq. in. port area at the valves, decreasing to 9.6 sq. in. at the point of discharge. In the test the discharge at this point was directly to the atmosphere.

Certain differences in the arrangement are to be noted where a cylinder head is tested instead of the model. Firstly, the test does not represent a reversal of flow through an identical path, for inlet flow was tested only through the inlet passage and exhaust flow through the exhaust passage. Secondly, the measurement of static pressure before the valve in inlet flow required a modified arrangement. In Figs. 6 and 7 the entrance pipe at the right is the flow-meter tube. Both tube and passage are of 3 in. diameter. The desired pressure before the valve is total pressure. This is taken as P_1 (pressure before the flow meter); the slight flow resistance in the short 3-in. tube is neglected.

Comparing Figs. 6 and 7 with Fig. 5, we find (1) a slight decrease in efficiency for inlet flow and (2) a large decrease for exhaust flow for valves mounted in the cylinder head. This is shown by the data in Table 2, taken from the diagrams. Like Fig. 5 these diagrams show no sharp discontinuities with pressure changes.

TABLE 2 STANDARD VALVES, IN MODEL AND IN CYLINDER HEAD. VALUES OF A_n

(Valve lift = 0.600 in.; pressure ratio $\gamma = 1.05$)

	Inlet	Exhaust
One valve in model, sq. in.	2.84	3.70
One valve in cylinder head, sq. in.	2.90	2.74
Two valves in cylinder head, sq. in.	2.71	2.74

Figs. 8 and 9 give the results with so-called "drop" and "tulip" types of valve, respectively. These models represent an effort to reduce resistance by giving the valve a favorable shape,

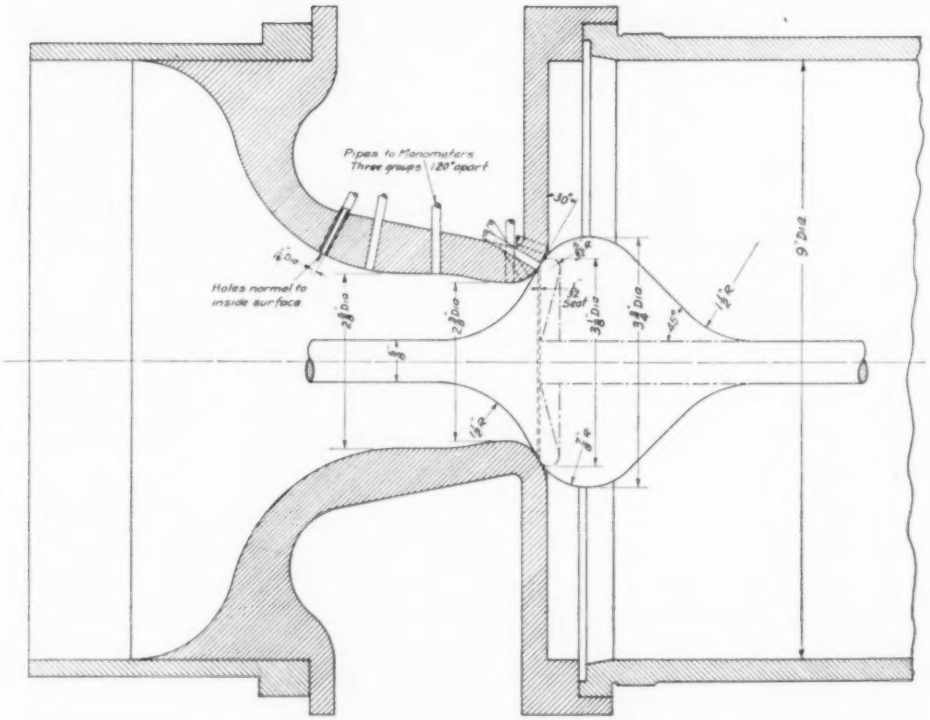


FIG. 13 TEST MOUNTING FOR "DROP" AND "TULIP" VALVES

regardless of the suitability of the valve in practice. For purposes of comparison the valve mean seat diameter D (= 2.82 in.) and the cone seat angle (= 30 deg.) are the same in both as in the standard valve. (See Fig. 12.) In addition to the special valve shape, the valve port and seat are modified also. The effect is to produce an expanding flow area extending from the minimum section in either direction. The increase is of course much less gradual for inlet flow than for exhaust; this seems to be unavoidable. In this respect the "drop" type appears somewhat superior, but the valve of Fig. 9 is a closer approximation to practical needs.

Comparing the test results, both differ substantially from those with the standard valve. We note (1) generally much higher values of A_a. These data show μ as high as 1.60 at the high point and Ψ in several cases above unity, (2) a large variation in the value of μ with change in γ. This variation of μ is accompanied by a discontinuity particularly in exhaust flow at large lifts. The curves of A_a show maxima which occur at decreasing values of γ as the lift L is increased. The general character of the inlet flow curves is similar to the exhaust curves, but the discontinuities are replaced by more gradual changes at all lifts.

The general results in Figs. 8 and 9 are very similar. Only at small lifts the "drop" valve appears superior.

Figs. 10 and 11 are for standard and tulip valves, respectively, mounted in a model cylinder head of a "turbulent" type. The valve pocket forms the combustion space, and opens on the cylinder proper immediately above the top position of the piston. Fig. 14 shows the shape of the pocket as tested with the two types of valve. It is apparent that resistance to flow in and out of the cylinder will be increased by such an arrangement. Furthermore, as the lift increases, a point is reached beyond which the section will decrease, and therefore resistance will become larger.

Inspection of the curves shows that the tulip valve greatly reduces resistance in exhaust flow, but has almost no effect in inlet flow. Inlet resistance is seriously increased as compared with the ordinary mounting without a pocket.

Figs. 15 to 20, inclusive, summarize the data of Figs. 5 to 11, inclusive, in a manner which permits a more direct comparison between various valve performances, and which is suitable for design application. For this purpose it was necessary to make an arbitrary assumption of a representative value for γ, the pressure ratio. This is taken as 1.05 for all these curves, corresponding to a pressure drop in engine use of 0.700 lb. per sq. in. for intake flow, or of 0.735 lb. per sq. in. for exhaust flow. The effect of operation with other values of γ is slight for standard types of valve, but may be considerable for special valves.

These diagrams are non-dimensional, permitting their application to any size of valve. The abscissa is L/D. The ordinate is area in various aspects, referred to $\pi D^2/4$ as unity. The quantities plotted include A_l, A_a, and A_s. To another scale the variation of Ψ is plotted also.

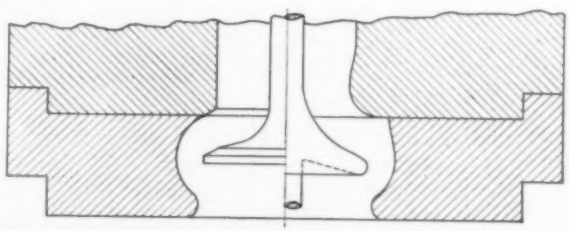


FIG. 14 STANDARD AND "TULIP" VALVES IN TURBULENT HEAD

Fig. 15 compares standard valves in inlet flow tested in the model and in the cylinder head. Fig. 16 makes a similar comparison for exhaust valves. The model test is seen to be a fair guide to performance in actual mounting in the case of the intake valves, but very misleading in the case of the exhaust valves.

Figs. 17 and 18 compare "tulip" and "drop" types for inlet and exhaust flow. A striking feature of Fig. 18 especially is that while the actual section A_a of the special valves is less, the equivalent orifice is considerably greater than for the standard valve.

Figs. 19 and 20 compare results obtained using the turbulent head with the performance of the standard valves in the simple model. By using the tulip type of valve in the turbulent head, good performance is attained in exhaust flow, but intake resistance is comparatively large and not much improved with the special valve.

SPECIAL TEST OF TULIP-TYPE VALVE

After these tests an attempt was made to explore flow conditions in the restricted section of the valve by observing static pressure at various points in the path. Pressure connections were made as indicated on Fig. 13. Care was taken to locate the

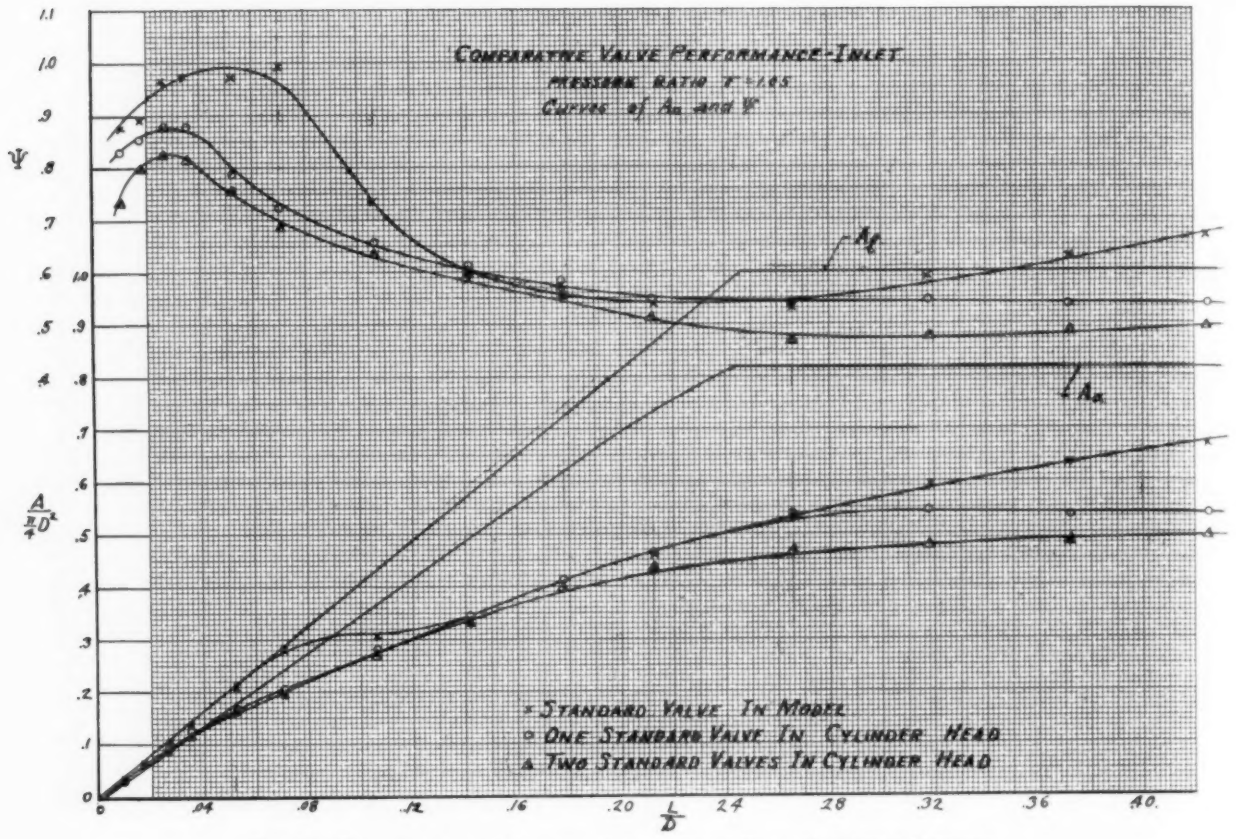


FIG. 15 PERFORMANCE OF STANDARD VALVES, INLET FLOW, VARIOUSLY MOUNTED

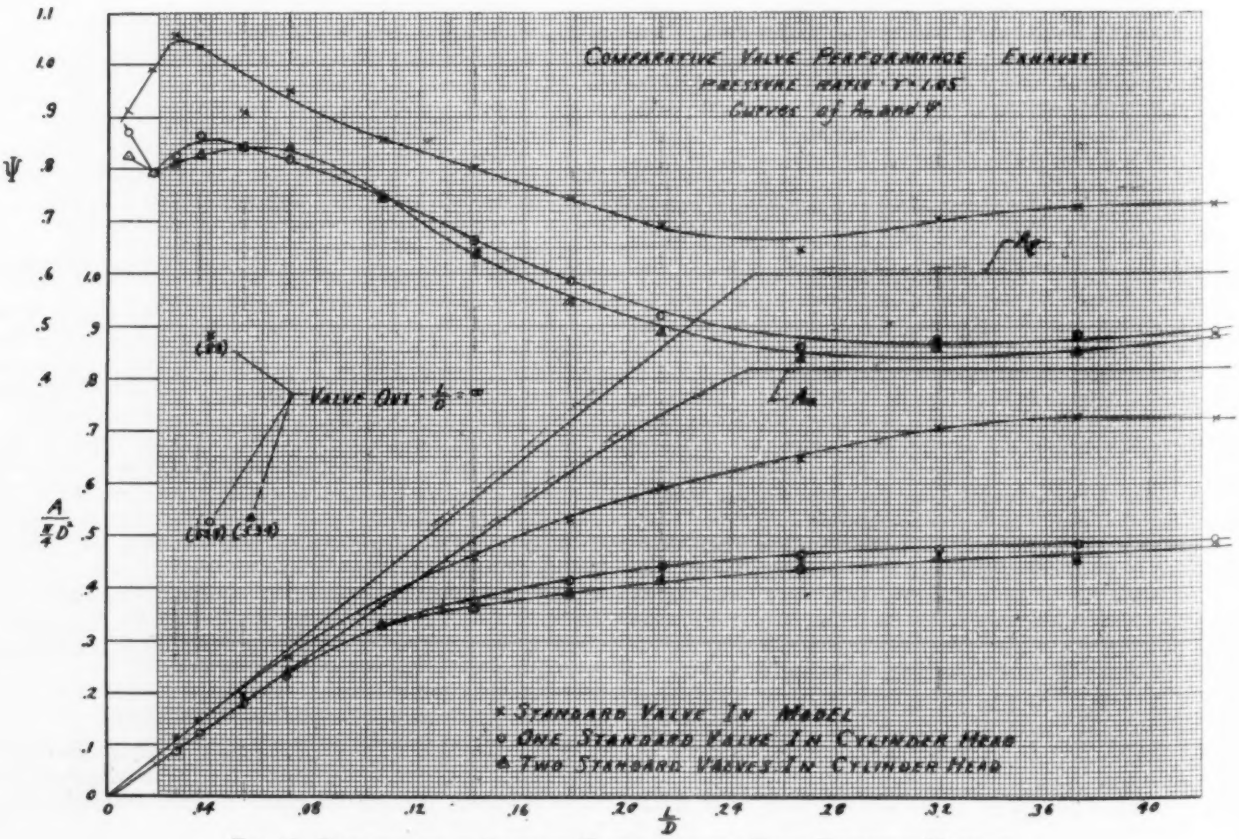


FIG. 16 PERFORMANCE OF STANDARD VALVES, EXHAUST FLOW, VARIOUSLY MOUNTED

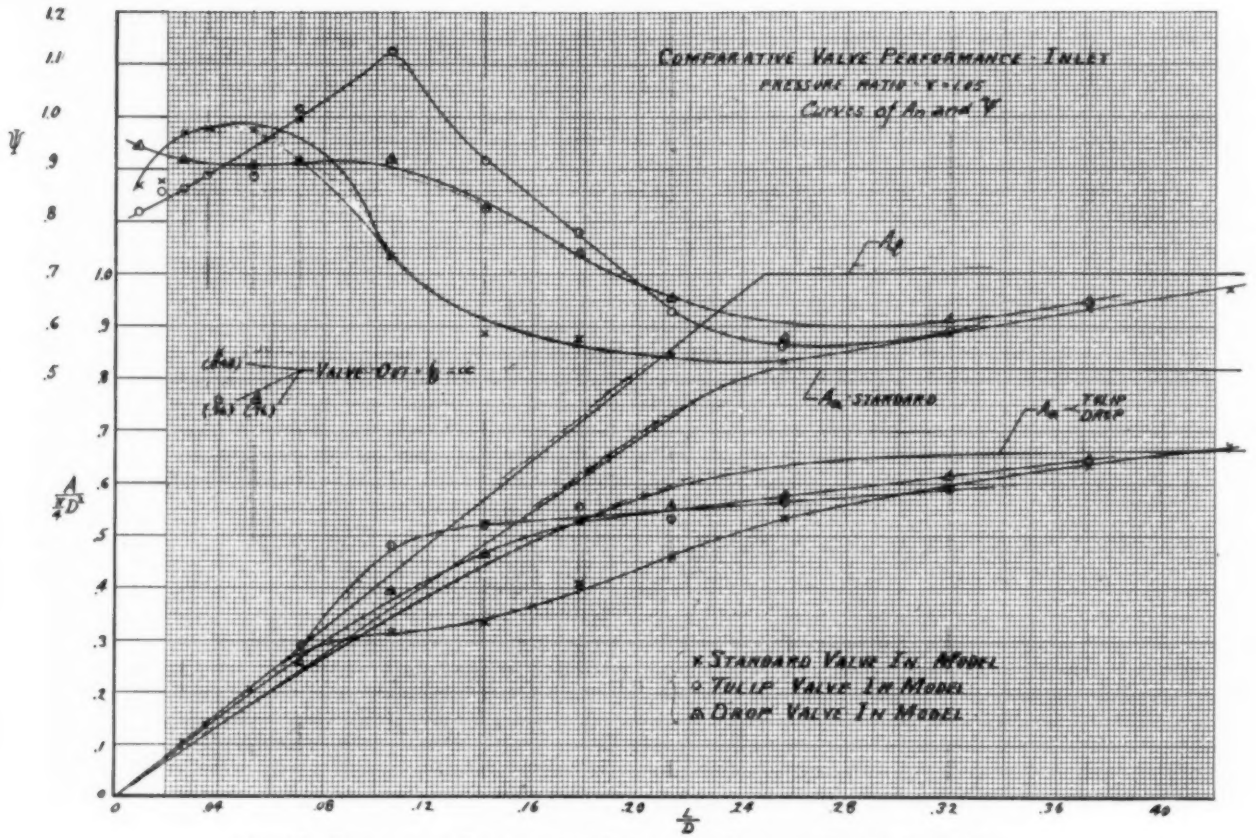


FIG. 17 COMPARATIVE PERFORMANCE, SPECIAL AND STANDARD VALVES, INLET FLOW

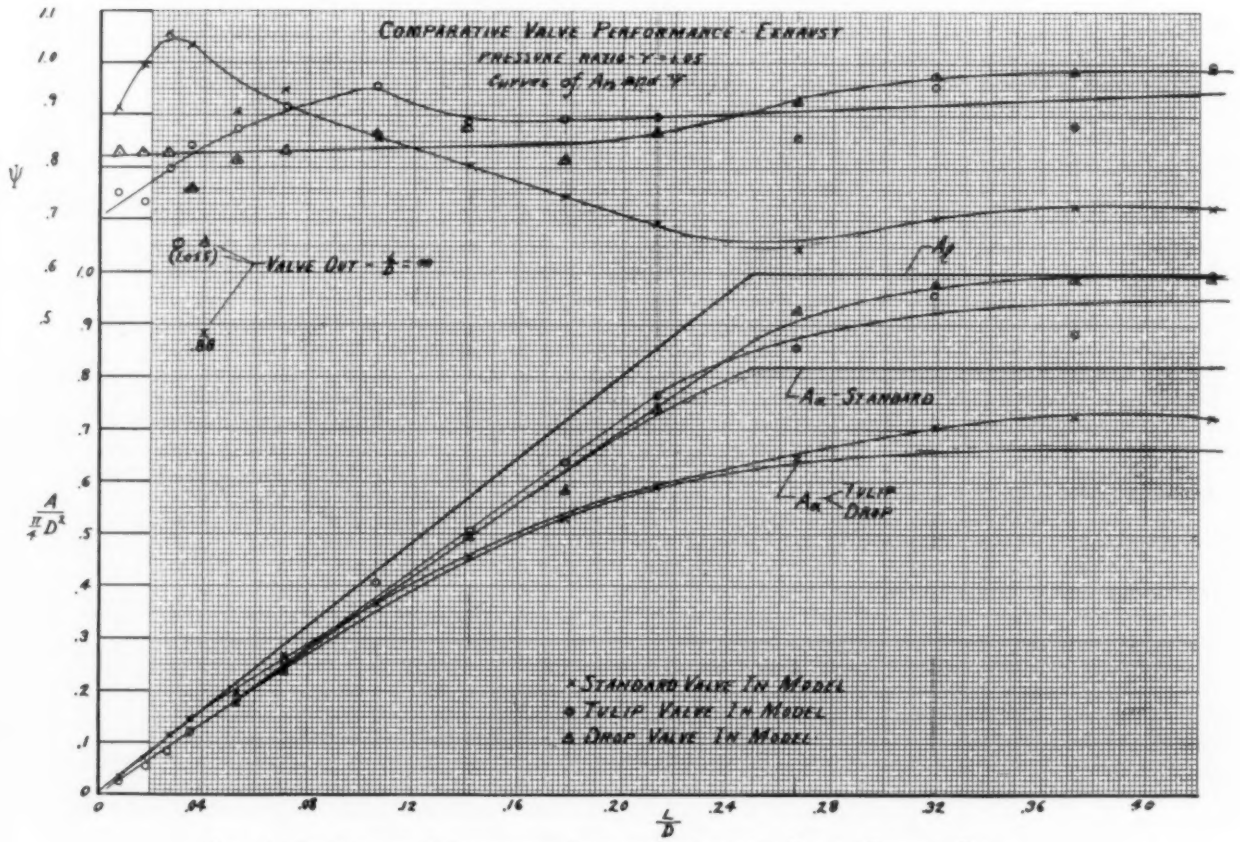


FIG. 18 COMPARATIVE PERFORMANCE, SPECIAL AND STANDARD VALVES, EXHAUST FLOW

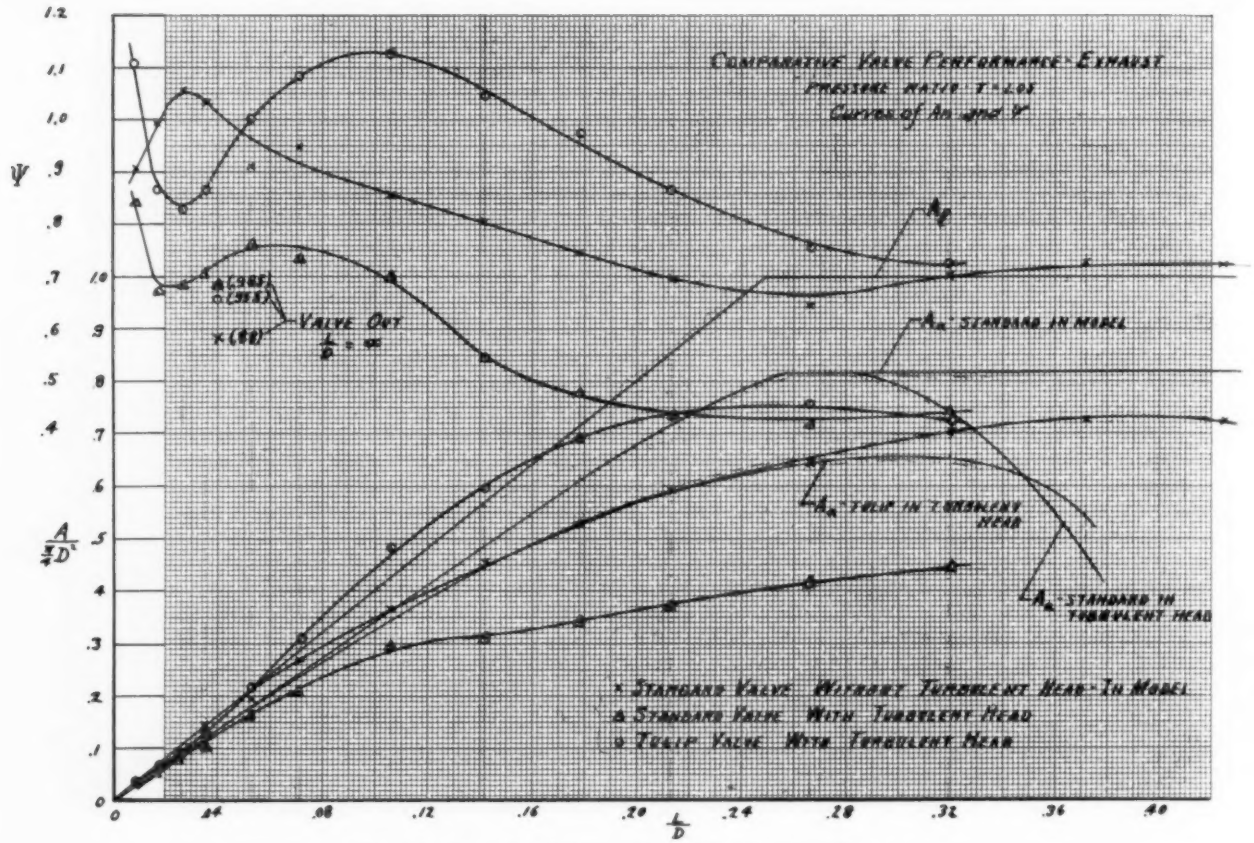


Fig. 19 Valves in "Turbulent" Head, Exhaust Flow

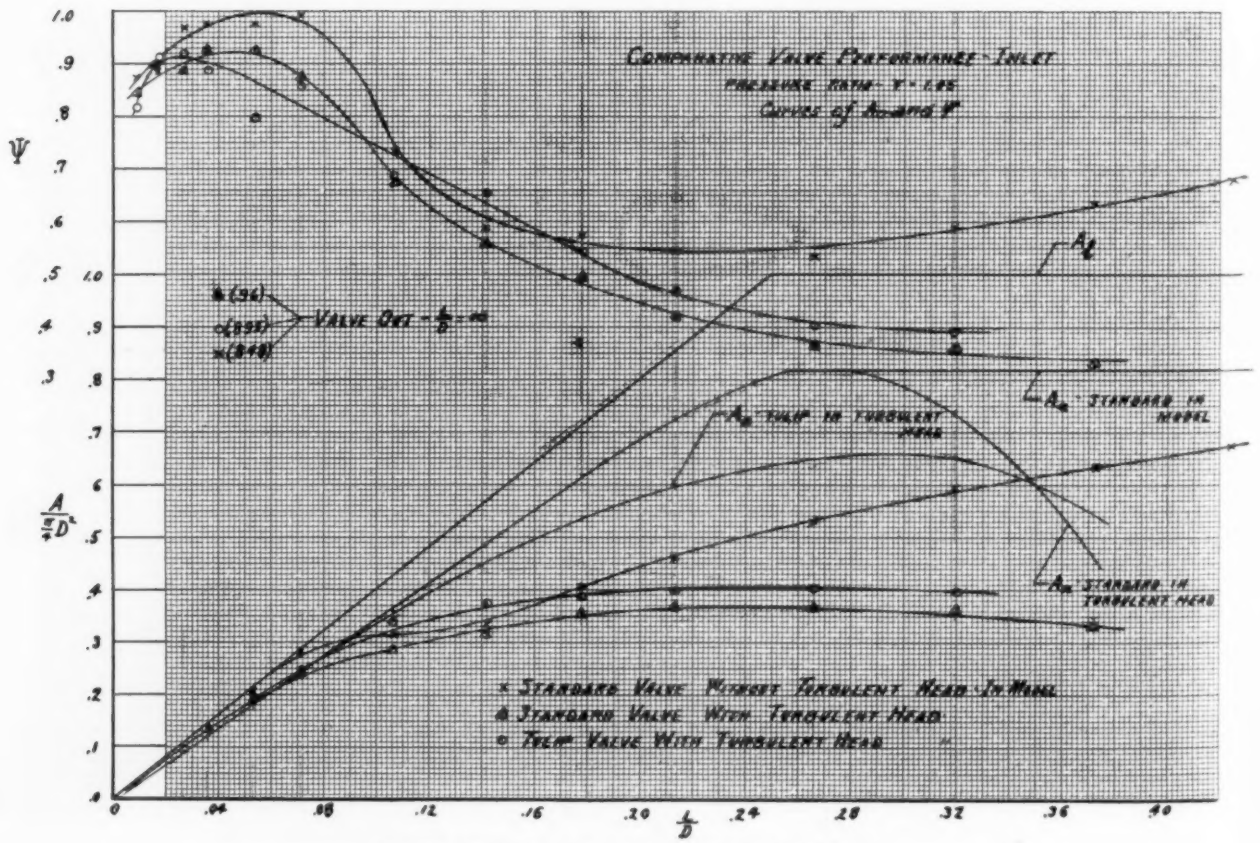


Fig. 20 Valves in "Turbulent" Head, Inlet Flow

holes normally to the surface, and to finish the edges smoothly. The model was then tested under some conditions identical to the original ones. The pressures at various points were observed in addition to other measurements taken as before. The purpose was to find, if possible, the location and cause of losses.

The results were in several respects unsatisfactory. The tests themselves were marred by violent instability when testing flow in the exhaust direction, at lifts of 0.300 in., 0.500 in., and 0.700 in. No readings could be taken with 0.500 in. lift. This model differed from the original in being made of metal whereas the first was of wood; this seems to be the only reason for the change in behavior, which was one of degree and not of kind. Also, extremely loud whistling occurred with inlet flow at several openings.

The connections as made failed to give satisfactory static-pressure readings. This was apparent when the sum of calculated velocity pressure plus observed static pressure in some cases exceeded P_3, the initial pressure.

No positive conclusions of value could be drawn from this test. The data were in some respects contradictory, and were unsatisfactory for analytical purposes. One fact of interest was that in several cases the velocity of the air exceeded that of sound in the expanding section of the path.

Comparison With the Results of Others

Efforts were made to establish a correct basis for comparing the present results with those of other experimenters. The comparison is not in all respects satisfactory because of differences in presentation of results, differences in types of valves and test methods, and lack of detail information, especially in the case of Nutting and Lewis.

Table 3 gives the particulars of four different types of valve and the kind of test arrangement used with each. A brief inspection will show that no two valves and test conditions are identical. The closest similarity is between the authors' standard valves ($\gamma = 1.25$) and Tanaka's 30-deg. valve ($\gamma = 1.245$).

In Fig. 21 the various results are placed upon a common basis. The abscissa is lift/port diameter, a different quantity from L/D as used elsewhere in the figures. The ordinate is coefficient of discharge based upon the area $\pi D_p L$, also a different quantity from that used elsewhere in the work.

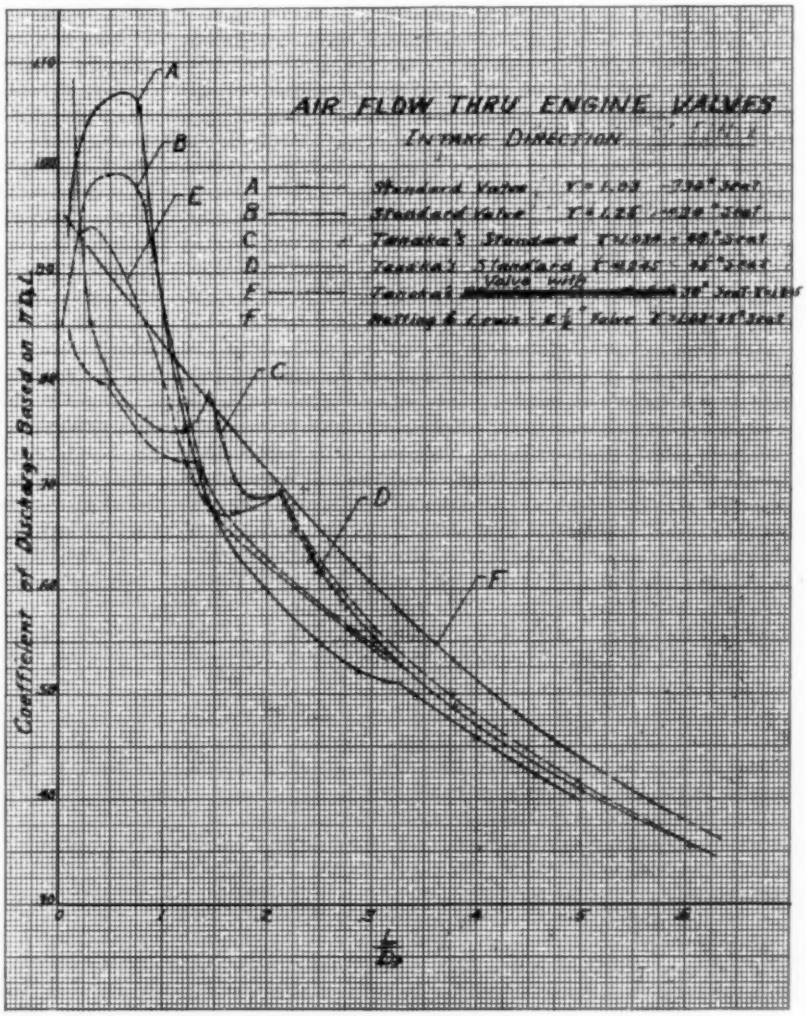


Fig. 21 Comparative Results for Flow Through Inlet Valves

Curves B and E represent results with the two valves above mentioned, which form the best available basis for comparing the present results with those of Tanaka. His results in terms of flow coefficient are available only for his standard 45-deg. valve. The corresponding figures for his 30-deg. valve were arrived at by assuming his W_∞ (flow weight with valve removed) to be the same for either valve, a justifiable assumption.

The agreement of the two curves is good except with small lifts; here precision is in no case so good as with larger lifts. The general shape of the curves is also very similar. In the absence of other opportunities for comparison, this agreement in the two curves is taken as a satisfactory but by no means positive check between the two series of experiments.

Nutting and Lewis' valve was similar to Tanaka's standard; the exact differences are unknown. But their results do not check those of Tanaka, and particularly fail to show the discontinuities which are a feature of Tanaka's curves. Tanaka's results are undoubtedly to be preferred on grounds of superior precision and completeness. There is no direct comparison between the present test and those of Nutting and Lewis.

Tanaka's available results with exhaust valves are quoted upon a relative basis (W/W_∞) which does not lend itself readily to comparison. No common basis for a check of results could be established. In view of this fact, and also because of other criticisms of Tanaka's exhaust-flow experiments, which will be

TABLE 3 COMPARATIVE VALVE DATA FROM INLET-VALVE TESTS

	Tanaka	Tanaka	Nutting and Lewis	Authors
Diameter of port, in.	2.60	2.60	2.50	2.625
Angle of seat, deg.	30	45	(45)[1]	30
Valve-head angle, deg.	0	0	..	15
Type of model	Concentric	Concentric	Cylinder head	Concentric
Approximate pressure ratio	1.245	1.034 & 1.245	1.03 (approx.)	1.03 & 1.25
Number of valves	1	1	2	1
Corners of valves and seat	Sharp	Sharp	(Sharp)[1]	Sharp
Type of flow	Continuous	Continuous	Continuous	Continuous

[1] Estimated from sketch.

discussed later, attempts at a comparison were abandoned. The work of Nutting and Lewis did not include exhaust-flow tests.

Discussion of Results

The value for design and other purposes of experimental results such as the present ones is predicated firstly on the precision of the results in themselves, and secondly on the influence which operating conditions may have upon their interpretation.

In the case of valve tests, conditions of application which must be taken into account include intermittent operation, and particulars of valve mounting and connecting passages. Differences in gas condition, as for hot exhaust gas, are adequately cared for by the proper use of flow equations and gas constants. On the other hand, when a special phenomenon is known to exist, such as the discontinuities found in the tests of special valves, the nature and effects of this condition must be fairly well understood if the data are to be applied with confidence.

Some facts bearing on the mounting of "standard type" valves were demonstrated. For inlet flow, the valve had nearly equal coefficients whether mounted in the model or in the cylinder head. Some loss of efficiency developed where both inlet valves were in use. This is attributed directly to friction loss in the somewhat narrow passage. The effect of inlet-valve mounting with this type of valve may they be disregarded unless the approach passage is unusually restricted in any way. Furthermore, the present tests in model and in cylinder head not only agree reasonably well among themselves, but they serve as a check upon Tanaka's model test of a similar valve. Doubtless an equal valuation is to be placed on his tests with all types. Hence, so far as Tanaka's results for inlet valves may be placed on an absolute basis instead of the inconvenient relative basis which he employs, they should be used with confidence. His results are available for a great variety of detail modifications. A design basis for inlet valves, therefore, is to be found in Figs. 5, 6, 7, 15, and 21 and in the data of Tanaka. An allowance of 5 per cent efficiency loss for cylinder-head mounting as compared with a model test should be sufficient, unless the arrangement is peculiarly unfavorable. The assumption of similar behavior for geometrically similar valves will be slightly too favorable for smaller valves and the reverse for larger valves.

Exhaust-valve conditions are quite different. Fig. 16 shows the remarkable loss of efficiency for valves mounted in the cylinder head as compared with the valve in the model. The reason is apparent in view of the other results found. In reversed flow, the rounded mouthpiece of the model acts as a diffuser, and artificially cuts resistance much below what may be expected where no such provision is specially made. The model as constructed is far from an ideal diffuser for exhaust flow, yet it is probably more favorable in this respect than any practical design could be. Therefore the model test of Fig. 5 (exhaust direction) is thought to be quite misleading. This criticism applies with the same force to all the results of Tanaka on exhaust valves. As his model is sketched, it will act in reverse flow as an even better diffuser than did the present model. Tanaka's relative basis for showing his results (W/W_∞) is even more unsatisfactory for exhaust than for inlet valves, for with valve removed his model is certainly an excellent diffuser. No satisfactory basis for design is therefore to be found in this phase of his work. The data of Figs. 6 and 7 and the curves for "valves in cylinder head," Fig. 16, are more reliable and should be only slightly different if, for example, a 45-deg. valve were substituted; this much may be deduced from the relative values of Tanaka's original curves. But in the application of exhaust-valve data, the possible effect of the exhaust-passage shape must be considered. If this can be given a gradually expanding section, the result may be highly favorable. The effect may be evaluated only by test. In any case the character of the passage has a major effect on exhaust-valve operation, but only a minor effect on the operation of inlet valves.

In considering the effect of intermittent action as in engine operation, the length of intake and exhaust passages is important. If intake and outlet were direct to the atmosphere without intervening columns of air or gas, flow during intermittent operation should differ very little at any instant from steady flow at an equal lift. The air or gas is quiet up to the point of restriction at the valve. There is little or no mass of fluid to be accelerated, and the flow may assume any rate instantaneously. Where long passages lead to the valves, the column of gas in the passage is alternately accelerated and retarded, and the effect on performance may be considerable. Of such inertia effects it is known that the net result may be favorable, as, for example, in the "ramming" effect of a long intake pipe. No means is here offered for evaluating such effects; any proposed method should be checked against suction indicator cards, and data for such a comparison are not now at hand. Until new data bearing on the question may become available, it is recommended that flow during the opening and closing of the valve be treated as similar to continuous flow at the same point of lift. Where passages are long the criterion is unreliable.

The two special types of valve gave similar results; we shall refer to the "tulip" type which approaches a practicable shape. For intake flow this type showed a good improvement over the standard valve, amounting to between 10 and 50 per cent at operating lifts. This is probably near the limit of what can be made good in practice, for the limitations to the diffuser principle in inlet flow are obvious.

In exhaust flow the gains are greater and more consistent. Also the possibilities of a further development may be considered more promising, for there is less to limit the shaping of the opening past the exhaust valve and port into a good expanding section.

Discontinuities of the orifice coefficient as the pressure ratio increases are a disturbing factor in interpreting and applying data, as are also such instabilities as resulted in loud whistling with some flow conditions. There can, therefore, be less certainty in using for design the data which were obtained than was the case with standard valves. To these considerations must be added the fact that modification to the "tulip" valve would be required to make it practical, resulting in some change of behavior.

The peculiarities of the tulip and drop valves remain for the present without full explanation. The sudden change of μ with increasing flow is doubtless due to a shift in the type of flow, particularly a change of the point at which the high-velocity airstream leaves the wall surface. For example, considering Fig. 13 with flow in the exhaust direction, the lower μ may obtain when diffuser action ends at the termination of the $2^3/_8$ in. diameter, and the higher μ when it continues to the end of the $2^5/_8$ in. diameter. If so, the discontinuity would be removed if a cone of proper angle were substituted for the irregular curve. Also higher efficiency should develop if the diffusers were made longer. Photographic study of the flow, as used by Tanaka, is the best attack on this problem.

Of the "tulip"-type valve with specially formed port, it can be said in general that it is promising for either inlet or exhaust application. The prospective gain over the conventional valve is so large that a valuable reduction in size of valves and gear may be anticipated, or in some cases the use of one valve instead of two. It is not thought that the present data are sufficient for design. Continuation of development is needed both in the valve itself and in experimental engine application.

The diffuser principle has had two forms of application in the experiments: (1) Locally at the valve head, over a path of perhaps 2 in. (2) In the connecting passage, say over a path of 6 to 12 in. above the valve. The former was effective in the tests of the "tulip" and "drop" valves, and showed its value for both inlet and exhaust flow. The second was tried in the inlet-valve test of the cylinder head, where the passage expanded from the cylinder-head entrance to the port openings. Here it failed to show any advantages. In exhaust flow, on the other hand, the model as before stated acted as a diffuser and caused a major improvement in valve performance. Since both types of diffuser are applicable to exhaust flow, they should preferably be combined; that is to say, the entire exhaust passage from the restriction under the seat to the point of release into a receiver or to atmosphere should be a continually and gradually expanding conduit.

The "turbulent-head" tests are of less general interest. It was shown that with such a special head decrease in flow resistance is more readily accomplished for the outlet direction than for inlet.

The special valves tested for inlet flow behaved similarly to a kind of resonator described by J. Hartmann.[9] He placed cup-like obstructions in the path of an air jet discharged from a rounded orifice into the atmosphere. His reservoir pressure is above critical. He states that this device will convert as much as 10 to 20 per cent of the energy of adiabatic compression into sound-wave energy. He thus produced as much as 74 watts of sound-wave power when using an orifice 1 mm. (0.040 in.) in diameter. He was able to produce waves in both audible and inaudible ranges. In the present case velocities over that of sound can be accounted for only by expanding air path. Assuming otherwise that phenomena are similar, this accounts for the whistling and for the low-frequency oscillations and instabilities. The conversion of so much of the energy to sound incidentally means considerable increase of apparent flow resistance.

[9] J. Hartmann, "On the Production of Acoustic Waves by an Air Jet of a Velocity Exceeding That of Sound," *Phil. Mag.*, London, no. 72, April, 1931.

Conclusions

1 Inlet valves of conventional types perform in engine assembly approximately as they do in model tests. The results of Tanaka and those of the present experiments are suitable for design purposes.

2 Exhaust-valve tests in models are misleading unless the possible extent of diffuser action is known and understood. Such action should be eliminated or should be made to correspond to actual engine conditions. Model tests of exhaust valves by Tanaka and by the authors have given results unsuitable for design purposes. The present results with exhaust valves mounted in the cylinder head are to be preferred as design data.

3 Improvement in valves may be effected by designing the flow path as a diffuser. This may be done locally at the valve head and seat, or in the passage above the valve, or in both ways. The former method is effective both for inlet and exhaust; the latter for exhaust only. Preferably both methods should be combined in the exhaust valve and passage design.

In closing, the authors wish to acknowledge their indebtedness to Dr. O. Tietjens for helpful comments and suggestions, and to Messrs. J. B. Ames and R. L. Taylor for valuable assistance in the experiments and in the labor of calculation.

Development of the Double-Acting Engine

By LOUIS R. FORD,[1] NEW YORK, N. Y.

In less than a decade there has been development of the Diesel double-acting engine into a reliable prime mover. Double-acting engines for large power-plant work are attracting attention. A committee studying types of equipment for the extension of the Copenhagen electric-power station has recommended acceptance of a Burmeister & Wain double-acting, two-cycle engine of 18,500 b.hp., with continuous overload capacity of 22,500 b.hp. At present the two-cycle type has forged ahead of the four-cycle design. The steps in the development of the double-acting engine are reviewed by the author.



THE development of the Diesel engine as a type has been so swift that at least an annual survey is required if one wishes to keep fairly conversant with current design practice. The particular phase of this development that has moved most swiftly in the last few years has been the transformation of the double-acting engine from a laboratory mechanism into a reliable prime mover, available commercially in a number of different makes. Although attempts were made 18 years ago to adapt the Diesel engine to double-acting operation, little progress resulted from these attempts, and the double-acting Diesel engine of today may be said to represent the result of serious and active development work of less than a decade.

Several factors acted to retard earlier development, one of the most important ones being the general disruption of engineering and industrial plans by the World War. But even if this interruption of orderly progress had not occurred, it is doubtful that this early work would have put us any further ahead with double-acting engine construction than we are today. In the first place the practical realization of the Diesel principle, as represented by the single-acting engine, had not proceeded far enough 18 years ago for us to be ready for the double-acting type. Also, the necessary stimulus was lacking; there was not enough demand for engine power high enough to be beyond the limits of the single-acting type

At that time the field of the Diesel engine was considered to be a definitely limited one, but as more was learned about injection, combustion, cooling, and other problems involved in Diesel-engine construction and operation, and better materials and mechanical processes became available, the boundaries of the field constantly receded. In the marine field the very successful application of Diesel-engine drive to larger and larger vessels brought the single-acting engine up to what appears to be limiting cylinder diameter, and the demand for higher power, coupled with the space limitations inherent in marine construction, supplied the stimulus required for active development of the double-acting engine.

It is in the marine field, then, that we must look to get a picture of this development which has now reached the point where double-acting engines for large power-plant work are attracting wide attention. As an indication of the trend in this direction, it is interesting to note that at the time this is written a committee appointed to study the types of equipment offered for the extension of the electric-power station at Copenhagen, Denmark, has recommended acceptance of a Burmeister & Wain double-acting, two-cycle engine of 18,500 b.hp., with continuous overload capacity of 22,500 b.hp.

The beginning of the period of active development referred to was marked by the start at about the same time of design studies for both the two-cycle and four-cycle types, but the latter was put far in the lead commercially by the great prestige of the famous firm by which it was sponsored, as well as by that firm's superior facilities for promoting it. The four-cycle type was heavily handicapped, however, by the mechanical complication inherent in the four-cycle principle of operation, a complication far more pronounced when we try to use both ends of the cylinder than when combustion is confined to the upper end. The elimination of much of this mechanical complication, made possible by the use of the two-cycle principle, is responsible for a steady trend toward the two-cycle type. At the present time it has forged ahead of the four-cycle in number of engines under construction. An indication of this trend is found in the fact that during the last year the firm that has been almost fanatical in its adherence to the four-cycle principle has bowed to the inevitable and placed a two-cycle design on the market. Discounting the risk incurred by all would-be prophets of being confounded by unforeseen conditions that may develop in the future, it is safe to predict that the four-cycle, double-acting engine of large size will pass out of the picture before very long.

Although it is of course expected that continued improvement of the double-acting engine may bring about great changes in construction, the first stage of its development, from an experiment to a practicable machine of proved performance, may be considered as completed, and it is of interest to review some of the steps in that development. Without considering the non-engineering factors that have a great influence on the commercial development of any machine, it is of interest to the engineer to view the mechanical development by observing the structural elements that prove so troublesome in operation as to require corrective redesign. The author is in a position to make such observations by reason of having had, throughout this development period, to make maintenance repairs on a number of large double-acting engines of different makes. It should be remarked at this point that the author does not pretend to have a knowledge of the inside history of the design work of any engine builder. The remarks that follow are based on observation of various en-

[1] Consulting Engineer, Physics Building, Columbia University. Mem. A.S.M.E. Mr. Ford, upon completing his apprenticeship in the machinist trade, entered the United States Navy as a machinist, second class, passed through the various enlisted grades to warrant rank, and in 1912 passed his examination for ensign. He was commissioned with the Naval Academy class of that year, and after a year's service at sea was assigned to take a post-graduate course in engineering at the Naval Academy and Columbia University, receiving his M.A. from the latter in 1915. He served throughout the World War in the submarine force, and in 1919, after 17 years' service in the Navy, resigned as Lieutenant Commander to become associated with the Worthington Pump & Machinery Corporation as chief operating engineer. In 1922 he resigned from this position to take charge of the Diesel engineering activities of the Morse Dry Dock & Repair Company, now a unit of the United Drydocks, Inc., and to establish a general consulting practice.

Presented at the National Oil and Gas Power Meeting, University of Wisconsin, Madison, Wis., June 23–26, 1931, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

gines in service, repairs and changes made on these engines, and some slight contact with the engineering analyses of problems that have arisen in operation.

In the preliminary stages of double-acting engine design it was assumed that the chief difficulties would be to obtain good combustion in the lower end of the cylinder and prevent leakage around a piston rod operating in an atmosphere of flaming gas at high pressure. This despite the fact that for many years double-acting gas engines had operated without any particular difficulties of this sort. As a matter of fact, these features prove to be the source of practically no trouble, but other troubles developed at points where they were least expected. In the Burmeister & Wain engine—the first design to pass successfully the tests of commercial use—the anticipated piston-rod trouble was guarded against by removing the rod from the flaming gas. This was accomplished by enclosing the rod in a tubular sleeve, extending from the under side of the piston to the crosshead, the internal diameter of the sleeve being large enough to provide an

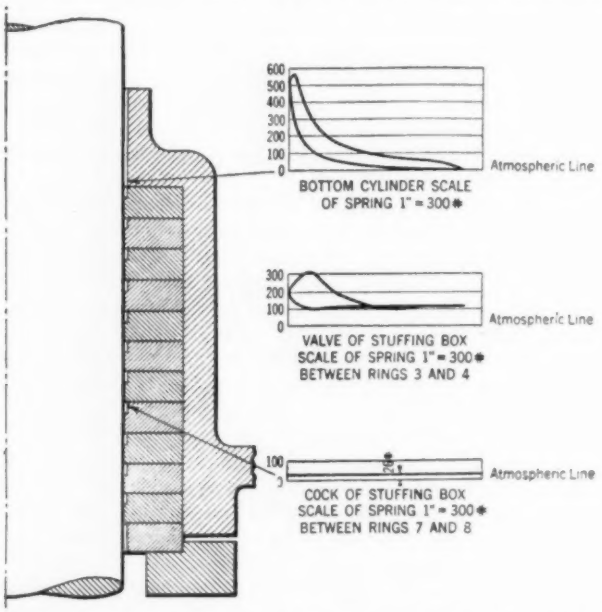


FIG. 1 PISTON-ROD PACKING, WORTHINGTON ENGINE, SHOWING PRESSURE AT DIFFERENT POINTS

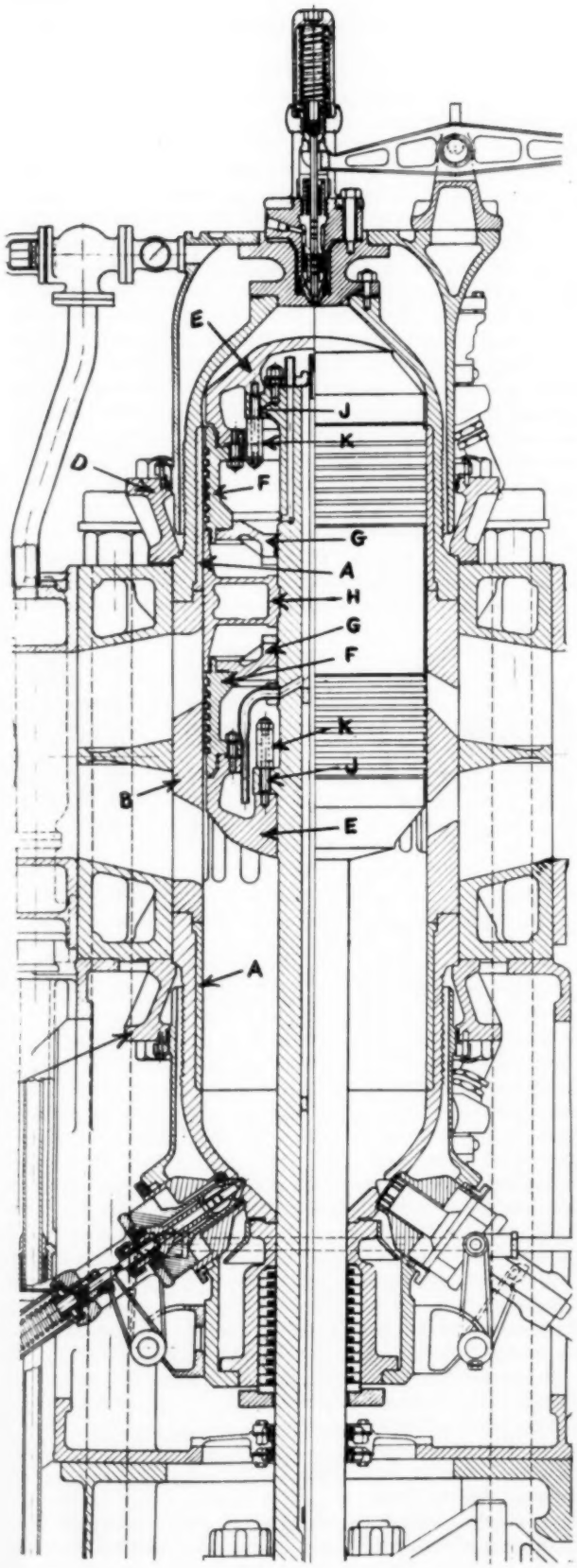
annular space between rod and sleeve. This annular space was utilized as a passage for the piston-cooling oil. Where the piston-rod sleeve passed through the cylinder head a stuffing box was provided, with cast-iron snap rings, just like ordinary piston rings, except that the spring was inward instead of outward. This produced a condition almost similar to that found within the cylinders of all engines, the stuffing box with its packing rings representing the ordinary piston and piston rings, and the surface of the piston-rod sleeve representing the cylinder liner. This device proved highly successful in service and has been incorporated in this firm's newest two-cycle engine.

That this arrangement, although successful, was not needed is indicated by the success of other builders in using plain cast-iron rings in the stuffing box, bearing directly on the piston rod, without any protecting sleeve. As a builder of successful double-acting gas engines in which such packing was used, it was logical that Worthington should use the plain-ring stuffing box in its double-acting Diesel engine. When the first Worthington engine was tested, indicator diagrams were taken to determine the pressure at different points in the stuffing box. Fig. 1, taken from the Worthington catalog, shows that the pressure dropped from 570 lb. at the first ring to 28 lb. at the eighth. The absence of any leakage from the gland indicates that a further drop to zero pressure occurs somewhere before the last ring. These rings are of a good grade of cast iron, made in sections, and held against the rod by simple garter springs. Almost exactly the same type of packing is used in the Deutsche-Werke engine. The author had occasion to remove the packing from two Worthington engines and two Deutsche-Werke engines after one year's service and six months' service, respectively, and all of the rings and piston rods were found to be in excellent condition.

This same principle has been followed by practically all engine builders except Burmeister & Wain, but in some cases the mechanical arrangement is rather elaborate. An example of this is the packing used in the A.E.G. engine, which consists of a packing sleeve, or ring carrier, built up by stacking on top of one another a series of solid rings shaped like the follower, or bull ring, of the steam-engine piston, and placing packing rings in the slots formed between these solid rings. There are two packing rings in each slot, and these are pressed against the rod by a spring ring behind each pair. The whole assembly, consisting of packing rings, spring rings, and solid rings, is held together by bolts and slipped into a stuffing box in the cylinder head. In general it may be said that experience with the engines in service has shown that the piston-rod packing may be very simple in design if the shopwork is sufficiently good. The bearing surfaces must be carefully finished, and the fitting must be very accurate.

The other anticipated trouble—obtaining good combustion in the lower end of the cylinder—did not develop to the extent expected. All of the mechanism related to fuel injection and combustion in the upper end of the cylinder presents no problems not common to both single-acting and double-acting engines. Development of the mechanism of injection and combustion is, then, in so far as it relates solely to the double-acting engine, confined to the lower end of the cylinder. In both the four-cycle and two-cycle types the first problem is to obtain good distribution of fuel in the air in a combustion chamber obstructed by a large piston rod. Since this rod, with its stuffing box, uses up a large amount of the space available in the cylinder head, a second problem is introduced—that of finding a place for the fuel valves without unduly complicating the head casting and without too much interference with circulation of cooling water. In the four-cycle type a further complication results from the necessity for inlet and exhaust valves.

A very practical solution of these problems was worked out by Burmeister & Wain in their four-cycle design, in which the combustion chamber formed by the clearance space in cylinders of orthodox design was replaced by a combustion chamber cast on one side of the lower cylinder head. Practically all of the combustion space volume is concentrated in this side chamber, which is connected by a port to the very small clearance space under the piston. This arrangement puts the combustion air where it can be easily reached by the injected fuel and provides a convenient place for mounting the fuel, exhaust, and inlet valves. In the Werkespoor design the same idea was adopted, but the combustion chamber was made so large that the compression volume is too great for full Diesel compression. As a result the engine must be started single-acting and operated as such until the lower end is warmed up enough to raise the compression temperature to the fuel-ignition point. In the Burmeister & Wain design the possible ill effect of the burning gases sweeping out of the combustion chamber on to the piston rod is prevented by the previously mentioned sleeve which surrounds the rod. The Werkespoor has no such sleeve, but the rod is protected by the expedient of casting a boss on the bottom of the piston, around the bore for the piston rod, and providing a corresponding recess



in the bottom head to receive this boss when the piston is at the end of its stroke. The length of this boss is such that it does not leave the recess until about 8 per cent of the stroke has been completed and combustion is well over. So far as the author knows, the only four-cycle design that places the inlet and exhaust valves directly in the head is that of McIntosh & Seymour. Experience in dismantling and assembling both Burmeister & Wain and McIntosh & Seymour cylinders indicates that the side-combustion-chamber arrangement gives better access for such work.

The builders of two-cycle, double-acting engines, not having the inlet and exhaust valves to contend with, have not used the separate combustion-chamber idea, preferring to inject the fuel directly into the clearance space under the piston. Since this space, because of the presence of the piston rod, has the shape of an annulus, a single valve will not give the desired dispersion of the fuel mist throughout the air, and it is general practice to use two or more injection valves.

As will be seen from Fig. 2, the Worthington design uses two valves, set at an angle to the vertical center line through the cyl-

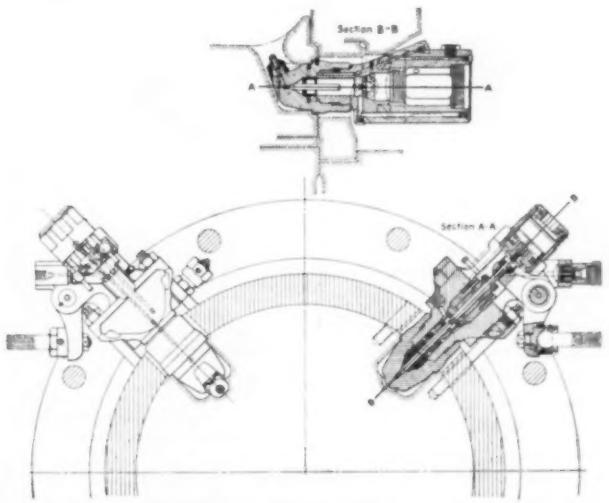


FIG. 3 ARRANGEMENT OF FUEL VALVES ON M.A.N. ENGINE

inder, an arrangement to which the domed shape of the cylinder end is particularly well adapted.

The Deutsche-Werke design uses two valves, but they are not placed in the cylinder head. An intermediate ring, called a valve plate by the builders, is inserted between the cylinder and cylinder head, and the fuel valves are placed horizontally in this ring, diametrically opposite each other and positioned so that the center of the fuel spray is approximately tangential to the piston rod.

The M.A.N. design does not use the tangential spray idea. To obtain wide distribution of the fuel and avoid striking the piston rod with the spray, four fuel valves are used, spaced evenly around the cylinder, as shown in Fig. 3. It will be noted from the figure that the valve bodies are positioned horizontally, but the tips are turned upward so that the sprays enter the cylinder almost vertically. It is obvious that the use of four valves complicates the operating gear. An effect of the vertical sprays that has been noted in operation is the burning of cavities in the bottom face of the piston. It is necessary at times to fill up these cavities by welding in new metal.

FIG. 2 WORTHINGTON DOUBLE-ACTING ENGINE, SHOWING ARRANGEMENT OF FUEL VALVES AND DETAILS OF PISTON

(*A*, cast-iron cylinder liners; *B*, port section liner; *C*, lower clamp rings; *D*, upper clamp ring; *E*, forged-steel piston crowns; *F*, cast-iron ring carrier; *G*, split plate closing piston head; *H*, intermediate skirt; *J*, piston crown studs; *K*, distance pieces.)

The success achieved years ago in the use of mechanical, or airless, injection of fuel in small engines, and the attractive possibilities in the way of simplification offered by this method of injection, led to its intensive development. This resulted in a steady increase in the size of cylinder in which it is possible to use mechanical injection. When the success of the double-acting engine became assured and the use of mechanical injection on large single-acting engines had demonstrated the practicability of that system, it was but logical that the combination of the double-acting, two-cycle principle with mechanical injection of fuel should appeal to designers as the most desirable means for obtaining greater simplicity.

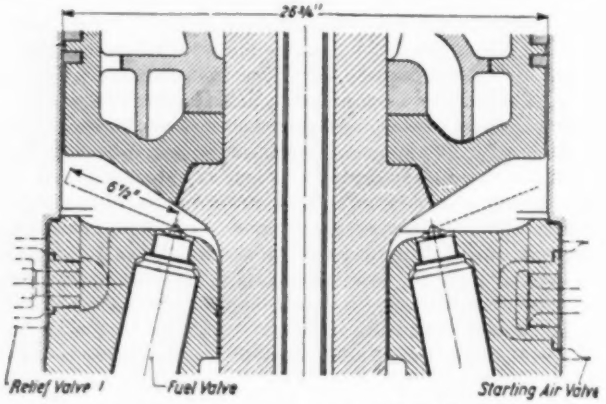


FIG. 4 ORIGINAL ARRANGEMENT OF A.E.G. FUEL VALVES



FIG. 5 LATER ARRANGEMENT OF A.E.G. FUEL VALVES

From the great amount of experimental work done with mechanical-injection systems we have definitely established two important features of the mechanically induced spray. The form of spray that breaks up the fuel fine enough for effective mixture with the air will not penetrate the dense air in the combustion chamber more than 7 or 8 in., and the spray that will penetrate for longer distances does not give the necessary pulverization. As a result of this, in all double-acting engines in which mechanical injection is standard equipment each builder uses the same method to obtain proper distribution of the fuel in the lower combustion chamber—a multiplicity of sprays of limited penetrating power.

A feature of the spray induced by forcing the fuel at high pressure through an orifice is that various shapes and lengths of sprays can be obtained from a multiple-jet valve by simply varying the shapes of the orifices. Examination of three makes of double-acting engines that use mechanical injection shows some different ways in which this principle is applied.

As originally designed, the A.E.G. engine used five spray valves in the lower head, arranged around the piston-rod stuffing box as indicated in Fig. 4, the jets being projected outward toward the cylinder wall. This arrangement was later changed to that shown in Fig. 5, in which only two valves are used, each discharging two sprays, the outer spray being in each case shorter than the inner, because of the difference in the distances through which the fuel must be projected. Theoretically this second arrangement should be superior to the first because of the influence of the sprays causing the air to whirl around the piston rod and promote mixture of air and fuel.

In the M.A.N. engine there is a spray-valve arrangement quite similar to that of the original A.E.G. design, except that only two valves are used, and wide dispersion of the fuel charge is obtained by using six orifices, as shown in Fig. 6. Here, also, the sprays are of different lengths, according to the length of path to be traversed, but the whirling effect on the air is absent.

In the Richardson-Westgarth engine is found an interesting example of the use of different lengths of sprays, as shown in Fig. 7. Two valves, diametrically opposite each other, are placed vertically in the cylinder head, near the outer periphery of the combustion space. As shown in the sketch, each valve discharges five sprays, four of them horizontal and one vertical. Because of the short distance between the valve nozzles and the face of the piston, the vertical sprays are widespreading, with little penetrating power. It is pertinent to remark at this point that the author does not claim that the several sketches accurately portray the spraying actions obtained; they simply show the effects sought by the designers in each case.

A feature of Diesel-engine operation that is easily taken care of in single-acting designs, but which has largely controlled the general arrangement of the double-acting cylinder structure, is the longitudinal expansion of liner and jacket. This presents

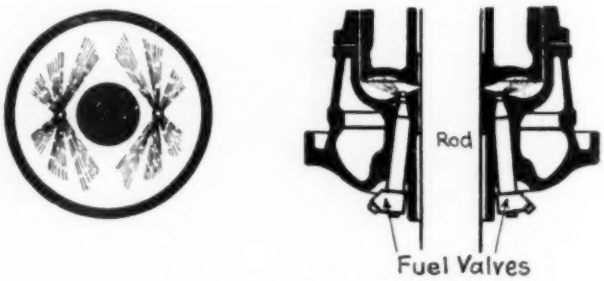


FIG. 6 ARRANGEMENT OF M.A.N. FUEL VALVES

no problem in the case of a cylinder open at one end, but when this cylinder has to be closed and used as a combustion and pressure vessel at both ends, it becomes an important factor. It is this factor that is responsible for the generally used method of construction whereby the cylinder consists essentially of two single-acting cylinders placed end to end.

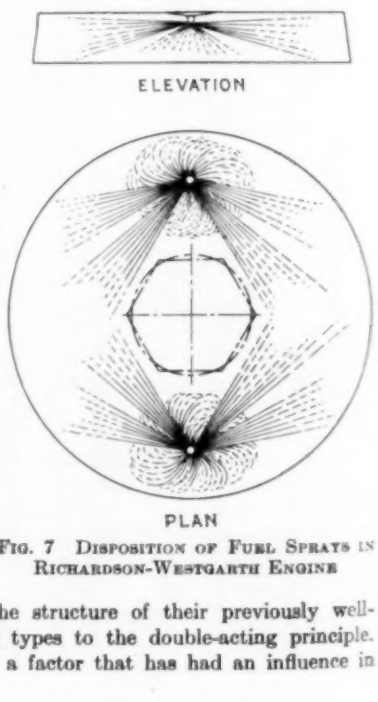


FIG. 7 DISPOSITION OF FUEL SPRAYS IN RICHARDSON-WESTGARTH ENGINE

In considering the cylinders and frames, it may be noted that in the cases of the two successful four-cycle designs the builders have simply adapted the structure of their previously well-established single-acting types to the double-acting principle. In the two-cycle design a factor that has had an influence in

producing almost a standard type of structure is the necessity for establishing, at the middle of the cylinder, connections between the ports and the exhaust and scavenging passages. This suggested the idea of a main fixed structure, of beam type, containing the exhaust and scavenging passages and supported by the frame, with the upper and lower elements of the cylinder attached to its top and bottom, respectively. This arrangement, first used in the Worthington engine, is now used in the Earle, Krupp, and Richardson-Westgarth. A notable departure from this standard type of construction is found in the M.A.N. two-cycle design. Originally the main fixed structure of this engine consisted of the frames, supporting an entablature on which were mounted secondary frames, which supported on their tops an upper entablature. The cylinder jacket and lower half of the liner rested on top of the lower entablature, while the upper half of the liner hung from the under side of the upper entablature. This gave great freedom for expansion, but made the lower head inaccessible. For this reason the design was later changed so that the entire cylinder hangs from the upper entablature, leaving room below it to remove the lower head and lower section of the liner. It may be noted in this connection that in order to make removal of the lower liner section possible it has been shortened and all of the ports placed in the upper liner section.

This matter of liner removal did not receive the attention it deserved until liner breakages in service made it evident that such removal must be made easier if overhaul expense was to be kept within bounds. An example of the development of this feature is found in the Worthington engine, first used in marine service. As originally designed, removal of the top cylinder section and its liner was quite easy, but to get the bottom section out it was necessary to remove the entire cylinder block, weighing several tons, and turn it upside down while suspended in the air. This required expert riggers and equipment not ordinarily available on shipboard. This design was changed so that the liner is now made in three sections. The upper cylinder section can be lifted off, the middle liner section taken out, and the lower cylinder section pulled up through the housing, all with the ordinary ship's gear, without distributing the more massive elements in the engine structure. The same feature has been incorporated in the designs of the Deutsche-Werke, Earl, Krupp, and Richardson-Westgarth. In general it may be said that great progress has been made in the direction of simplification of the cylinder unit to permit easy assembly and disassembly.

This particular phase of development has been accelerated by the frequent breakage of liners in the earlier engines, a condition that gave the liner an undeserved reputation as a troublesome element. Experience, however, developed the fact that liner breakage is in practically every case due to piston trouble and not to any fault in the liner itself. There is no difficulty originating within the liner itself that is not common to both the single-acting and double-acting types. The chief problem with both is the prevention of rapid wear. This wear usually proceeds at a maximum rate during the first 1000 hours of operation, due to initial wearing-in. This wearing-in that has heretofore been considered a necessary part of the "breaking in" of an engine can be eliminated if the engine builders will adopt the same standards for cylinder finish as used by the automobile manufacturers. Already some builders have adopted the grinding process for finishing small cylinder liners. Further progress will be made when it is recognized that a final finish by honing will do a lot toward reducing the rate of liner wear. Experience with numerous double-acting engines in service indicates a wide variation in the rate of wear of liners, but in any case where wear occurs at an unusually rapid rate its cause can usually be traced to some abnormal condition of operation. For liners as at present produced it may be assumed that 0.005 in. wear per 1000 hours of operation is a liberal allowance. Any rate of wear greater than this is excessive.

Much of the operating trouble with double-acting engines has been due to the pistons, although in some cases it required repeated casualties to reveal the identity of the piston as the source of the trouble. In the beginning of double-acting engine design the structural features required to make a good piston were fairly well understood. The educating effect of experience is shown by a better understanding of the most effective mechanical form these features should take, rather than by any change in the features themselves. Fundamentally the piston is a highly stressed

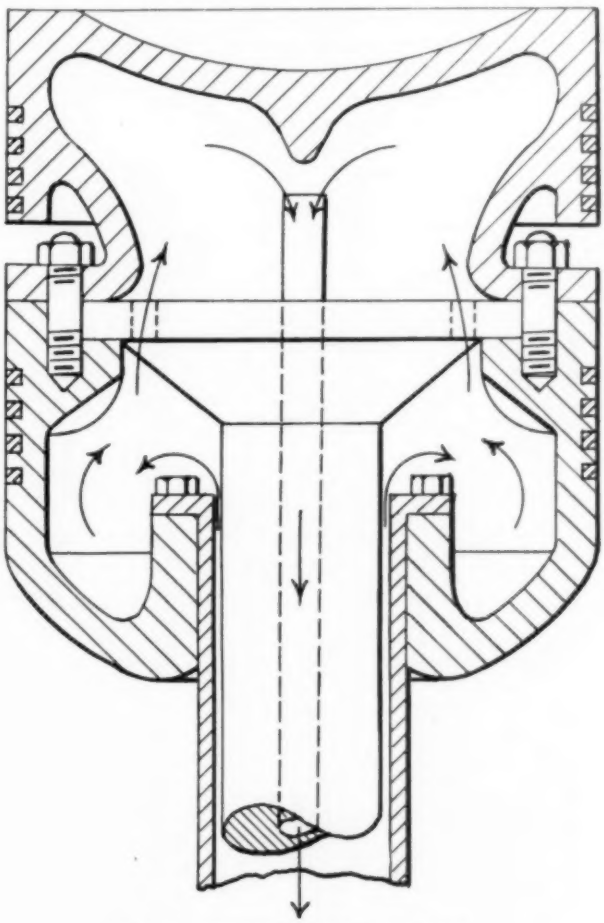


FIG. 8 BURMEISTER & WAIN PISTON

member that must be rigidly connected to the piston rod, yet free to distort under the influence of high heat without excessive stress. Provision must be made for intensive cooling with water or oil, and it is the devising of a practical method of passing this cooling fluid into, through, and out of the piston in such a way as to provide adequate cooling without leakage into the cylinder that constitutes the chief problem of piston design. The only available path through which the cooling fluid can reach the piston is through the piston rod, and the most practicable way of making two flow paths, one for inlet and one for outlet, is to insert an axial tube in the piston-rod bore, utilizing this central tube for one path and the annular space around the tube for the other. A variation of this arrangement is possible with the Burmeister & Wain engine with a protective sleeve around the rod. The space between this sleeve and the rod is used for the incoming oil, and the outflowing oil is passed through the bore of the rod.

In every double-acting engine the piston is essentially two single-acting pistons placed end to end and fastened together, but there are some distinctive differences in the details of construction of the different makes.

The simplest of all double-acting pistons is the Burmeister & Wain, as may be seen in Fig. 8. The two sections are bolted to a singly flange on the rod, a single set of studs serving to secure both sections. The path of the cooling fluid is shown by the arrows in the sketch. It is worthy of note that it has been found necessary to make very little change in this design since the first engine was produced. It is also interesting to note that this same design was adapted to the two-cycle engine recently put out by this firm by simply lengthening the side walls of the upper section, thus giving the extra length needed for two-cycle operation without the addition of the intermediate skirt found on practically all other designs.

What appears to be the most elaborate piston design is found in the A.E.G. engine. Referring to Fig. 9, it may be seen that there are five separate parts. The upper and lower heads, *U* and *P*, are separated by a trunk, in two parts, *D* and *O*, these two sections being bolted together by means of flanges. The complete assembly, consisting of upper and lower heads and the two trunk sections, is held on the piston rod by the large nut *M*, which presses the lower trunk section down on shoulder *B* on the rod. The space between upper and lower piston heads is closed by the split cast-iron skirt *A*. To provide intensive cooling of the piston heads the cooling water is guided through channels formed by extending the center column sections into the piston heads and casting ribs on these extensions. The path thus formed may be traced by means of the arrows in the sketch. It may be noticed that this design eliminates all sliding joints in the piston assembly. An interesting detail not found in other designs is a provision for changing the piston length, for adjusting compression, by means of liner *Z*, inserted between the two halves of the inner column. The piston heads are of forged steel and the central column of cast steel. Since in the original design of this engine the liners were of forged manganese steel, rubbing contact between piston heads and liner was avoided by making the cast-iron skirt larger in diameter than the heads. Trouble was experienced due to heating and distortion of the skirts, this distortion causing the skirt bolts to stretch or break. To correct this the skirt was made in two parts—an upper part, about 14 in. long, being made in the form of a solid ring, and the lower part in the original split form. The split section was given 2.8 mm. greater clearance than the solid ring to insure all the bearing being on the latter. After further experience with the first engine the manganese-steel liners were abandoned for cast-iron liners.

The changes that have been made in the Worthington piston as the result of operating experience may be seen in Figs. 10 and 12, which show, respectively, the original design and the modified

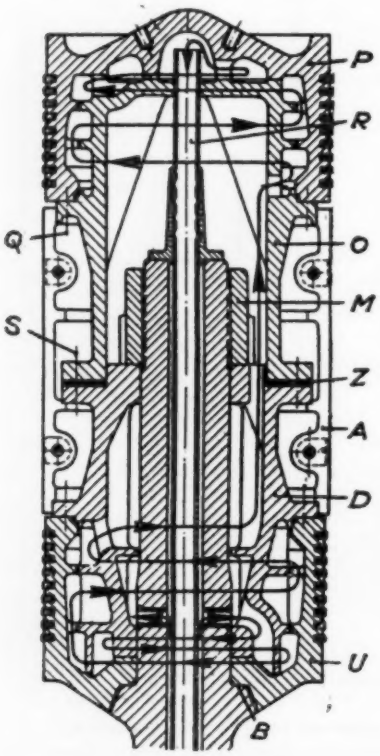


FIG. 9 A.E.G. PISTON

(*U-P*, lower and upper piston heads, of forged steel; *D-O*, center column sections of cast steel; *Q-S*, studs; *A*, intermediate skirt, of cast iron; *M*, piston nut; *B*, seat for piston assembly; *Z*, liners for adjusting piston length.)

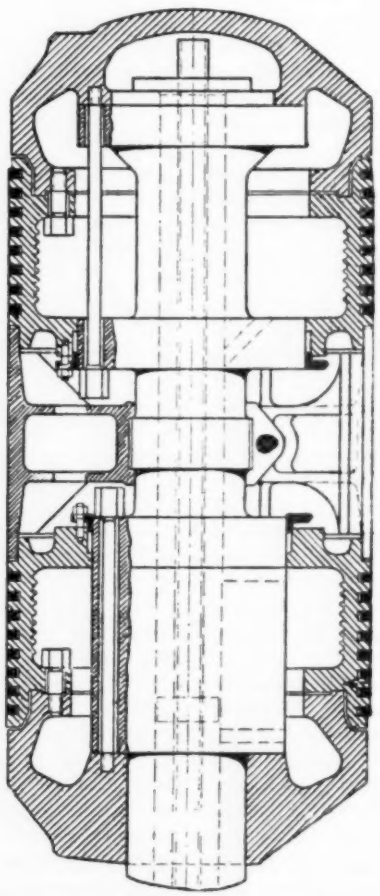


FIG. 10 ORIGINAL DESIGN OF WORTHINGTON PISTON

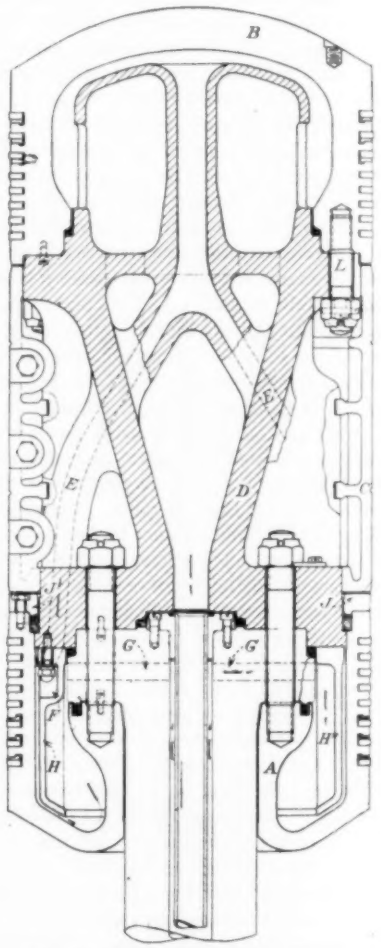


FIG. 11 DEUTSCHE-WERKE PISTON

(*A-B*, lower and upper heads; *C*, intermediate skirt; *D*, distance piece; *E*, water passages cast in distance piece; *F*, inlet to water passages; *G*, water passages in rod flange; *H*, baffle to guide water in lower head; *J*, rubber-packed stuffing box; *K*, water ports in upper baffle; *L*, upper head studs.)

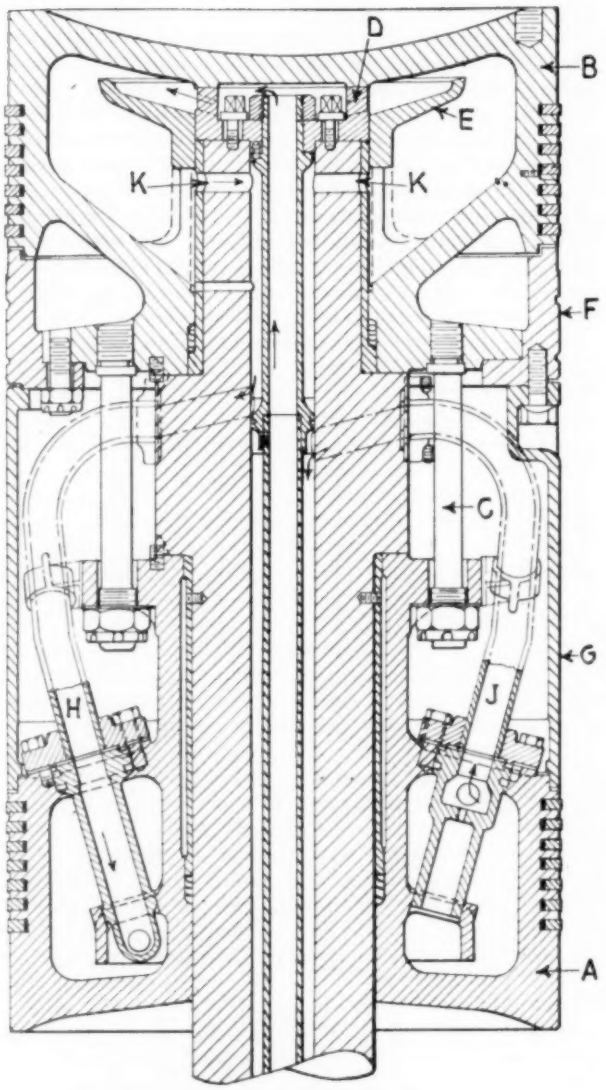


FIG. 12 M.A.N. PISTON

(*A-B*, lower and upper heads; *C*, studs holding both heads on rod; *D*, liner to support crown of top head; *E*, water spreader; *F*, wearing ring; *G*, intermediate skirt; *H*, water inlet to lower head; *J*, water outlet from lower head; *K*, water outlet from upper head.)

design found in later engines. The original make-up, consisting of forged-steel piston heads with cast-iron ring carriers and a cast-iron skirt for closing the space between upper and lower piston sections, has not been changed. The principal change is the removal of the packing gland that closed the joint between the flanged lower end of each ring carrier and the piston-rod flange. To effect this the modified design provides for closing the end of each ring carrier with a flat plate, made in halves and bolted rigidly to the ring carrier. The inner edge bears against the rod and is made tight by simple rubber rings in grooves. This does away with the troublesome gland and removes the packed joint from a region of high heat to a much cooler one. This change permitted the use of short studs, shown at *J* in Fig. 2, in place of the long studs used in the original design for fastening the piston heads to the rod. This later change developed another source of trouble. Distortion of the steel piston head caused the flange held by these studs to cock on its seat and break the short studs. In one of these pistons opened by the author it was found that the broken studs, with their nuts, had rattled around in the piston until holes were worn through plate *G*. To remedy this, the studs were lengthened, and a sleeve *K* was placed under each nut.

In the Deutsche-Werke piston, shown in Fig. 11, there are the conventional elements, two piston heads, the lower of cast steel and the upper of cast iron, with a split skirt of cast iron to close the space between them, but in place of the separate ring carriers of the Worthington piston the piston heads have extended skirts carrying the rings. The upper piston head, instead of being attached directly to the rod, is bolted to a cast-steel distance piece of approximately cone shape, which is secured at its lower end to the flange on top of the piston rod by means of studs, which serve also to fasten the lower piston head to the same flange. A troublesome feature of this piston is the stuffing box *J*, packed with rubber packing and closed by a gland held down by studs and nuts. The author has had occasion to open up 24 of these pistons, and in nearly every case this stuffing box was leaking. In several the rubber packing was cooked to a pasty mass. Repeated breakage of the gland studs also has been found. This emphasizes the fact that a packed gland on the outer wall is a dangerous element in a piston. Another troublesome element in this piston is the split skirt which closes the space between the upper and lower piston heads. Several of these skirts have seized in the cylinders and were distorted enough to break the bolts holding the two halves together. That there is also some distortion of the upper piston heads is indicated by the fact that many of the nuts on studs *L* have become loose in service. It is understood that changes in this piston design have been made, but the author is not familiar with the new design.

In the M.A.N. piston, shown in Fig. 12, there are found the usual upper and lower piston heads, with rings in the skirts and an intermediate skirt to close the space between heads. In addition there is a solid wearing ring *F* attached to the upper piston head. As originally designed, removal of this ring involved dismantling the upper head. To permit easier removal of the ring, some of the Shipping Board pistons were modified by making this ring of two halves, bolted together. The correctness of this change is doubtful in view of the trouble experienced by A.E.G. and Deutsche-Werke with split skirts used as wearing rings. A feature of the M.A.N. piston is the use of bent pipes for circulating the cooling water, as shown in Fig. 12.

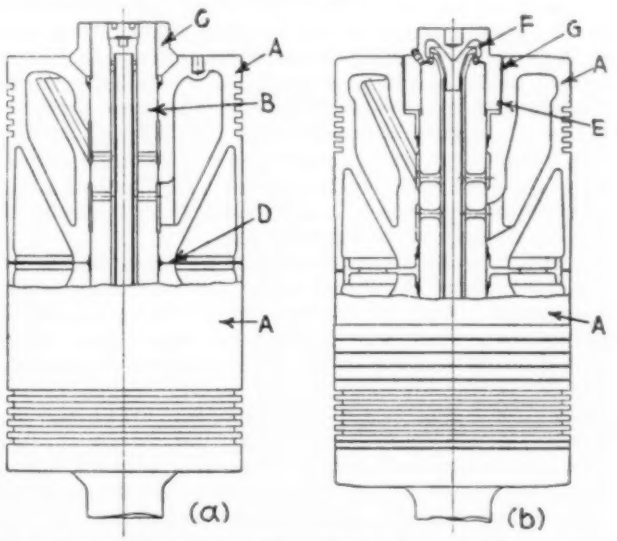


FIG. 13 ORIGINAL (*a*) AND MODIFIED (*b*) DESIGNS OF RICHARDSON-WESTGARTH PISTON

(*A*, upper and lower piston sections; *B*, piston rod; *C*, large nut with cone seat; *D*, seat on lower piston section to receive upper section; *E*, deep recess for nut in modified design; *F*, water passages in nut; *G*, undercut notch for attaching lifting gear.)

The Richardson-Westgarth piston, in its original and in its modified forms, is shown in Fig. 13. It will be noticed that although two similar halves are used for top and bottom there is no intermediate skirt, the skirts of the two piston heads being extended downward to meet at the center, with enough clearance between edges to permit expansion. Another feature of this piston is that the piston rod extends right through both sections and a large nut on the upper end holds the assembly together. The lower piston head rests on a shoulder on the rod, the central boss on the upper head rests on the upper end of the lower head, and the nut presses the whole assembly together. This makes a very simple piston, but it has some glaring defects; the whole assembly is rigidly restrained, with no provision for expansion, the large nut is exposed to the heat of combustion, and the cone-shaped seat of the nut tends to split the top head.

In the modified design these faults were corrected as follows: The seat for the nut was made square and deeply recessed. The seat for the shoulder in the bottom head was similarly recessed, thus leaving the two heads free to expand axially and diametrically. The recessing of the upper head brings the nut below the top of the piston, with the exception of enough projection to permit attaching a wrench. To prevent burning of the nut, water channels are cut in it, and the cooling water leaving the piston flows through the nut to reach the outlet tube in the rod. In addition, the outer surface of the nut is calorized to prevent oxidation in the flame of combustion. A small detail of construction that has given trouble in nearly all double-acting engines is the lifting arrangement. In practically every case where tapped holes in the piston crown were provided for attaching lifting bolts, cracking of the pistons has occurred, the cracks starting at the lifting-bolt holes. Worthington, A.E.G., Richardson-Westgarth and Deutsche-Werke all have eliminated such holes from their latest pistons.

In general it may be said that the main structural features of double-acting pistons have been fairly well standardized in principle, but there is considerable variation in the manner in which these principles find mechanical expression.

Probably the most trouble experienced with double-acting engines has come from a part from which the least was expected, the piston rod. The only anticipated trouble was rapid wear of the rod by its packing. This led the Worthington Company to develop plating equipment and technique whereby their rods could be chromium plated. Their methods proved very successful, but experience with unplated rods in service indicates that this treatment is an unnecessary expense. Measurement by the author of a number of unplated rods after varying lengths of service shows that if the packing is accurately made and carefully fitted very little wear need be expected. The length of service to be expected before such a rod becomes unfit for further use because of wear is as yet undetermined.

Shortly after the first double-acting engines were placed in service, breakages of piston rods began to occur. These were at first ascribed to faulty material or defects in design of the particular engines in which they occurred, but when almost exactly similar breaks occurred in Burmeister & Wain, Worthington, M.A.N., Deutsche-Werke, and other engines it became evident that neither of these causes was the true one. The nature and location of some of these breaks indicated that the generally used method of making cooling-water connections to the rod was responsible for them. Since, as previously stated, the piston rod offers the only means of getting cooling fluid into and out of the piston, and the crosshead offers the most convenient point for making the connection between the moving and the stationary members of the cooling system, nearly all builders used some sort of bracket on the crosshead with connections to drilled holes through the sides of the piston rod. The method of making these connections was usually to mill a flat on each side of the rod in the way of the drilled holes and bolt the pipe flanges to these flats with studs tapped into the rod. The combination of water passages, stud holes, and facings for flanges decidedly weakened the rod at this point, with the result that several builders had rods break in this weakened section. The sketch in Fig. 14 made from one such rod is typical of what occurred. This sort of thing, occurring in several different makes of engines, led to the abandonment of this method of connecting the cooling system to the rod. It is general practice now to make such connection by bolting a small divided chamber to the bottom end of the rod, under

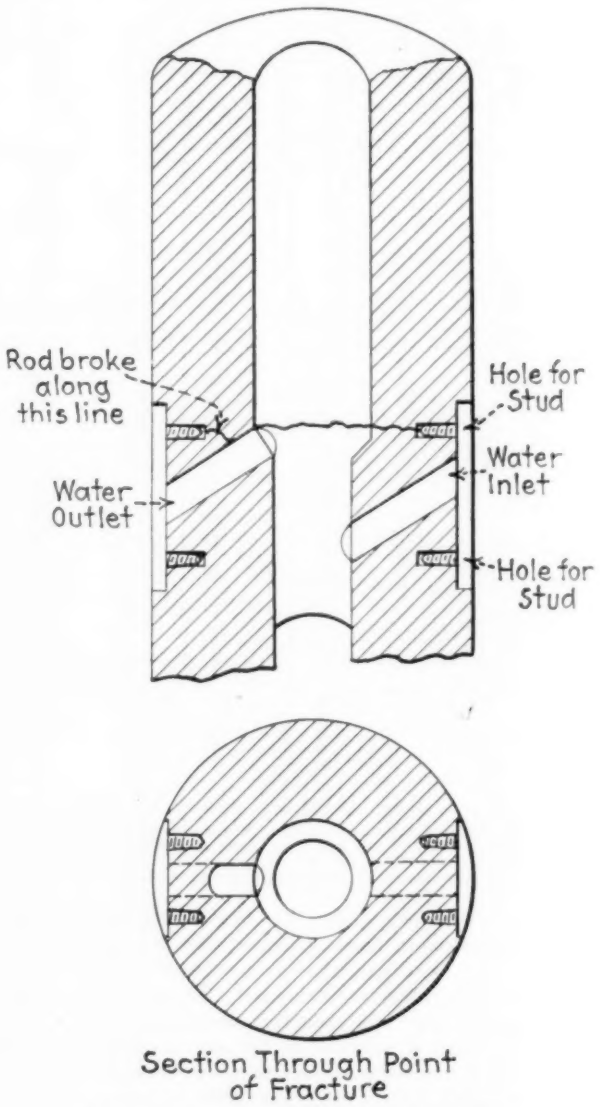


FIG. 14 TYPICAL FORM OF PISTON-ROD BREAK

the crosshead, with one part of the chamber connecting with the axial pipe in the bore of the rod and the other to the annular space surrounding the pipe. The inlet pipe is connected to one and the outlet pipe to the other. This eliminates any weakening of the rod by holes or flats.

Unfortunately this did not stop the breakage of piston rods, although it did reduce the number of such breakages. The breaks that occurred after this were confined to no particular point on the rod. One rod removed by the author broke off just

inside the bore in the piston almost as cleanly as though it had been cut in two. Others broke near the crosshead. The cause for such breaks is yet to be definitely determined, but a clue that is being followed with some promise of results is the possible effect of roughening the surface on the inside of the bore of the rod. One manufacturer who has made extended tests tried an experiment that points to a promising line of study. A rod made of high-grade material that gave uniform bending-test results was operated in the engine until it had made several million alternations and the surface of the bore had been roughened by the action of cooling waters. Samples were then cut from the outer and inner surfaces and given a bending test. The sample from the outer surface bent double without breaking, while the one from the inner surface broke after a very slight bend.

It would appear, then, that the surface of the rod bore should be finished as carefully as the outside surface, which is always polished to make a good rubbing surface for the packing. This is borne out by the experience of the Bethlehem Steel Company some years ago with cast-steel crankshafts. They found that such shafts were safe to use if the bore was polished like a gun barrel. A fine finish in the piston rod bore is of no value, however, unless it is protected against roughening by the corrosive effect of the cooling water. Coating of the surface with zinc or cadmium, or the application of metal by the spraying process, does not give a sufficiently durable coating. Chromium plating is durable, but is very difficult to apply in a long bore of small diameter. The company previously mentioned as having made the bending tests has experimented with rods of stainless steel and also with nitrided rods. It is hoped that the information gained by these experiments will be made public.

Another piston-rod trouble, of a minor nature in itself, but having serious consequences, was experienced by several builders before its nature was discovered. This was the corrosion of the lower ends of the internal pipes in the piston rods. The author knows of cases of two different makes of engines in which such corrosion short-circuited the stream of piston-cooling water and caused repeated piston seizures before the nature of the trouble was discovered. It is general practice now to make these pipes of stainless steel.

Of the many elements that go into the structure of a workable double-acting engine there are few other than those discussed in the foregoing that present any problems not already encountered in the development of the single-acting engine. This has been an important factor in the rapid advancement of the double-acting engine to a position of preeminence in the large-engine field. At the present time there is a very definite trend toward the two-cycle type with mechanical injection of fuel. There is no reason to assume that we have yet reached the limit of power that can be built into a single unit, and the increasing number of power-plant installations being made in Europe leads to the hope that similar applications may be seen in the United States. From the author's viewpoint, that of the repair man, the maintenance of such installations offers no more difficulties than does that of the older single-acting engine.

Discussion

HARTE COOKE.[2] The paper brings out a number of interesting points, one being the difficulty encountered with piston rods of very hard steel.

The wear of one of our steam-engine piston rods may be of interest. This was a 6-in. high-pressure piston rod running in metallic packing on an engine in a cement mill which operated practically continuously, with a shutdown two or three hours per week for maintenance. The steam pressure was 180 lb. superheated, and the engine was carrying approximately 50 per cent more load than its original rating.

Due to the breaking of a connecting-rod bolt, the piston rod was bent. A section from this was micrometered and showed the diameter after twelve years' service to be 5.995 in., the wear being approximately 0.005 in. The surface test gave a Brinell hardness of 124. Analysis showed it to be our usual steam-engine piston-rod material of approximately 0.20 carbon plain-carbon steel.

This definite information raises a question as to the necessity of excessively hard piston rods, and these very hard piston rods for other reasons are not desirable, as illustrated by the experience given in the paper.

In regard to vessels with large powers, it is interesting to note that two of the newest large high-speed vessels, the *Reina del Pacifico* and the *Venus*, use four-cycle, trunk-piston-type engines, supercharged. The fact that they are used in these vessels which require so much power would seem to indicate that in the builders' mind the cost per horsepower must compare favorably with the double-acting two-cycle engines.

However, in fairness to the author, it should be stated that one of the desirable features of these supercharged trunk-piston-type four-cycle engines is the low head room required; that is, they could go under the deck.

---

[2] Mechanical Engineer, McIntosh & Seymour Corporation, Auburn, N. Y. Mem. A.S.M.E.

Light Metal-Alloy Pistons and Rods in High-Speed Industrial Engines

By BENEDICT J. ISIDIN,[1] CLEVELAND, OHIO

The speed of oil and gasoline internal-combustion engines has been brought into a higher range following the availability of improved materials and the necessity of obtaining increased power output without sacrificing space and weight. This often makes it necessary to reconsider some points of design and to adopt better grades of material to solve the problems of higher heat stresses and increase in dynamic bearing loads. Therefore the piston is sometimes cast of steel in thinner section and the cylinder liner is made thinner and is cast of steel to reduce thermal stresses. Connecting rods are made of special alloy steels and can withstand higher stresses than are allowed in plain carbon steel. As a satisfactory means of reducing the weight, the author discusses the use of aluminum, allowing heavier sections, preserving the rigidity, and removing any thermal difficulties encountered in the piston.



IN THE last decade, internal-combustion engines of small and medium output have gained a firm foothold in many special branches of service where the motive power is required at instant notice with guaranteed reliability and economy. Because of the small space it requires and because of its light weight, the internal-combustion engine has become indispensable in solving many problems of agriculture, irrigation, dredging, cotton ginning, excavating, pumping, refrigerating, etc. Both types of this engine (gasoline and oil) are successfully used, depending upon preference or local conditions.

With the course of time, the availability of improved materials, and the economic necessity of getting increased power output without sacrificing space and weight, the speed of these engines has been brought into a higher range. For example, in the case of a dredge, a six-cylinder, 180-b.hp., $6^1/_2$ by $8^3/_4$, 1000-r.p.m. Diesel engine is used; in another installation of a portable air compressor, a 92-hp., four-cylinder, 6 by 8, 1000-r.p.m. Diesel engine is used to advantage. Both of these engines can be classified as high-speed engines.

In order to preserve durability and economical upkeep in high-speed engines, it often becomes necessary to reconsider some points in design and adopt better grades of material to solve the problems of higher heat stresses and increase in dynamic bearing loads. Thus, to withstand higher stresses and to reduce its weight, the piston is often cast of steel with thinner sections. The cylinder liner is made thinner and cast in steel to reduce thermal stresses. Connecting rods, of special alloy steels, are made lighter and can safely withstand much higher stresses than are allowed in plain carbon steel.

In spite of all these refinements in design and materials, the aforementioned problems are only partly solved, mainly because of the limitations of certain physical properties of the materials themselves. The heavy weight and insufficient heat conductivity of steel and cast iron are the main handicaps of these materials in the construction of connecting rods and pistons for these engines. The special alloy-steel connecting rod can be made very light and sufficiently strong, but always with considerable sacrifice of rigidity of construction, as saving in weight is attained by reduction of sections. Rigidity in connecting rods is vital to the life of the bearings, as many minute deflections of the shank throw the bearings out of proper alignment, resulting in quick wear.

Cast-iron or steel pistons, when cast light and in thin sections, are subject to overheating, with resulting carbon formation on the piston head. Local hot spots in the center of the piston head result from the insufficient heat conductivity of iron, and the heat absorbed by the piston head does not flow rapidly enough to the ring belt through the thin sections of the cast-iron piston head. This induces detonation, and in extreme cases is likely to cause the piston to crack, as considerable stress is set up because of the difference in temperature between the red-hot center and the comparatively cool outer edge of the piston head. High temperatures destroy the oil film and carbonize the oil. This results in sticky rings, upsets the lubrication of wristpin bearings, and shortens the life of the piston generally. With increased thickness in the head sections, all of the aforementioned thermal difficulties will be diminished, but the weight of the piston becomes prohibitive for use in high-speed engines.

When the industry itself was in its infancy, low speed, heavy-duty industrial engines were necessarily of low power output. The inertia loads were of no consideration, and all moving parts were made heavy and substantial. It was easy to keep bearings in good alignment, well-lubricated and cool, which resulted in enviable long life.

The designer of modern high-speed industrial engines must preserve this reliability and long life, which is a very honorable but difficult tradition to fulfil, as inertia loads are becoming the predominant factors in engine design. He will be compelled to find new means to reduce the weight of pistons and connecting rods. The most satisfactory way of reducing the weight of these parts is to change the material and to use aluminum, which will allow the use of heavier sections, preserve the rigidity, and remove any thermal difficulties encountered in the piston. By use of aluminum alloys of proper physical properties, it is possible to fulfil all the requirements and obtain pistons and rods which will operate considerably cooler and result in better performance than is obtained from conventional iron and steel parts.

---

[1] Engineering Department, Aluminum Company of America. Benedict John Isidin was born February 25, 1895, in Saratov City, on the Volga, Russia. He was graduated from high school in 1912, and the same year he passed through the competitive examinations to join the Naval Engineers and Architects College, Kronstadt, from which, after completion of a thesis work entitled "Diesel Engines Versus Turbines in Battleships," he was graduated in 1916, receiving a commission in the Imperial Russian Navy as Engineer-Lieutenant (Junior). After graduation from the Submarine Officers' School, he held the position of Chief Engineer in ocean-going submarines, seeing active service in the World War. After the Russian Revolution, Mr. Isidin came to the United States in 1921. Since 1928 he has been with the Aluminum Company of America.

Presented at the National Oil and Gas Power Meeting, University of Wisconsin, Madison, Wis., June 23–26, 1931, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

In general the physical properties of aluminum alloys with which a designer is concerned are as follows:

Specific gravity, 2.7 to 2.9 (cast iron, 7.2; steel, 7.8).

Heat conductivity, 7.7 B.t.u. per cu. in. per hour per 1 deg. fahr. (cast iron, 2.2 B.t.u. per cu. in. per hour per 1 deg. fahr.).

Modulus of elasticity, E = 10,000,000 lb. per sq. in. (cast iron, E = 15,000,000 lb. per sq. in.; steel, E = 30,000,000 lb. per sq. in.).

Aluminum alloys, cast and heat-treated, recommended as a material for pistons: Yield point, 20,000 to 25,000 lb. per sq. in.; tensile strength, 35,000 to 40,000 lb. per sq. in.

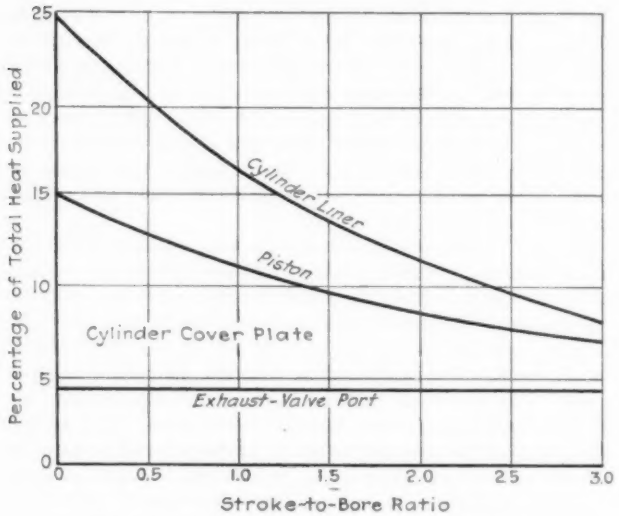


FIG. 1 HEAT FLOW IN A FOUR-STROKE ENGINE

Strong wrought-aluminum alloys recommended for connecting rods: Tensile strength, 55,000 to 63,000 lb. per sq. in.; yield point, 30,000 to 40,000 lb. per sq. in.; Brinell, 90 to 110; endurance limit, 15,000 lb. per sq. in.

Coefficient of thermal expansion, 0.0000123 in. (cast iron, 0.00000556 in. per 1 deg. fahr. per in.).

By using these alloys for pistons and connecting rods, a weight saving of 50 per cent and 40 per cent, respectively, as compared with ferrous materials, becomes possible.

In order to insure satisfactory results when designing parts in aluminum, certain peculiar problems of piston and connecting-rod designs will be considered.

PISTONS

The problems in piston design are mainly those relating to piston-head design and are as follows:

1 Estimation of heat absorbed by piston head
2 Determination of thermal stresses
3 Determination of stresses due to explosion pressure
4 Proper provision for heat transfer from head to rings and piston skirt.

A piston head forms the bottom of the combustion chamber and is the only part of a piston which comes in actual contact with burning gases. These gases being extremely hot, supply, during combustion and exhaust strokes, a certain amount of heat which flows almost entirely (disregarding radiation, which is considered very small) to the cylinder wall through contact areas.

H. F. P. Purday, on the basis of the experiments carried out by Professor Hopkinson and others, evolved diagrams, Figs. 1 and 2, showing heat flow to the piston, cylinder liner, etc. (See "Diesel Engine Design," pp. 170–171.) In these diagrams the percentage of heat passing to the piston liner, etc., is plotted on the basis of the stroke-bore ratio. The total jacket heat is assumed to be 25 per cent of the total heat supplied. When an approximate estimate of heat flow is needed, as is the case when designing engine details such as pistons, liners, etc., these diagrams are very helpful.

When using these diagrams, the quantity of heat flow per square inch of piston head per hour can be estimated as $F = \frac{18{,}000 \times a \times 0.42 \times \text{b.hp.}}{\text{Piston head area, sq. in.}}$, where 18,000 is the calorific value of fuel oil in B.t.u. per lb., a is the heat absorbed by piston in per cent of total heat supplied, 0.42 is the usual fuel consumption in lb. per b.hp. per hour for Diesel engines. The value of this data, as will be shown later, is needed in computing head thickness and thermal stresses.

Heat absorbed by the piston head flows through the head toward the cylinder wall. This condition naturally necessitates an assumption of a temperature drop from the center toward the edge. In the case of a plain disk of even thickness, which condition approximates that of a conventional flat-head piston, the heat flow is governed by formula: $T_C - T_E = \frac{F \times D^2}{16 \times t \times K}$, where $T_C - T_E$ is the temperature difference between the center and the edge of the disk, F is the mean quantity of heat absorbed per square inch per hour, D is the disk diameter in inches, t is the thickness of a disk in inches, K is the heat conductivity in B.t.u. per cubic inch per hour per 1 deg. fahr. This formula indicates that with a given heat flow F B.t.u. per sq. in. per hour, diameter D, and the thickness of the disk t, the temperature difference will be inversely proportional to the heat conductivities of the materials involved. Thus, in the case of aluminum alloy and cast iron, under these conditions, the temperature difference for the aluminum disk will be about one-third the temperature difference of cast iron. Actual temperature measurements by means of thermocouples, made in cases where experience indicates that cracking is not to be anticipated, show that a safe maximum value of $T_C - T_E$ for cast-iron pistons is about 400 deg. fahr. with a temperature in the center of about 800 deg. fahr. If the cast-iron piston is replaced by an aluminum-alloy piston of the same head thickness, the temperature distribution in the aluminum piston head follows essentially the inverse ratio of heat conductivities, as described, or the temperature difference becomes reduced to 130 deg. fahr., and the temperature in the center of the piston head will be in the neighborhood of 500 deg. fahr.

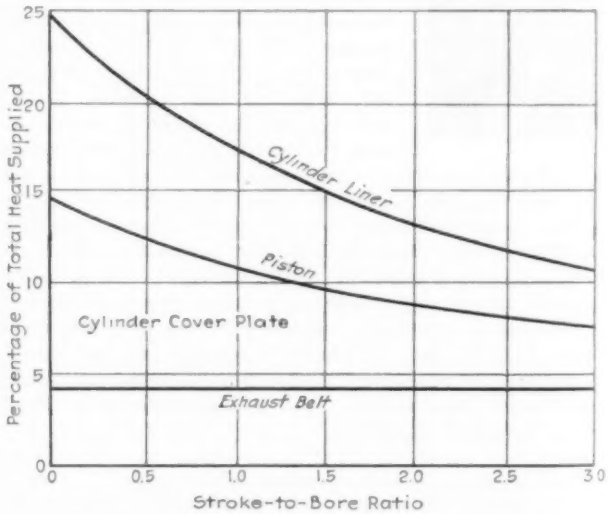


FIG. 2 HEAT FLOW IN A TWO-STROKE ENGINE

Thermal stresses in plain disks can be estimated by the formula $S = \frac{(T_C - T_E) \times E \times \alpha}{2}$, where E is the modulus of elasticity and α is the coefficient of thermal expansion of metals used. The thermal stresses set up in the piston head will be more complicated than in the plain disk, but less dangerous because of the strengthening of the head by the trunk of the piston. Thus, the formula would not represent the actual thermal stresses in a piston head and is used only for comparative purposes.

With the extreme temperature difference of 400 deg. fahr. allowed in the cast-iron piston, the thermal stresses will be about 13,500 to 15,000 lb. per sq. in., depending upon the value of modulus E (12,000,000 to 15,000,000 lb. per sq. in.). These stresses appear to be high for cast iron (ultimate strength, 25,000 lb. per sq. in.) and may serve as a forewarning, showing the thermal limitations of the metal. In the case of the aluminum alloys used in pistons, though the modulus of elasticity and tensile strength are affected by high temperatures, and at 500 deg. fahr. become correspondingly 5,500,000 and 26,000 lb. per sq. in., the thermal stresses in aluminum disks with a temperature difference of 130 deg. fahr. will be only $S = \frac{(T_C - T_E) \times E \times \alpha}{2} = \frac{130 \times 5{,}500{,}000 \times 0.0000123}{2} = 4400$ lb. per sq. in. (yield point of aluminum alloy for pistons at 500 deg. fahr. = 18,000 lb. per sq. in.). This is indicative of a better safety factor and of a wider range of application.

Returning to the formula $T_C - T_E = \frac{F \times D^2}{16 \times t \times k}$, which can be rewritten as $t = \frac{F \times D^2}{16 \times K \times (T_C - T_E)}$, it becomes possible to

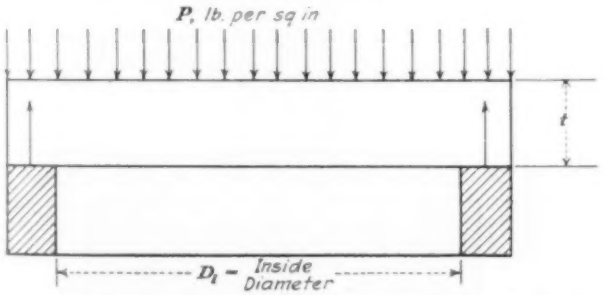


Fig. 3 Flat, Round Disk Under Action of Two Bending Moments in Opposite Directions

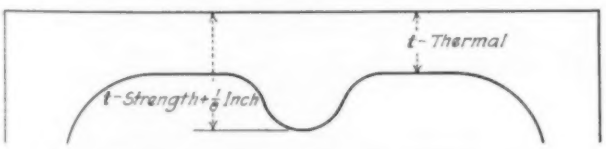


Fig. 4 Reinforcement of Disk by Longitudinal Rib

calculate the head thickness of a piston for any given heat flow F in B.t.u. per sq. in. per hour, diameter D, and for any predetermined safe temperature difference. As was shown, for the lightest cast-iron piston design, which is reasonably safe, the value of $T_C - T_E$ becomes 400, and the piston-head thickness can be expressed as $t_{CI} = 0.000071\,FD^2$. In order to get a temperature difference of 130 deg. fahr. in an aluminum piston, under the same conditions, the head thickness will be $t_{AL} = 0.000063\,FD^2$. This indicates the possibility of a weight saving of 65 per cent as compared with cast iron, but usually the sections are cast heavier in aluminum than in cast iron since the main motive for substituting aluminum for cast iron is to increase the power (this increases F), and also to provide rigidity comparable or better than cast iron. As a result the total weight saving will be about 50 per cent as compared with the cast-iron piston.

In the foregoing discussion, the stresses caused by explosion pressure were not taken into consideration, and the thickness figured by the formula must be checked to be within safe limits to withstand this load. The calculation of the head thickness for strength is based on the supposition that the head of a piston is

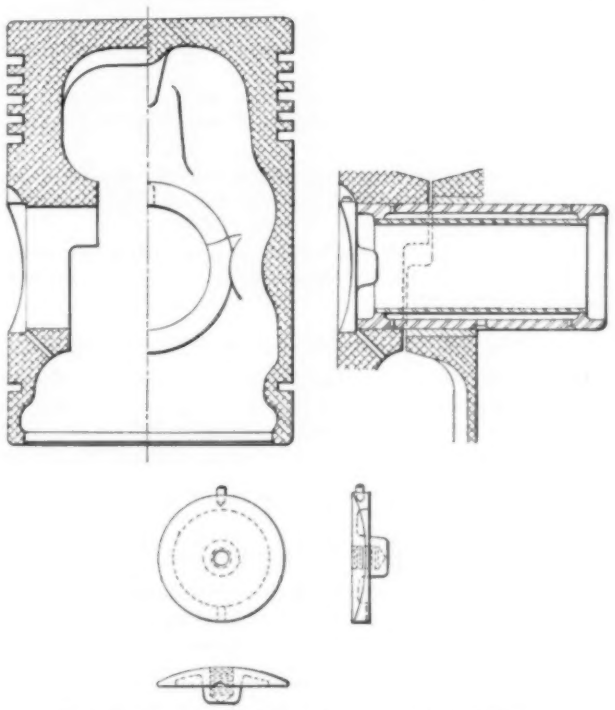


Fig. 5 Typical Design of Aluminum-Alloy Piston

a flat, round disk, supported around the edge and uniformly loaded.

The dangerous section of a flat, round disk will be along the diameter. This section is under the action of two bending moments acting in opposite directions. The first one is due to the explosion pressure and the second is due to the reaction forces of the supports (Fig. 3). The resulting moment is the algebraic sum of both and is equal to $M_B = \frac{D_1^3 \times P}{8} - \frac{D_1^3 \times P}{12} = \frac{D_1^3 \times P}{24}$. The section modulus of the flat, round disk with thickness t is equal to $\frac{D_1 \times t^2}{6}$. If S designates stresses, then $\frac{D_1 t^2}{6} \times S = \frac{D_1^3 \times P}{24}$, or $S = \frac{D_1^2 \times P}{4t^2}$ and $t = \frac{D_1}{2}\sqrt{\frac{P_{max}}{S}}$. Allowable stress for aluminum alloys should not exceed 5500 lb. per sq. in. If the t calculated for strength exceeds that calculated for thermal stresses, it is usually recommended to reenforce the head by a longitudinal rib of a depth in excess of t (strength) by $^1/_8$ in. (Fig. 4). The rib should be cast with as big radii as consistent with the size of the piston. This diminishes the high cooling stresses which are likely to occur with sharp changes of section.

Fig. 5 shows a typical design of an aluminum-alloy piston. As the details of the drawing are self-explanatory, it is sufficient to add only a few comments. Pistons are usually cast in semipermanent molds, that is, in metal molds having baked sand cores.

The chilling effect of the metal mold produces a hard, close-grained surface which insures long life for the thrust faces, ring grooves, etc. Further, pistons are heat-treated to attain hardness and strength and to remove casting strains, thus making a uniform product. In the following are summed up a few practical observations resulting from experience with aluminum-alloy pistons:

1 A large number of deep, narrow rings are better than a small number of shallow, wide rings of equivalent contact area.

2 A full-floating wristpin in combination with stepped-up bosses and wristpin plugs, as shown in Fig. 5, is preferred to any other type.

3 A stiffening rib cast around the skirt and between the wristpin bosses controls distortion and keeps the thrust faces round under all conditions.

4 A full-pressure lubrication of the wristpin bearing is desirable; aluminum wristpin plugs prevent an over-lubrication of the piston.

5 In many instances the clearances with which aluminum pistons can be operated are essentially the same as for cast-iron pistons.

Connecting Rods

From a consideration of Rankine's column formula, around which the design of connecting rods is usually built, it becomes apparent that considerable weight reduction can be attained when a steel rod is replaced by a rod made of strong aluminum alloy. The Rankine formula reads as follows: $S = \frac{P}{A} + 0.000526 \times \frac{P}{A} \times \frac{L^2}{R^2}$, where P is the total load on the rod, A is the area at the section of maximum stress, S is the allowable stress, L is the length of the rod, R is the radius of gyration of section. The first member of the formula, P/A, represents the compression per unit of area of the cross-section of the shank, and the second member, $0.000526 \times \frac{P}{A} \times \frac{L^2}{R^2}$, takes account of the additional stress due to the bending of the shank. R^2 is the ratio I/A where I is the moment of inertia of the section. This ratio is very flexible and is within the control of the designer. The moment of inertia depends upon the shape of the section and may have different values for the same area.

Allowable stresses in the case of plain carbon-steel connecting rods are approximately 18,000 lb. per sq. in. and approximately 24,000 lb. in the case of chrome-nickel steel. Allowable stress in aluminum-alloy connecting rods is approximately 12,000 lb. per sq. in. With a given P and L, the Rankine formulas for these three rods may be written as follows:

1 Carbon-steel rod: $\frac{P}{A_1} + 0.000526 \times \frac{P}{A_1} \times \frac{L^2}{R_1^2} = 18{,}000$

2 Chrome-nickel steel rod: $\frac{P}{A_2} + 0.000526 \times \frac{P}{A_2} \times \frac{L^2}{R_2^2} = 24{,}000$

3 Aluminum rod: $\frac{P}{A_3} + 0.000526 \times \frac{P}{A_3} \times \frac{L^2}{R_3^2} = 12{,}000$

It is evident that $A_1 > A_2$, and consequently the rigidity of the rod with area A_2 is impaired as compared with the rigidity of plain carbon-steel rods. An attempt can be made to restore the rigidity by increasing R^2, but this increases the rigidity only in the plane of motion with considerable loss of rigidity at right angles to the plane of motion. This indicates that an attempt to save weight by using special steels invariably leads to decrease in the rigidity of the structure. This is the result of the elastic modulus of steel being essentially independent of the composition, mechanical working, and heat treating of steel. In an aluminum rod the sectional area A_3 is bigger than A_2 or A_1 of steel rods. With increased area A_3, a designer has an excellent opportunity to shape the section for the best rigidity in both planes, having the considerations of clearance between the edges of the cylinder wall and rod as the only limiting factors against this ambition.

With a given P and L, the rigidity of the shank will be in direct proportion to the product of the elastic modulus of the material and the moment of inertia of the section. This indicates that in order to attain the same rigidity in the aluminum rod as in the steel rod, it is necessary to have the equation $E_{AL} \times I_{AL} = E_{ST} \times I_{ST}$.

As the elastic modulus of steel is three times greater than that of aluminum, this equation becomes $I_{AL} = 3 \times I_{ST}$. As the moments of inertia are in direct proportion to the fourth power of linear dimensions, it is evident that to satisfy the equation, it is necessary to increase the linear dimensions by $\sqrt[4]{3} = 1.316$ or 31.6 per cent, with consequent increase of area by $\sqrt{3} = 1.732$ or 73.2 per cent. From the consideration of specific gravities for forged aluminum alloy and steel (correspondingly 2.8 and 7.9), the apparent weight saving for the foregoing conditions becomes $\frac{7.9 - 2.8 \times 1.732}{7.9} = 38.6$ per cent. From the

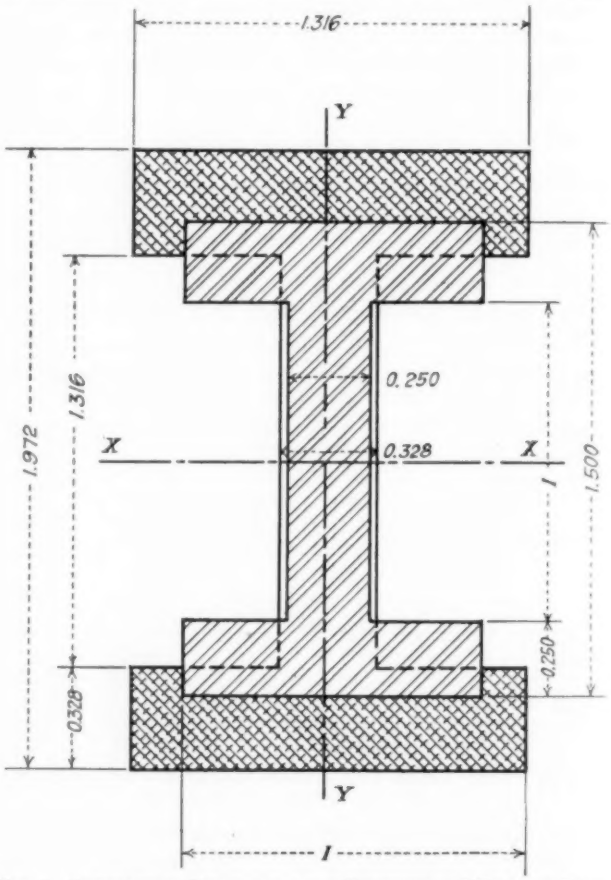


Fig. 6 Increase of Dimensions for Rigidity of I-Beam in Both Sections

	Area	I_{XX}	I_{YY}
Steel rod	0.750	0.219	0.043
Aluminum rod	1.297	0.655	0.129

$$\frac{\text{Area aluminum rod}}{\text{Area steel rod}} = \frac{1.297}{0.750} = 1.732$$

$$\frac{I_{AL}}{I_{ST}} = \frac{0.655}{0.219} = \frac{0.129}{0.043} = 3$$

example given in Fig. 6 proper increase of dimensions in the I-beam section for equal rigidity in both planes can be observed. Whenever there is no interference anticipated between the rod and cylinder wall, this change in cross-section is possible; otherwise, it requires less efficient metal distribution, with consequent increases of sectional area and weight. But even under the most unfavorable condition, the weight saving in the shank of the rod is nearly 30 per cent.

The weight saving in the wristpin and crankpin ends of the rod is considerably greater than is possible in the shank. This becomes evident from the following consideration: As the bearing width remains the same for aluminum and steel rods, the only possible change in dimensions will be an increase in the wall thickness. In this case the ratio $\frac{I_{AL}}{I_{ST}} = 3$ becomes equal to $\frac{t_{AL}^3}{t_{ST}^3}$, where t = thickness of wall.

This indicates that a necessary increase of thickness, in order to attain the same rigidity as in steel, will be $\sqrt[3]{3} = 1.442$ or 44.2 per cent, with a weight saving of nearly $\frac{7.9 - 1.442 \times 2.8}{7.9} =$ 49 per cent. From the foregoing considerations it becomes apparent that the total weight saving of 40 per cent in replacing the steel rod with aluminum is a very conservative estimate.

As the rules applicable to steel-rod design remain the same for aluminum, it will be sufficient to mention only a few suggestions, as follows:

1 It is necessary to avoid sharp corners in the machining or the spot facing of a recess for the bolt nut and head. Radius should be $^1/_{16}$ in. minimum (Fig. 7).

2 Direct babbitting of the big end of aluminum rod is given preference over that of steel or bronze-backed shells. A natural bond between babbitt and aluminum insures a very quick evacuation of heat from the bearing, and the bearing metal is more rigidly supported by the solid sections of the rod itself.

3 A bronze bushing with floating pin in small end of the rod.

4 Minimum draft angles 10 deg. in drop-forged rods.

Polar diagrams of big end-bearing loads for aluminum and steel rods are shown with a weight reduction of 40 per cent established in pistons and connecting rods where aluminum is substituted for iron. This is the case of a six-cylinder Diesel engine, $6^1/_2$-in. bore, $8^3/_4$-in. stroke, with a normal operating speed of 1000 r.p.m. This engine may be considered at the present time as belonging

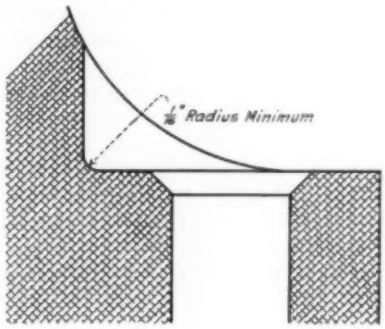


FIG. 7 MINIMUM RADIUS IN MACHINING RECESS

to the high-speed class. Improvements in fuel-injection methods, however, indicate that in the near future engines of this type may have the rotative speed increased to 1500 r.p.m. The complexity of thermal and mechanical difficulties arising from attempts to increase the engine speed and still preserve intact the reliability and longevity of parts common to slower speed engines will not be discussed here; therefore only the problem of bearing-load reduction in connection rods will be considered.

Table 1 indicates the comparative weights of iron and aluminum pistons and connecting rods and the weight reduction of the reciprocating and rotating masses thus effected.

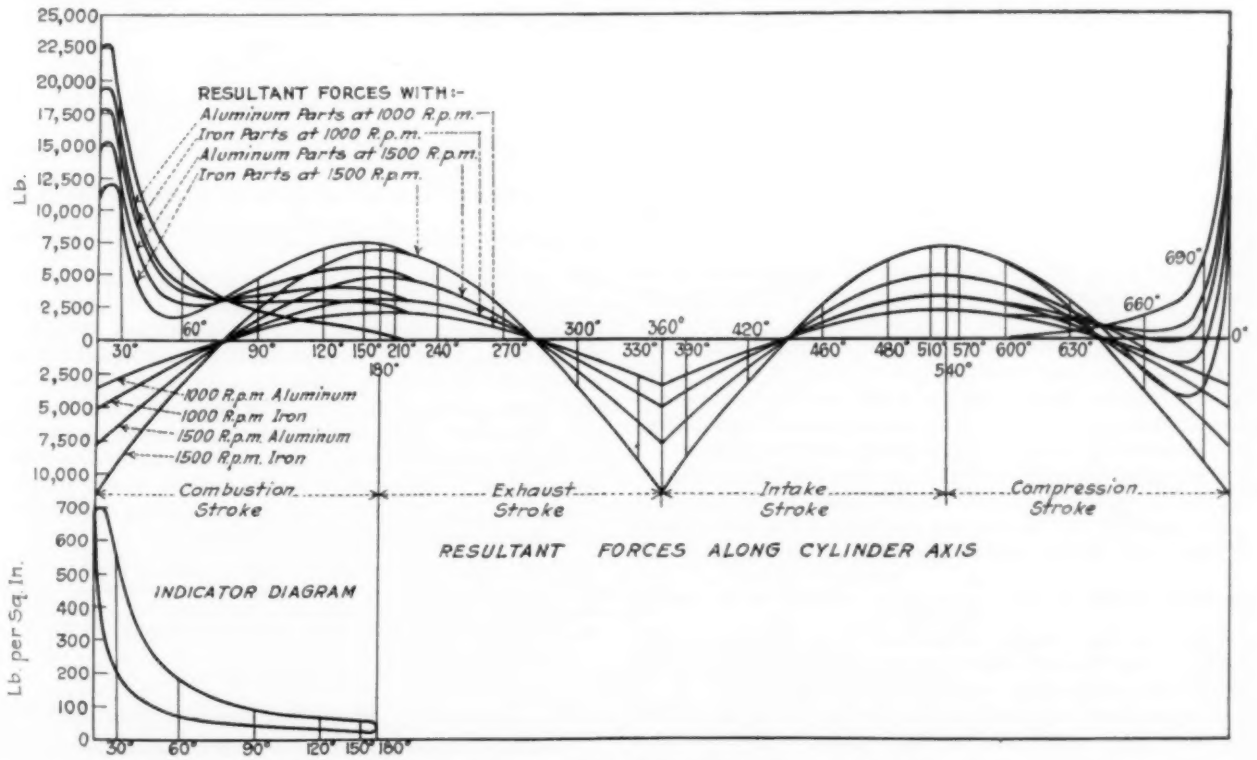


FIG. 8 DIAGRAM BASED ON THE COMPARATIVE FIGURES OF TABLE 1

TABLE 1

	Iron, lb.	Aluminum, lb.
Piston	22	11
Connecting rod	25	15
Total reciprocating weight	33.5	22.65
Rotating weight	16.6	10

The diagram, Fig. 8, was made by using the figures of Table 1 as a basis. The polar diagram, Fig. 9, compares unit bearing load and PV factor for both rods at 1000 r.p.m. Though bearing loads and PV are not very high at this speed for both steel and aluminum rods, it is of interest to note, however, that to get the same PV using aluminum rods as when using steel rods, the engine speed can be increased 10 per cent or up to 1100 r.p.m. (unit bearing load, 410 lb. per sq. in.; V = 18 ft. per sec.; PV = 7400 ft. lb. per sq. in. per sec.). At 1000 r.p.m., the normal rating of this engine is 180 b.hp. and it is safe to assume that at 1100 r.p.m. the output is increased in direct proportion with the increase of speed, or almost to 200 b.hp. The polar diagrams, Fig. 10, are prepared for the same rods in 1500 r.p.m. At this speed the advantage of lower unit bearing load and PV will be still more appreciated if the heat-dissipating ability of aluminum is also considered.

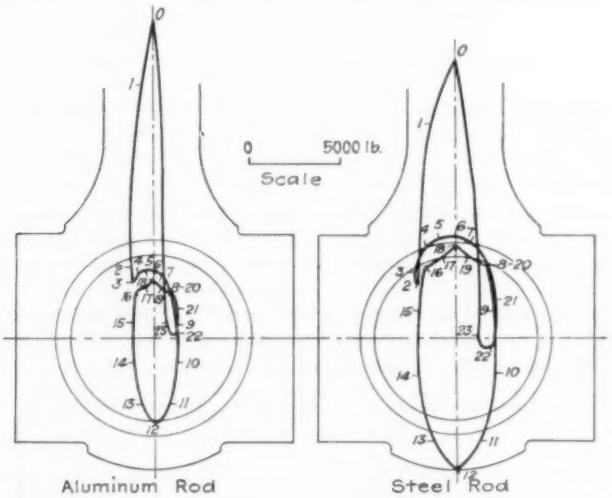


FIG. 9 LOADS IN BEARINGS OF ALUMINUM AND STEEL CONNECTING RODS AT 1000 R.P.M.

(Diameter of crankpin, 3.75 in.; projected bearing area, 11.5 sq. in.; rubbing velocity, 16.35 ft. per sec. For aluminum rod: Mean bearing load, 340 lb. per sq. in.; mean PV at 1000 r.p.m., 5560 ft-lb. per sq. in. per sec. For steel rod: Mean bearing load, 450 lb. per sq. in.; mean PV at 1000 r.p.m., 7350 ft-lb. per sq. in. per sec.)

If the PV value is not the limiting factor, as may be the case in some instances, quite often the consideration of critical speed sets the limit for speed increase. In this case the weight reduction in reciprocating and rotating masses again is very helpful, as it removes the range of critical speeds to higher values.

The critical speed of an engine occurs when the natural frequency of revolving masses coincides with the frequency of power impulses. In the case of a six-cylinder, four-cycle engine, if 1000 r.p.m. will be a critical speed, the natural frequency will be equal to 3000 per min. $\frac{F_1}{F_2} = \sqrt{\frac{W_2}{W_1}}$, where F is the natural frequency of the rotating system and W is the total rotating weight. This equation holds true only when the location of the node is not changed with change of rotating weight.

Weight reduction in reciprocating parts in the engine is 10.85 lb. per cylinder, or 65 lb. for the whole engine. Only half of this weight, or 32.5 lb., can be included when considering rotating masses. On the big end of the rod the weight reduction effected is 6.66 lb. per rod, or 40 lb. per engine. As this engine has a counterbalanced crankshaft, in order to preserve balancing, the same weight must be removed from the counterweights. This makes a total weight reduction in rotating masses of 112.5 lb.

As was shown, the natural frequencies for a given system are inversely proportional to the square root of the rotating weights ratio, and the new natural frequency with a weight reduction of 112.5 lb. will be equal to $F_2 = \frac{3000}{\sqrt{\frac{W_1 - 112.5}{W_1}}} = \frac{3000}{\sqrt{1 - \frac{112.5}{W_1}}}$. In the case of $\frac{112.5}{W_1} = 0.25$, which is very possible, as the total original rotating weight for this size of engine hardly could exceed 450 lb., the new frequency will be equal to $F_2 = \frac{3000}{\sqrt{0.75}} = \frac{3000}{0.866} =$ 3470 per min., or the engine speed can be safely increased to 1156 r.p.m. with corresponding increase in power. If PV and critical speed are not the limiting factors, and a speed of 1500 r.p.m. can be attained, consideration of the diagram of Fig. 10 will indicate that, for given conditions, the bearing in an aluminum rod will outlast that in a steel rod. The aluminum-rod bearing will run considerably cooler, not only because of a smaller unit bearing load and PV, but because of the superior heat-dissipating ability of aluminum. This results in cooler lubricant, which, retaining heavier body, separates two rubbing surfaces in a more efficient way, thus prolonging the life of the bearing.

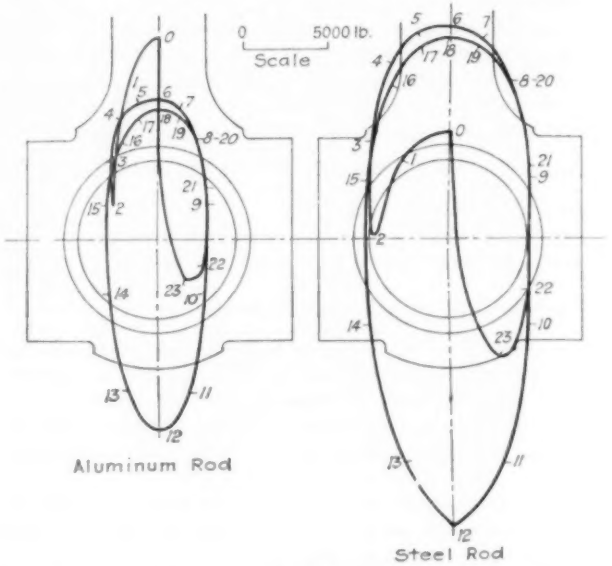


FIG. 10 LOADS IN BEARINGS OF ALUMINUM AND STEEL CONNECTING RODS AT 1500 R.P.M.

(Diameter of crankpin, 3.75 in.; projected bearing area, 11.5 sq. in.; rubbing velocity, 24.57 ft. per sec. For aluminum rod: Mean bearing load, 563 lb. per sq. in.; mean PV at 1500 r.p.m., 13,830 ft-lb. per sq. in per sec. For steel rod: Mean bearing load, 794 lb. per sq. in.; mean PV at 1500 r.p.m., 19,500 ft-lb. per sq. in. per sec.)

[NOTE: The derivation of the basic formulas, treated briefly in this paper, are explained in more detail in the author's paper, "Diesel-Engine Alloy Pistons, an Engineering Analysis," *Oil Engine Power*, vol. 7, nos. 7 and 9 (July and Sept., 1929), pp. 339–402 and 516–517.]

Discussion

B. J. ROBERTSON.[2] The author speaks of direct babbitting

[2] Associate Professor, Internal Combustion Engines, University of Minnesota, Minneapolis, Minn. Mem. A.S.M.E.

of the bearings in the big end of the connecting rod. What provision is then made to take care of the additional clearance in this bearing when it becomes hot? The coefficient of expansion of aluminum is approximately twice that of the steel in the crankshaft, which should produce excessive clearance when the bearing is hot or too tight a fit when the bearing is cold. Assuming 100 deg. fahr. temperature rise, the clearance of a 3-in. bearing would increase about 0.002 in.

Some automobile manufacturers use steel caps in combination with aluminum connecting rods to reduce the amount of expansion. Is this practice found necessary in heavy industrial engines?

Author's Closure

Steel bearing caps, as a measure of reducing the amount of expansion in the big end of aluminum connecting rods, are not needed in industrial engines. It has been found sufficient to fit this bearing with somewhat closer clearances than is customary in steel rods. A temperature range in bearings of industrial engines between the cold starting conditions (70 deg. fahr.) and the normal operating conditions (180 deg. fahr.) permits satisfactory and safe clearances to be obtained. The tendency of aluminum rod bearings to increase the clearances when hot is sometimes found to be beneficial as more oil passes through increased clearances, with considerable cooling effect on the bearing.

At the present time the use of aluminum connecting rods with steel caps has been confined to automobile engines, as here the temperature range in bearings may vary from sub-zero at starting to 250 deg. fahr. or more at operating conditions. The steel cap removes the trouble sometimes experienced with the all-aluminum rod, if initially fit too tight, of the freezing of the rods on the crankpins in sub-zero weather.

OGP-53-9

The Diesel-Engine Situation Today

By MAX ROTTER,[1] ST. LOUIS, MO.

The first commercially successful Diesel engines were a product of American manufacturers, but now large Diesels are being built principally in Germany. This country, however, is making considerable headway in manufacture of engines up to 1000 b.hp. per cylinder. The author discusses developments in atomization, combustion, and supercharging. He feels that there is now a need for standardization, but it must not interfere with progress.

IN 1898 the first Diesel engine in the world, to be put into commercial service, was built by Adolphus Busch and installed in the Anheuser-Busch brewery in St. Louis. Between that time and 1912, the Busch organization designed and sold the only Diesel engines built in this country. In 1912, when the Busch-Sulzer company started its new factory, this type of engine was being built, both here and in Europe, solely for stationary service, and when the plant was laid out, the engineers considered themselves quite optimistic when they set as the probable limit in engine size, a cylinder of 400 b.hp. and an engine of six such cylinders. Fortunately the overall dimensions and weight of this engine were determined on the basis of the then current piston speeds and heavy designs, so that the factory, as actually built, can turn out engines of more than three times the then estimated probable limit of power. These circumstances are mentioned to bring out the phenomenal expansion that has taken place in the intervening 18 years.

Today there is not one field of power generation and use that has not been penetrated by the Diesel, from the aeronautic engine to the large stationary engine of 15,000 b.hp. And by far the greatest and most rapid expansion has been in the application of the Diesel to marine propulsion, executed for the first time, in 1905, by Sulzer Brothers, in Switzerland, as a two-cycle, and in 1908, by Nobel, in Russia, as a four-cycle engine.

It is but fitting that this survey of the past include the recognition that is due, but has never been accorded to the American engineer who, at the very beginning, impressed his ideas upon the Diesel of two decades; namely James D. McPherson. It was he who designed the first trunk-piston type and the first multicylinder, vertical Diesels in the world; who developed a fuel pump and its governor control that are typical of those still being built; who showed, in his design of the combustion chamber, that he had a true understanding of penetration, diffusion, and turbulence that by many years antedated the adoption of these terms; and who installed the first Diesel engines in the world to run on coal tar, which they did successfully, for many years, in the plant of the United Gas Improvement Co. in Philadelphia.

McPherson's engines were the original type A, Busch-Sulzer. They were not as refined as modern designs; but 65 of them have been in regular operation more than 20 years; the dean being a 225 b.hp. engine, installed in Texas in June, 1904, and still going strong. It is worth noting that the fuel consumption of these early engines was about as good as that of quite modern Diesels, namely 0.42 to 0.43 lb. per b.hp.-hr. Incidentally, such records should serve to condemn the habit of many engineers, of estimating the useful life of a Diesel engine as 10 to 15 years.

It is a lamentable fact that Europe has now far outrun us although it was this country that in the beginning established the type for Diesel engines. Those who read trade journals that contain news of Diesel engines, cannot fail to be struck by the fact that we have not built a single stationary engine of over 4000 b.hp.; while, for example, Germany last year built two of 11,500 b.hp. each, and is building 5 stationary engines of 5000 b.hp. each for a single South American installation. In the marine field the comparison is still less pride-inspiring. Last year European countries completed 135 motorships of 3000 gross tons and over, with an aggregate propelling power of nearly 600,000 s.hp. of Diesels, only 3 of which ships were for American account; while we did not turn out a single such motorship, our output being limited to the conversion by the Shipping Board of 10 ships, of a little over 6000 gross tons each, with a total of less than 40,000 s.hp. Among the European-built ships are notable the *St. Louis* and *Milwaukee*, each of 17,000 gross tons, and each powered by 12,500 b.hp. in M.A.N. 2-cycle Diesels; also 3 ships for the New Zealand Shipping Company, each of 17,600 gross tons, and each powered by two 5000 b.hp., 5 cylinder, 2-cycle, Brown-Sulzer Diesels. And none of the foregoing figures include Japan, which last year completed the *Asama Maru*, of 16,500 gross tons and 15,000 s.hp., in 2-cycle Sulzer Diesels.

It would be ridiculous to assume from the foregoing that we, in this country, are incapable of producing large Diesels, equal to those built in Europe. We have always excelled in large things. In the sister industry we have designed and built the world's largest gas engines, which are operating successfully on leaner gases than are used anywhere in Europe. That the American Diesel industry generally turns to Europe for its large engine types, and restricts itself to adapting these to American methods and conditions, is natural, in view of the limited demand we have enjoyed, which has not justified the expense of independent development. Even our Navy has adopted a German design of submarine engine. One wonders whether our shipping industry is following the same fatuous course it did 70 years ago, when it lost its supremacy on the seas by clinging to the sailing clipper, of which it was justly proud, while Europe adopted the steamship.

While we are marking time in large Diesels, we have, in the past few years, proceeded very rapidly in engines of smaller

[1] Vice-President of Engineering Busch-Sulzer Bros.-Diesel Engine Co. Mem. A.S.M.E. The author was born in London, England, August 13, 1867. Educated in City of London School. From 1883 to 1888 he was apprenticed with Greenwood & Batley, Leeds, Eng., builders of steam engines and special machinery, while attending engineering courses of Governmental Science and Art Department, South Kensington, and of City and Guilds of London. From 1888 to 1891, draftsman for Maxim-Nordenfelt Co., Erith, England. He came to the United States in 1891, and worked as machinist with Browne & Sharpe; 1891 to 1894, chief draftsman for Walker Manufacturing Co., Cleveland; 1894 to 1911, first as draftsman and mechanical engineer for Fraser & Chalmers, Chicago, later as mechanical engineer, and finally chief engineer for Allis-Chalmers Company, Milwaukee, during which time made special investigations in Europe, as a result of which Allis-Chalmers Company took up manufacture of steam turbines and large gas engines. From 1911 to date, chief engineer, and since 1917 vice-president in charge of engineering for Busch-Sulzer Bros.-Diesel Engine Co., St. Louis, and doing private work as consulting mechanical engineer in Milwaukee. During World War, he supervised designing submarine engines for U. S. Navy.

Presented at a meeting of the Washington, D. C., Section, May 8, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

capacities. The Diesel locomotive is making headway in capacities up to 1000 b.hp.; the Diesel railcar is an established vehicle, the aeronautic Diesel has been perfected, and the automotive Diesel is coming to the fore. It is not egotistical to assert that we are doing more research work relating to liquid-fuel injection and combustion than any other country.

While the investigation of the processes of fuel injection is far from complete, and has unfortunately been confined to airless injection, we are learning the limitations of jet penetration and diffusion—knowledge that is vital in the design of large airless-injection engines. This explains why such large engines generally show mean pressures much below those attained in their smaller prototypes. At present, airless-injection engines with cylinder bores larger than 24 in. show mean indicated pressures below those attained with air injection; while in smaller engines the reverse is the case. These researches also explain the proved desirability of imposed turbulence in an airless-injection engine; and give a clearer understanding of some of the functions performed by the spray air of an air-injection engine, beyond that of merely spraying the fuel into the cylinder; namely, the creation of turbulence and the improvement of diffusion by the intimate mixing of air and fuel during injection, and the expansion of the spray air, as well as its jet velocity as it enters the cylinder.

Difference of opinions have developed concerning atomization and combustion, and, while some of them may be academic, they are interesting. As a matter of fact many will probably affect the design of Diesels as little as the theories of water-drop formation in the vapor affect the design of condensers. Little concerning atomization has yet been established beyond the fact that the drops of fuel break up as they travel through the compressed air. Undoubtedly a certain amount of atomizing is caused by the sudden expansion of the fuel, as it leaves the nozzle. After that, one theory is that the pressure against the leading side of the drop flattens it, until the fluid tension is overcome, and the drop breaks up; another, that the velocity of the drop creates a pressure reduction behind it, the drop assumes a streamline shape, and its trailing end is gradually triturated. Then there is the question whether vaporization or gasification precedes combustion, or the drop burns directly on its surfaces. Intricate calculations of flame propagation, which were inherited from gas-engine theories, seem to have been wasted effort. It may even be doubted that combustion proceeds according to the commonly accepted pictures; that is, accompanied by projected flames. Possibly it proceeds in the manner in which finely divided iron combines with oxygen, when introduced into it. That it is flame that issues from a cock in the cylinder may be due to the lower outside pressure.

The remarkable thing is that these processes can be accomplished in the time available for them, sometimes a little over two-thousandths of a second. Without practical demonstration this would seem to indicate the advisability of preventing the atomized fuel from striking a surface by which its combustion would be retarded until it is decomposed; nevertheless there is evidence that the drop may strike a hot piston surface and be broken up, the smaller drops rebounding from it, due either to their resilience or to the pressure of vapor formed under and within them, as a drop of water explodes on a hot stove lid. Likewise, the drops of fuel should not collide, unless while their velocity is still high enough to disrupt them, and before it is low enough to allow them to coalesce.

In conjunction with questions of fuel injection in Diesel engines, it is interesting to consider that of excess air; that is, the large percentage of the air, contained in a cylinder, that is not required for the combustion of the fuel. This excess air is generally about one-half of the total, sometimes nearly two-thirds, and rarely less than 40 per cent. The natural snap judgment is to ascribe this to lack of perfection in the fuel-injection process, and to assume that perfecting the process would afford some of the advantages now attained by supercharging. While it may be true that, even if it were desirable, we are not yet able to consume a greater proportion of the air and attain smokeless combustion; it is doubtful that we could, at this date, go much farther in that direction, being limited by the temperature resistance of available materials; and it is equally true that we could not attain anywhere near the complete utilization of the oxygen, without a substantial loss in overall efficiency, and without an increased engine cost. Ordinarily the air, in a Diesel cylinder, is compressed to about $6^1/_2$ per cent of its original volume, attaining a pressure of 480 lb. absolute, and a temperature of 880 deg. fahr.; in this air is burnt a given weight of fuel, say at normal engine load about one-half of the weight for which the air could supply the necessary oxygen, the volume of air and gases expands to about $2^1/_2$ times the original volume of compressed air, with little increase in pressure but with an increase in temperature to about 2700 deg. fahr. The air and gases then expand about $6^1/_2$ times in volume, and to a pressure of about 50 lb. absolute, performing work equivalent to a mean indicated pressure of about 115 lb. per sq. in. of piston area. Should 50 per cent more fuel be burnt in the same quantity of air, the temperature at completion of combustion would increase to at least 3700 deg. fahr., too high for ordinary present-day materials and design. These conditions do not exist in a properly proportioned supercharged engine in which the air excess is considerably greater than in a non-supercharged type.

Supercharging, by which is understood the artificial increasing of the weight of combustion air with which the cylinder is charged before the fuel is injected, is gaining some adherents. Witness the propelling engines installed by Harland & Wolf in the motorship *Llangibby Castle*, each engine normally rated at 3200 b.hp., and capable of developing 4250 b.hp. when supercharged.

The general idea of supercharging is quite old; 20 years ago Sulzer Bros. were experimenting with it, but were turned aside temporarily by more urgent lines of development, such as the two-cycle engine.

There are several systems of supercharging. The best known are the Rateau and the Buechi, both of which provide the supercharging air by means of centrifugal blowers, driven by turbines using the exhaust gases of the engine as motive fluid. In general, these systems are similar to that developed in this country, under Dr. Moss, of the General Electric Company, for supercharging the gasoline engines of aircraft, to maintain sea-level air conditions in the engines, at high altitudes and thus enable the engines to carry almost full load regardless of the rarified atmosphere. As stated before, by proper adjustment of the compression ratio and the air excess, these supercharged engines are not subjected to higher temperatures or higher maximum pressures than those existing in similar normally charged engines; although, of course, the average cylinder pressures are higher. The pressure of the supercharging air is about equal to the pressure of the exhaust gases, generally 18 to 20 lb. absolute; and the capacity increase of the engine, 30 to 40 per cent.

In the Buechi system the speed of the blower and therefore the pressure of the supercharging air automatically adjust themselves to the existing engine load, due to the variation of the exhaust pressure with the load. The result is high efficiency, and a very flat fuel-consumption curve. This system is, therefore, suitable for continuous operation. To date it has been applied exclusively to four-cycle engines; although it might well be applied to certain two-cycle types. The reason why the latter has not been done lies in conflicting patent interests.

Distinguished from the former systems of supercharging, intended as a regular condition of operation, is the Sulzer system

intended primarily to increase the overload capacity of a two-cycle Diesel engine. In this system the supercharging air is furnished by a compressor, attached to the engine, which compressor comes into and goes out of action under control of the engine governor, so that supercharging takes place only at the higher engine loads. The automatic valves in the upper tier of the Sulzer scavenging ports for the normal scavenging and charging are supplemented by timed valves, controlling the delivery of supercharging air to the cylinder, so that the latter air, furnished by the supercharing compressor, is admitted only toward the end of the charging period; that is, after the piston has covered the exhaust posts, and before it covers the upper scavenging ports. With this arrangement the capacity and power consumption of the supercharging compressor are reduced to the minimum. The capacity of the engine may temporarily be increased as much as with the Buechi system; but the fuel efficiency of the engine is adversely affected to the extent of about 5 per cent, due to the power consumed by the supercharging pump, even when not in action. However, it is an inexpensive means to increase the peak capacity of an engine, and when the available space does not permit the installation of a larger engine.

Still another system of supercharging has been applied to certain German four-cycle engines, for the same purposes as accomplished by the Sulzer system. In this system also, the supercharging air is furnished by a compressor attached to the engine. In addition to the normal admission valve, a timed supplementary piston valve is provided, which gives access of atmospheric air to the admission valve during the earlier portion of the suction stroke, and of supercharging air during the later portion of the stroke. This, as in the Sulzer system, reduces the dimensions and power consumption of the charging compressor, below what they would be if all of the charging air were furnished by it; but it, too, increases the fuel consumption at all loads.

As in the Buechi system, the compression and air excess in the Sulzer and German supercharged engines are adjusted so that the combustion temperatures will be substantially the same as in the un-supercharged engine. The process of scavenging, that is, clearing the cylinder of the products of the previous combustion, has a close relationship to supercharging, and is receiving considerable attention.

The scavenging of a four-cycle engine is an obvious process. When the exhaust valve is opened, the major portion of the products of combustion are expelled from the cylinder by their expansion; the remainder is pushed out by the piston. This necessitates opening the exhaust valve against the terminal pressure in the cylinder, about 35 lb., requiring rugged valve gearing. Attempts have been made to reduce this work by providing exhaust ports, uncovered by the piston near its outer dead center, but the complication of this construction has not been justified by its advantages. A recent invention provides a smaller relief valve, within and opening in advance of the main exhaust valve, but it is almost a foregone conclusion that this, also, will not be worth while.

The scavenging of a two-cycle engine is a more difficult problem, including, as it does, the simultaneous recharging of the cylinder. These processes must both be accomplished in 40 to 50 deg. of crank travel, as compared with 360 deg. available for the same, in a four-cycle engine. Moreover, the process is not a definite mechanical one, as it uses air to displace the spent gases, and it has therefore been and still is the subject of considerable experimentation, including, as it does, the creation of the desired character of turbulence. All such experiments suffer from the same limitations. They are made with a substantially constant stream of air, through a constant cylinder volume, against a constant pressure, through constant openings, without disturbing mechanical motions within the cylinder. They have been of value, however, in pointing the direction of "cut-and-try" experiments, but can scarcely be considered as more than remote indications of processes that are complicated by the variation in pressures, and stream velocities and directions, caused by the progressive uncovering and covering of ports by the piston; by the variable cylinder volume; and by the already existing turbulence of the gases due to their expansion and outflow. Every catalog picture, illustrating a scavenging system, shows clean-cut curves and loops, embellished with arrowheads, sometimes in two colors, red for the hot exhaust gases, blue for the pure cold air, which indicate what might reasonably be expected if the cylinder were a vertical slice through a hemisphere, with a scavenging port to the east, and an exhaust port to the west.

It does not appear that any Diesel designer has yet attempted to scavenge and charge a two-cycle cylinder through exhaust and scavenging valves, both in the cylinder head. All other combinations include piston-controlled ports, either for exhaust, or for scavenging, or for both. Most of the earlier two-cycle engines exhausted through ports and admitted scavenging air through valves in the head; but this arrangement has been almost universally abandoned. Reversing the process, we now find the new B. &. W. two-cycle, double-acting engine admitting scavenging air through piston-controlled ports, and exhausting through timed valves in the cylinder heads. This, including the use of the exhaust valves as supplimentary power pistons, is decidedly the most novel development of last year, and the performance of the engine in the *Amerika* will be watched with keen interest. Its reported efficiency and capacity are remarkable for a double-acting, airless-injection engine.

The simplest arrangement has been most generally adopted, piston-controlled ports for the scavenging air around one-half the cylinder circumference, and the same for the exhaust around the opposite half, but in at least one construction the scavenging and exhaust ports alternate around the whole circumference. Usually the exhaust ports are first uncovered by the piston, and the scavenging ports somewhat later, the interval being sufficient to allow the pressure in the cylinder to fall below that of the scavenging air. This arrangement affords the least time for charging, and under certain conditions allows exhaust gases to blow back into the cylinder. An improved condition is obtained, in engines like the Fiat and Setz, by locating the edges of both scavenging and exhaust ports so that they will be uncovered simultaneously, and providing the former with inwardly opening valves. Neither of these arrangements enables the charging of the cylinder to a pressure higher than the exhaust back pressure; in fact, the eductor action of the exhaust may cause an under-pressure in the cylinder charge. There are examples of increasing the back-pressure, by throttling the exhaust, but this is liable to affect the scavenging adversely, and should be resorted to only to assist in equalizing the work among the several cylinders.

The arrangement that most nearly fulfills the desired conditions is that developed by Sulzers, and improved by Reale, in which the scavenging ports are arranged in two tiers, the upper being provided with automatic inwardly opening valves, and being uncovered before and re-covered after the exhaust ports, thus providing the maximum time for charging.

A marked tendency at this time is toward airless injection. Although this system has long been in use on engines of smaller capacities, and has been applied to large engines, by M.A.N. occasionally, by Doxford more extensively, and with marked success in the A.E.G.-Hesselman engine, its more general use in large engines has recently increased notably. A comparison of the performance of air injection and airless quickly dispels the former common belief that the efficiency of the airless is

higher to the extent of the power consumed by the injection air compressor, which consumption has ordinarily been stated to be 10 per cent of the power developed by the engine. The average airless-injection engine actually consumes only about 5 per cent less fuel than an equivalent engine with air-injection.

The injection of 1 lb. of fuel requires about 20 cu. ft. of free air, compressed to 800 to 1000 lb. pressure. If these quantities are considered as being per minute, they would apply to an engine of about 150 b.hp.; and the compression of the spray air would consume about 10 b.hp. In ordinary practice, adjustments are not made so closely, and the compressor load may readily be 50 per cent higher than this, apparently justifying the stated consumption of 10 per cent. The spray air is compressed in three stages, and cooled to about atmospheric temperature before it reaches the fuel valve. Thus the conditions are, roughly, that air at say 900 lb. pressure and atmospheric temperature in the fuel valve, enters the power cylinder with a pressure drop to say 475 lb. There its temperature is raised by the compressed air in the cylinder, to about 840 deg. fahr. From this point the spray air acts as part of the compressed-air charge in the cylinder, expanding and performing work on the piston, just as does the air compressed in the cylinder itself, whether the injection be by air or airless. The actual power chargeable to the compressor is, therefore, its friction (say $2^1/_2$ hp.), plus the power consumed in compressing the spray air from 475 to 900 lb. (say $1^1/_2$ hp.); this sum, liberally figured, is thus less than 3 per cent of the net engine output. The weight of the air in the compressor space of the engine is about 30 lb. per pound of fuel, and its temperature, as stated, about 880 deg. fahr. before being cooled by the injection air. The acceleration and atomization of the fuel consumes about the same power as in an airless-injection engine, say $^3/_4$ hp. per lb. of fuel per min., and the temperature of the whole cylinder contents is reduced about 40 deg. fahr. To overcome this and place the engine on a par with airless injection, the compression in the power cylinder of the air injection is increased about 20 lb., adding to the work of compression about $2^1/_2$ hp., which is absorbed by the cooling effect of the spray air. Thus, the total power consumption, in such an engine, chargeable to air injection, ranges from $6^1/_2$ to 8 hp., or about $4^1/_2$ to $5^1/_2$ per cent of the engine capacity.

The foregoing accounts for the difference in fuel consumption, and the higher compression usual in air-injection engines. The figures are, of course, subject to modification according to the size of the engine, and the commercial value of the saving must be adjusted to the prices of the fuels that may be used successfully in the respective engine types. It must not be assumed that the higher efficiency, simplification, and reduced first cost of the airless-injection engine are gained without some sacrifice in ability to use fuels that may be cheaper, and this disability may become important if and when liquid fuels other than petroleum, shall be produced to sell at prices below those of petroleum, and this is not an unlikely development.

Other tendencies in construction and arrangement, some of which may be overstepping the desirable limit, and about each of which much could be said if time would permit, are the double-acting engine; Diesel-electric propelling engines, as installed in the coverted American ships *Triumph*, *Courageous*, and *Defiance*, a type of propulsion that has not found favor in Europe; Diesel-geared propulsion, as installed in a number of large European ships, including the recently commissioned *St. Louis* and *Milwaukee*, with 12,000 s.hp. each, and the much discussed *Ersatz Preussen*, with 50,000 s.hp.; but in only one moderate-sized American ship, the *Herman Falk*, with 2600 s.hp. There is also the composite steam turbine and geared Diesel installation, in a naval cruiser, in which the center propeller, consuming 12,000 b.hp. will be Diesel driven, to obtain high efficiency and long radius of action at the ordinary cruising speed of 18 knots and in which the two outboard propellers, each consuming 30,000 b.hp., will be driven by steam turbines, to enable a ship speed of 32 knots when necessary.

It is natural that in an industry that has grown to the magnitude of Diesel manufacturing, there should be an urge toward standardization, and standardization confined to proper channels is most desirable, but it must not overflow into domains in which it would retard progress, or discountenance efforts toward improvement. For example, it is clearly most desirable to standardize the form and dimensions of articles, also the composition and properties of materials, if such articles and materials are extensively used, but it is most undesirable to standardize the application and conditions of use and service of such articles and materials. Where would the steam-turbine industry be if the speed, steam pressure, and steam-temperature conditions had been standardized or limited 25 years ago, leaving no incentive for engineering skill and metallurgical research, beyond that of endeavoring to reduce the cost of manufacture? Standardization must not be such as will abridge the effort to improve, as well as to reduce cost, by skillful design and the intelligent selection of materials. The recently suggested limitation or standardization of Diesel-engine ratings, on the basis of mean indicated pressures, would appear to come within this proscribed territory. This limitation has been predicated on the theory that the Creator gave us certain materials, which possess definite properties; it must not be overlooked that He gave us also brains to modify such properties by combinations and treatment of such materials, and to select materials according to their intended use and service. Consider, for instance, the piston head of a Diesel. It may be well designed or poorly designed; it may be cooled or uncooled; it may be made of ordinary or special-alloyed cast iron; of forged or cast carbon or alloyed steel, or of aluminum or magnesium alloys; and superimposed on this choice of materials are infinite variations in molding, casting, and heat treating. In addition to which, every change in diameter and rotative speed presents changed problems. To require the application to all of these of a single rule, based on temperatures, is not reasonable.

A maximum basis of rating is not a protection for the purchaser. Its acceptance will not ensure good judgment or honesty, any more than constitutional amendments and laws ensure abstemiousness and chastity. As in these, the violators will discount the probabilities of being caught, and there are conditions of service, such as marine propulsion, in which the actual power consumption cannot be readily established. A maximum rating is likely to be a standard rating, when the conscience is flexible; whereas the rating of an engine should express the work that particular engine may be depended upon to perform, with safety and reasonable wear, and with a fair margin to allow for possible improper adjustments, with due regard for the available opportunities for correcting them. For this reason, Busch-Sulzer, for example, establish from 3 to 5 distinct ratings for one and the same engine, the highest being for stand-by service, in which the engine may operate only a few days in the year; the lowest for single-engine ocean-going ship propulsion.

The variety of the bases of rating, proposed by British engineers, indicates the complexity of the matter. For example, they have suggested that the rated normal capacity of an engine be the power it will develop continuously for 12 hr. (presumably with invisible exhaust), and that the continuous rating be 10 per cent below this; but they apparently recognize the deficiency of this method by limiting it to engines for driving electric generators. It has also been suggested that the ratings be fixed at established percentages below the stalling torque, or the load at

which the engine definitely begins to drop its speed; but this torque and load will vary considerably with the immediate condition of the engine, the character of fuel, lubricating oils, and atmospheric conditions. And neither of these methods differentiate between different qualities of engines; nor does the third European suggestion that the rating be a definite percentage below the maximum load the engines will carry with smokeless exhaust.

The recent American proposal to limit Diesel ratings to certain maxima in mean indicated pressures, assumed to reflect maximum permissible temperature stresses, is unsatisfying for the previously stated reasons, and, furthermore, selects a criterion that is confessedly unreliable, excepting in slow-speed engines, but which it proposed to apply to engines of speeds up to 800 r.p.m. It would appear more logical to limit the amount of fuel burned per minute per square inch of piston area, but this is not being advocated in this paper. It is rather the writer's belief that so long as the majority of purchasers compare engines on the basis of price per horsepower, per pound, and per cubic foot of cylinder volume, which latter, by-the-way, a purchaser recently did without considering that he was comparing a four-cycle with a two-cycle engine, it is each builder's job to justify his own rating, as applied to his engine for the contemplated conditions of service.

The foregoing leads to another question: What is the m.i.p. of a four-cycle engine? This is important, in view of the practice of many consulting engineers of limiting this pressure and requiring a guarantee of brake efficiency. That such limitations and guarantees are of little value, without qualifications that would practically specify the make of engine, is beside the question. The purpose of the practice is obviously to limit the temperature stresses at the specified useful power output. Therefore, the method of measuring the m.i.p. should be such that this pressure will truly reflect the temperature conditions.

In a four-cycle indicator card, the bottom loop area represents the work of pushing out the exhaust gases and drawing in the air charge. The more inadequate the valve areas and ports, the larger will be this work, and, if the area representing it be deducted from the upper diagram area, to determine the m.i.p. and brake mechanical efficiency, the poorer the engine design the greater will be the area of the upper diagram for the same m.i.p. and efficiency, and the higher the temperatures. Moreover, the bottom loop represents the same work as is performed by the scavenging pump of a two-cycle engine, yet it would be entertaining to hear the remarks of the purchaser's engineer if the builder of a two-cycle engine were to deduct the mean pressure calculated from the scavenging pump diagram from the mean pressure of the power-cylinder diagram in establishing his m.i.p. and brake mechanical efficiency, although there is little difference between the work consumed by the scavenging pump of a well-designed two-cycle engine, and that consumed in the exhausting and charging of a less well-designed four-cycle, and there is a substantial cooling effect exercised by the large volume of scavenging air passing through the two-cycle cylinder.

OGP-53-10

The Quiescent Combustion Chamber

By JOHN A. SPANOGLE,[1] HAMPTON, VA.

The term "quiescent" is originated as describing more accurately than "non-turbulent" the condition of the combustion chamber in some fuel-injection engines at the time of injection of the fuel. Such a combustion chamber is considered quiescent if the movement of the air has no appreciable effect on the distribution of the fuel to the air with which it must unite during the process of combustion. Performance tests of a single-cylinder unit furnished the material for the paper, in the general program of the National Advisory Committee for Aeronautics in investigating the compression-ignition, fuel-injection engine as a source of power for aircraft. The simplest combustion chamber is the quiescent one, and the simplest injection orifice is the round hole; and the main tests were directed toward obtaining the optimum combination of these two elements.



ANY discussion of the compression-ignition, fuel-injection engine is hampered by the lack of general acceptance and universal use of descriptive terms for the elements and functions of elements which are incorporated in this type of engine. Descriptive adjectives for combustion chambers denoting the conditions of air flow therein have not as yet come into general use for the fuel-injection engine, although the authorities on the carbureted engine seem to be generally satisfied with the use of the respective terms "turbulent" and "non-turbulent" for combustion chambers with and without certain kinds of air flow. The use of the term "turbulence" in this sense is by definition (deviation from mean velocity) apparently unsuited for use by any one connected with aeronautics. In addition to this objection, there is also the thought that turbulent flow is not suited to the requirements of the fuel-injection engine and that better results may be obtained if the air flow is entirely orderly and reproducible for every individual engine cycle.

This consideration discourages the use of the opposite term "non-turbulent," and in casting about for another properly descriptive term with the requisite background, it was found that there are objections to all such terms in common use. Therefore it was decided to use an adjective that is without definite engineering precedent, but which describes the condition of the combustion chamber at the time of injection of the fuel in a manner that is satisfactory but not necessarily ideal. Any combustion chamber for a fuel-injection engine is considered "quiescent" if the movement of the air in this combustion chamber has no appreciable effect on the distribution of the fuel to the air with which it must unite during the process of combustion.

The material in this discussion was obtained from performance tests of a single-cylinder test unit with a combustion chamber that seems best suited to operation without effective air flow. The work was done by the staff of the National Advisory Committee for Aeronautics at the Langley Memorial Aeronautical Laboratory in the course of the general program for investigating the possibilities and limitations of the compression-ignition, fuel-injection engine as a source of power for aircraft. This type of power plant has well-known, inherent advantages which make it particularly desirable for aircraft service, but the requirements for this service are so rigid that these natural advantages cannot be made available unless the performance of the compression-ignition, fuel-injection engine can compare favorably with that of the carbureted gasoline engine.

The cylinder head of the engine used in these tests is not necessarily ideal and probably has disadvantages for commercial

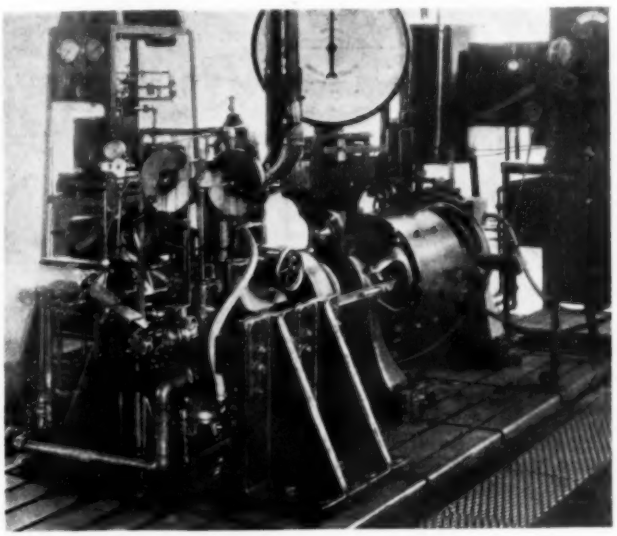


FIG. 1 SINGLE-CYLINDER ENGINE USED AS A TEST UNIT

production, but it is a convenient design from the standpoint of an investigator and can be considered as satisfying the requirements for a quiescent combustion chamber when used as such. To justify the investigation of the quiescent combustion chamber when design trends seem to show the increasing popularity of the use of air flow in combustion chambers, there are the examples of several sciences, notably biology, in which it is possible to learn much from the study of simple types before going on to the higher, more complicated forms. The simplest type of combustion chamber is naturally the quiescent combustion chamber, and the simplest injection orifice is the round hole. These two forms are the basis of this investigation, and the main series of tests were directed at the determination of a method for obtaining the optimum combination of these elements. In the course of this work some other investigations have been conducted on this test unit, but they may be considered as interesting digressions from the main investigation.

[1] Assistant Mechanical Engineer, National Advisory Committee for Aeronautics, Langley Field. The author was graduated from Pennsylvania State College, Department of Mechanical Engineering, with B.S. degree, 1917; First Lieutenant, Coast Artillery Corps, U. S. Army, 1917; Aerial Observer, U. S. Air Service (A.E.F.), 1918; Proof Officer, Aberdeen Proving Ground, Maryland, 1919; retired from active service, 1920; building construction, Washington, D. C., 1921 to 1927; Assistant Mechanical Engineer at the Langley Memorial Aeronautical Laboratory, November, 1927; head of Engine Research Section at Langley, April, 1929, to date.

Presented at the National Oil and Gas Power Meeting, University of Wisconsin, Madison, Wis., June 23–26, 1931, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

Apparatus

The test unit used in this work is shown in Fig. 1. It includes the engine connected to an electric dynamometer and the necessary auxiliaries and apparatus for securing the performance data. The parts of the engine are, as far as practicable, the same as those of the N.A.C.A. universal test engine, the main exception being the cylinder head, which is designated in our test series as N.A.C.A. cylinder head No. 4 and has a vertical disk-shaped combustion chamber formed between the heads of horizontally opposed intake and exhaust valves, as shown in Fig. 2. This cylinder-head arrangement makes it possible to have three alternative positions for the fuel-injection valve and the pressure elements of the various cylinder-pressure measuring and recording devices. Another feature of this cylinder head is the removable orifice between the cylinder and the valve chamber. The clearance between the piston crown and cylinder head was from 0.030 to 0.040 in. at top center.

The fuel-injection pump was the usual cam-actuated plunger pump with injection controlled by a bypass valve. The plunger displacement was much greater than was necessary for injecting full-load fuel quantities, and the cam gave the plunger a variable

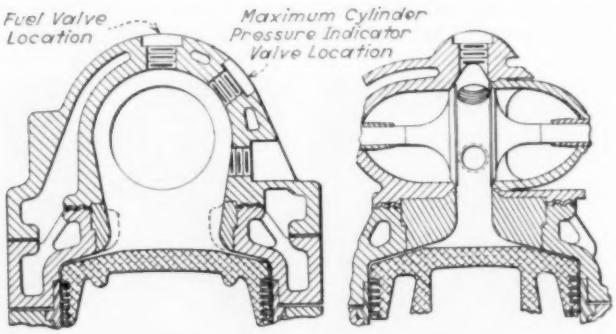


Fig. 2 Combustion Chamber

velocity throughout its stroke, so that by varying the settings for the closing and opening of the bypass valve, different rates of plunger displacement could be used for injecting the fuel. The pump was driven from the crankshaft through a reduction gear in which there was an adjustment for changing the angular relation between the crankshaft and the pump cam. Different fuel-injection valves were used, but they all were of the same general type, having spring-loaded stems hydraulically controlled. The pump and valve were connected by a steel tube 36 in. long and $^1/_8$ in. inside diameter.

The electric dynamometer was used to absorb the engine power and to motor the engine for starting and for the friction runs immediately after the power runs. The stop watch and revolution counter were actuated electrically. Fuel and air consumptions were determined by timing the displacements of 0.5 lb. of fuel and 80 cu. ft. of air. Water and oil temperatures (out) were maintained at 170 and 140 deg. fahr., respectively. The temperature in the air intake was 95 deg. fahr., except during supercharging.

Compression pressures and maximum cylinder pressures were indicated by a Bourdon spring gage in connection with a trapped pressure indicator for most of the tests, but toward the end of the investigation the balanced-diaphragm apparatus was used. Indicator cards were taken with the Farnboro electric indicator during the course of its remodeling at the Langley Memorial Aeronautical Laboratory, and the accuracy of the cards increased as the test work progressed. The cards presented may be considered sufficiently accurate, since the i.m.e.p. as determined from the area of the P-V diagram is within 5 per cent of the sum of the b.m.e.p. and the f.m.e.p., although the summation was used in plotting indicated performance curves. Any values presented may be considered average and readily reproducible.

Single-Stem Fuel-Injection Valve

The first idea which governed the design of fuel-injection nozzles for this cylinder head was that the sprays should not have penetration enough to reach the boundaries of the combustion chamber. It soon became apparent that restricting the sizes of orifices to reduce penetration also prevented enough fuel from being introduced into the engine in a sufficiently short time for high-speed operation even with undesirably high pressures in the injection system. The idea of restricting penetration was therefore abandoned, and a series of tests was started to determine, by actually varying the sizes of orifices systematically, the optimum size for each orifice in a seven-orifice nozzle.

The main orifice diameters were varied from 0.010 to 0.021 in., and best performance was obtained with the 0.018 in. diameter, although an inspection of carbon deposits on the piston showed that actual impingement started with the 0.012-in. orifices. Of course, some excess fuel is necessary to take care of the combustion air in the clearance between the piston crown and the cylinder head, but the surprising part of these data is that the power increases and the fuel consumption decreases consistently up to a diameter of 0.018 in., indicating that the distribution of fuel to the air is not impaired but actually improved by this impingement on the piston crown.

After the optimum size for the main orifices had been determined, the outside orifices were varied, and although the power increased up to a diameter of 0.012 in., it was decided to maintain these orifices at 0.010 in. diameter because the smokier exhaust did not warrant the utilization of the small increase of power available with the 0.012-in. orifices.

Enlarging the intermediate orifices decreased the power, and this decrease was rather hard to explain until spray photographs were taken inside a dummy combustion chamber. These spray photographs showed that the large outer sprays were hitting the combustion chamber walls and rebounding into the space which probably was not reached by the short intermediate sprays. Therefore it seemed quite likely that when these intermediate sprays were enlarged they merely put more fuel into air that was also being served by the outer orifices.

Even before these spray photographs had been taken a line of reasoning had indicated that it would be advisable to try a nozzle based on the proper mixing of the fuel and air by metering the fuel directly to the air and neglecting entirely any air flow, especially as work with air flow in a simulated combustion chamber had indicated that the actual velocities encountered were too low to affect the penetration and distribution of the oil sprays.

The basic assumptions on which this line of reasoning depends are that air flow is ineffective in this combustion chamber, that the coefficient of discharge of all orifices in a multiple-orifice nozzle will be the same, and that the pressure effective in causing the discharge will be the same for all orifices. These assumptions seem reasonable, and all factors determining the discharge of an orifice should be identical except the orifice area. If, then, these orifice areas are in the same proportion to the total discharge area as the volumes of air they serve are to the total volume of air served, the homogeneity of the resulting mixture of air and fuel is dependent upon the dispersion of the fuel in the spray and the shape of the volume served.

This proportional principle of course requires some orifice area as a starting point, and it was decided for the sake of comparison with previous data to adopt the two main 0.018-in.-

diameter orifices which seemed to be better established as an optimum than the other orifices. When calculating the air volumes to be served, the air in the clearance between piston crown and cylinder head was considered to be served by these main sprays. The spray angles as used in the previous nozzles were continued in these nozzles, and two more orifices were added on each end of the line, making a total of nine in one plane. There is nothing either in the performance data or in spray photographs to indicate that these orifices discharged enough fuel to make any difference in performance, and therefore their addition is considered an unwarranted complication. It is possible that when entering at an angle in the flow passage, as they do, they do not have the same effective discharge pressure as the other orifices.

The first nozzle in this C-series was built without the center, filler orifice, in an attempt to determine its value. This value was quite evident from the increase in the engine performance with the center spray and at the time was attributed solely to the fact that the greater part of the fuel in the main sprays traveled through the space they were supposed to serve and left but little fuel in the first few inches, thereby presenting the opportunity for the center spray of shorter range to serve the air in this space and improve the engine performance.

The work with the C-series of nozzles confirmed the proportional principle in a satisfactory manner; so it was decided to discard the original angular spacing and try to find out just how far it was necessary to follow this proportional principle for good performance. Spray photographs were consulted, and as the spray-cone angles for sprays under comparable conditions, except temperature, varied from 18 to 22 deg., an angle of 20 deg. was adopted, and a nozzle was designed which would have a total of 16 orifices in three planes, with an angle of 10 deg. between planes, as shown in Fig. 3.

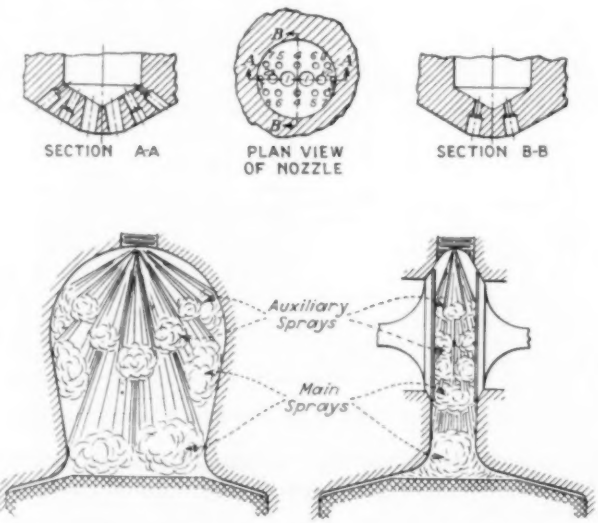


FIG. 3 PROJECTED SPRAY DISTRIBUTION, USING NOZZLE E-16

This nozzle was built as two separate nozzles, in which the orifices were drilled two at a time, and performance tests were made between the drillings. This method gave checked performance data which were desirable because of the complicated nature of the nozzles. These nozzles are denoted by the letter "E" and a number showing the number of orifices in the nozzle at the time of testing.

The performance data from the E nozzles showed rather conclusively that for this cylinder head the advantages gained by using more than six orifices were not commensurate with the complication of the nozzle. These tests determined the number of orifices to be used in future nozzles and simplified the work considerably.

During these tests the engine-operating characteristics indicated that the sprays were too close together, and about this time the results of the work on dispersion of fuel sprays at the Pennsylvania State College were published. Following the method of Dr. Schweitzer, the "boundaries of combustion" were laid out for the E-6 nozzle. Assuming this "boundary" to enclose the minimum space requirement for combustion, it was evident that the boundaries overlapped and that the sprays probably interfered with each other during combustion.

In view of these considerations it was desirable to obtain information on the effect on engine performance of the angular spacing of the fuel sprays, so another series of nozzles was designed in which the angle between the sprays was the only variation between the nozzles of the series. This series was characterized by the letter "F" and a number corresponding to the angle in degrees between sprays; thus the E-6 and the F-20 nozzles are identical nozzles. The proportional principle was not adhered to exactly for the F series, because, as the angular spacing changed, so did the volumes of air to be served by the different orifices, but the constant orifice sizes for all angles produced an average deviation small enough to seem negligible.

The results of the tests with these nozzles show very little difference between them so far as the performance curves are concerned, but observation of exhaust conditions and sensitiveness to controls led to the adoption of an angle of 25 deg. between sprays for further work with this cylinder head.

As a further check on the value of filler sprays, two 0.005-in.-diameter orifices were added in the center of nozzle F-29. This alteration brought the performance up to that of the F-26, and this improvement in performance advances the idea that the efficacy of the center jet in the C-series was in a measure, at least, due to the wide angle (30 deg.) between the main sprays.

The work thus far has shown that the number, size, and direction of the orifices in a multiple-orifice nozzle do not have critical values upon which good performance depends when used with a quiescent combustion chamber. Engine operation indicated that this combination is less sensitive to controls than an engine which uses air flow for distribution of the fuel. For example, the range of the injection advance angle from knock to miss is greater with the quiescent combustion chamber than with some test units using air flow in the combustion chamber.

SUPERCHARGING

These tests have, so far, neglected the total discharge area of the orifices in the fuel-injection valve nozzle as a definite part of the test program, but when supercharging tests were started with this cylinder head it was considered the proper time to investigate an increase in total orifice area, because with more air available for combustion it would naturally follow that more fuel should be supplied. The nozzles in this series are designated by the letter "H." The nozzles in themselves did not produce enough difference in performance to be of interest, although the oscilloscope showed that for the final H-4 nozzle the discharge orifice areas were definitely too large for the rest of the injection system and the main orifices maintained their discharge longer than the other orifices.

The significant features of these supercharging tests that can be taken from the performance curves are that the i.m.e.p. will be proportional to the weight of air available for combustion, provided that a suitable amount of fuel is distributed to this air. Therefore, from a superficial examination of performance curves (Fig. 4), supercharging merely extends the performance curves and does not essentially alter their characteristics. All this

could normally be expected, and it might seem that the tests produced nothing unusual.

However, if one looks into the conditions of engine operation, it is found that operation is considerably smoother when the engine is supercharged. The balanced-diaphragm, maximum-

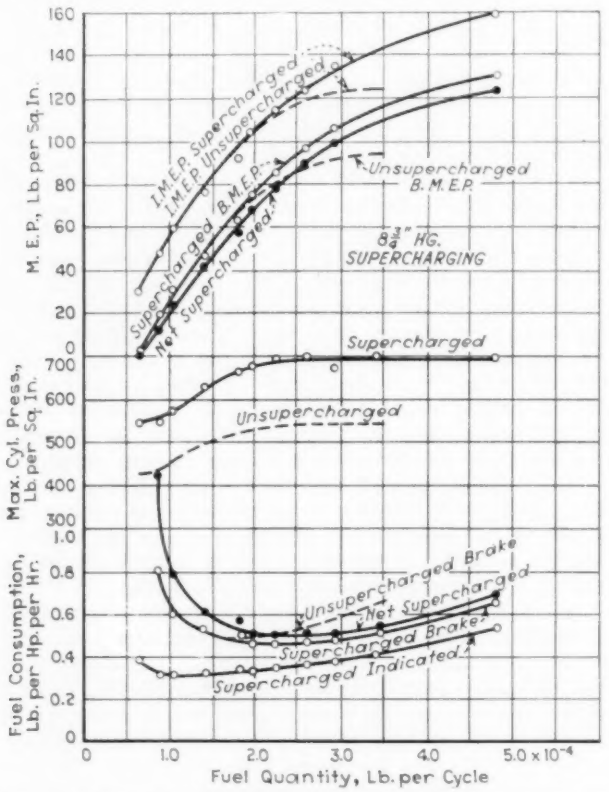


FIG. 4 COMPARATIVE PERFORMANCE CURVES, SUPERCHARGED AND UNSUPERCHARGED

cylinder-pressure indicator shows that the difference in maximum cylinder pressure between individual cycles becomes considerably less than for the unsupercharged condition, and this absence of variation of course makes the engine run smoother and allows higher injection advance angles to be used. In fact, with 9 in. of mercury supercharging pressure it was possible to peak over the power curve by advancing the injection until approximately a 10 per cent drop in power was recorded. This curve, as shown in Fig. 5, is quite flat, and in combination with the maximum-cylinder-pressure curve shows that the power gained by advancing injection beyond a certain point is too expensive in added stress on the engine parts.

This ability to peak over the power curve was not entirely due to the supercharging, but was in a measure due to the construction of the nozzle, which did not entirely conform to the specifications of any of the series mentioned, and will therefore be designated as nozzle X. For comparison, the performance of a nozzle with more nearly proportional orifices is shown, and these curves are parallel to the others, except the maximum-cylinder-pressure curve. The difference in these pressure curves indicates that higher rates of pressure rise may be achieved with homogeneous mixture of fuel and air and that an unbalanced mixture has a suppressing influence on both the rate of pressure rise and the maximum pressure attained.

The term homogeneous is used only in a comparative sense, as a study of dispersion data will show the futility of trying to obtain a truly homogeneous mixture when injecting fuel through a round-hole orifice nozzle into a quiescent combustion chamber. This improbability is reflected in specific fuel consumption values. As the time of injection was advanced the engine performance conditions diverged from those of the true quiescent combustion chamber because of the injection of an increasing amount of fuel into the air that still had some forced flow due to the motion of the piston.

From this work it appears that the quiescent combustion chamber in conjunction with a multiple round-hole orifice nozzle has inherent limitations that may prevent it from attaining the high performance which is ultimately expected of the compression-ignition, fuel-injection engine. But on the other hand it makes for a consistent engine and one which is not particularly critical, so that the combustion phenomena proceed in an orderly manner, and therefore these phenomena

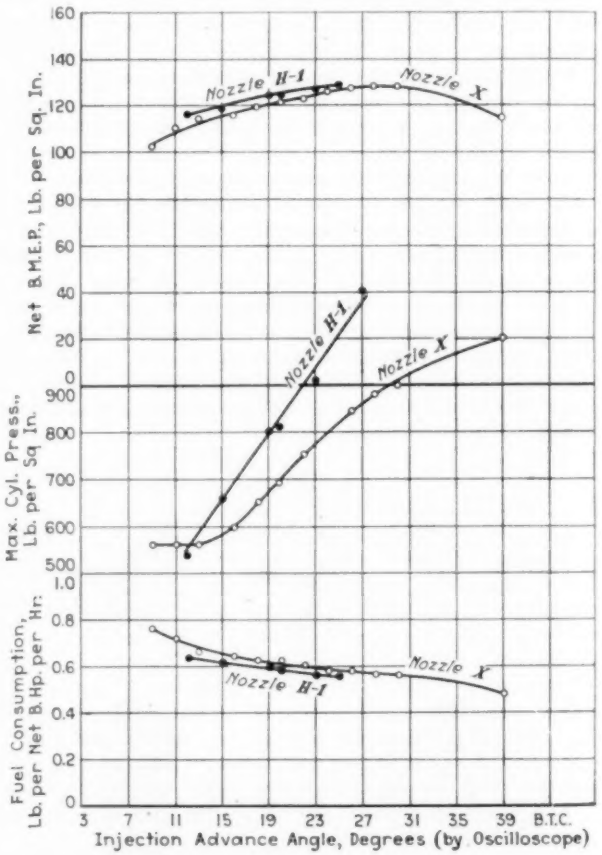


FIG. 5 COMPARISON OF PERFORMANCE WITH NOZZLES X AND H-1

can probably be studied and experimented upon to better advantage than in the more complicated forms of combustion chambers with air flow.

DOUBLE-STEM FUEL-INJECTION VALVE

One objection to the use of round-hole orifices for the injection of fuel is that the area is rigidly fixed, and within the range of pressures between valve-opening pressure and the maximum injection pressure, it is difficult to obtain much variation in the rate of discharge. For those who desire to control the rate of combustion by the rate of discharge this is a very definite handicap. An attempt was made with this engine to design a cam which would inject fuel at a theoretical rate determined from an ideal indicator card, taking into account the compressibility of the fuel, injection lag, and ignition lag. The results ob-

tained did not indicate that ideal combustion was taking place in the cylinder, so the problem was attacked from a different angle.

To produce a fuel-injection valve that would be more adaptable, the different means for varying the discharge area were investigated, and this resulted in the adoption of a multiple-orifice nozzle in combination with a fuel-injection valve in which two concentric stems uncovered two sets of orifices at different pressures in the injection system. The fuel-injection valve incorporating this feature is shown in Fig. 6, and although it is complicated in construction, it functioned satisfactorily. The only mechanical inconvenience with this valve was the necessity for lapping the flat seats occasionally to remove erosion marks, but this was by no means excessive.

The first nozzle D-1 tested with this double-stem valve had no homolog among the single-stem valve nozzles, and its performance was disappointing. To determine the cause of this decrease in performance a nozzle D-2 was designed homologous

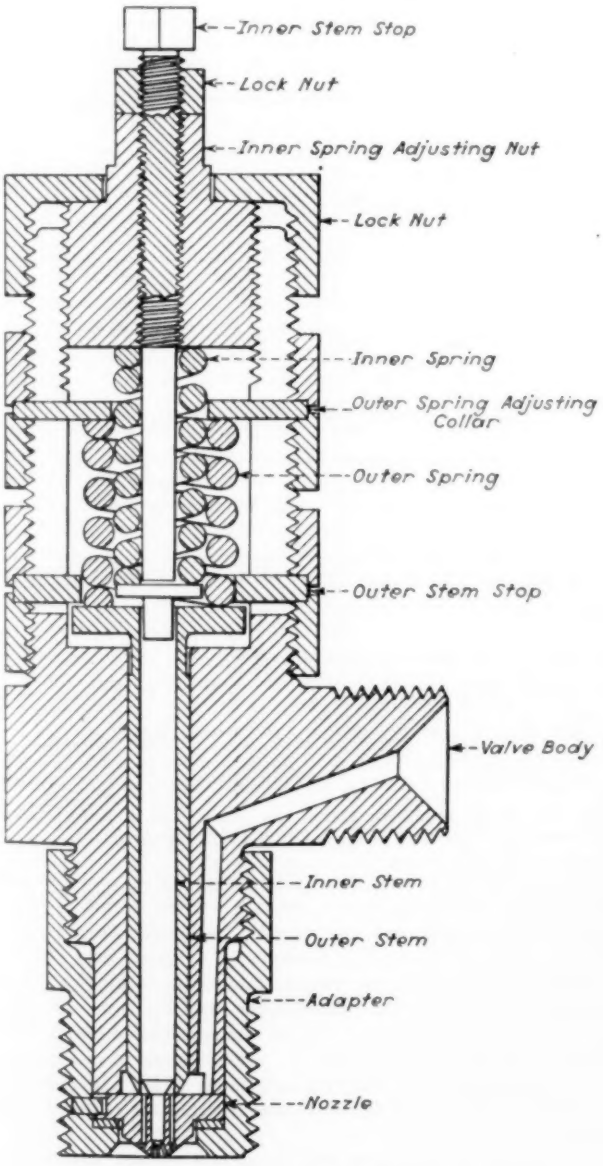


Fig. 6 Double-Stem Valve

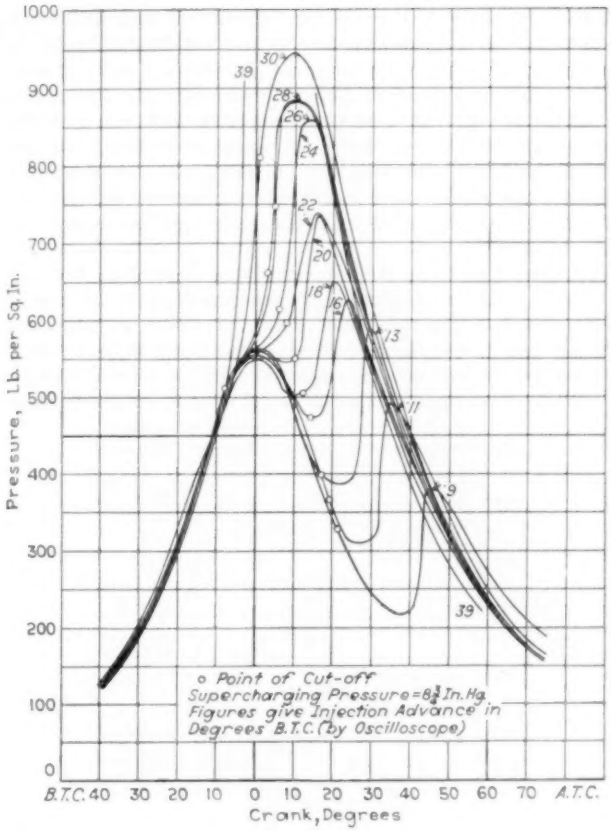


Fig. 7 Effect of Injection Advance on the Pressure Time Card

to No. 9 nozzle of the first single-stem series. This D-2 nozzle also incorporates the idea of placing the primary injection in the relatively cold outer portion of the combustion-chamber disk, and thereby should give an opportunity for determining whether there is any advantage to be gained by trying to preheat cooler sections. It is possible that these sprays in close proximity to cooled walls did not accomplish this purpose, but in the absence of proof as to whether the sprays heated the section or whether the section had a quenching action on the sprays it will be considered that this section did get some preliminary heating.

The opposite effect was obtained with the D-3 nozzle (homolog E-8) by directing the primary sprays down the central axis of the combustion chamber to create a super-hot spot between the valve heads through which the main sprays had to pass when the main injection was started.

Indicator cards were taken and injection rates determined with a receiver mounted on the flywheel of the engine. A coordination of these data with the usual performance data shows that the pressures in the engine cylinder can be controlled within limits by the rates of injection obtained with a double-stem valve, but the effect on the combustion process is usually deleterious, and the resulting m.e.p. is not so good as with the corresponding single-stem valve. One striking feature of the data is the apparently constant time interval of 0.0033 sec. between the first introduction of fuel and the start of pressure rise due to combustion.

Indicator Cards

This discussion has thus far considered only such information as may be gained through comparatively direct measurement

of the quantities involved. The analysis of indicator cards furnishes information which may not be as exact quantitatively as the usual performance data, but is none the less valuable in showing trends and tendencies which can be determined in no other way. An analysis of indicator cards taken during these engine tests not only checks many conclusions from the engine data, but also supplements them with additional information as to the manner in which the benefits indicated as possible by theoretical calculations may best be realized in actual engine practice.

The most interesting series of cards were those taken with the X nozzle in the single-stem valve corresponding to the engine performance in Fig. 5. These cards, as shown in combined form in Fig. 7, were taken with all engine controls in the same setting except for the variation of injection advance angle. They differ from the engine conditions of the unsupercharged tests

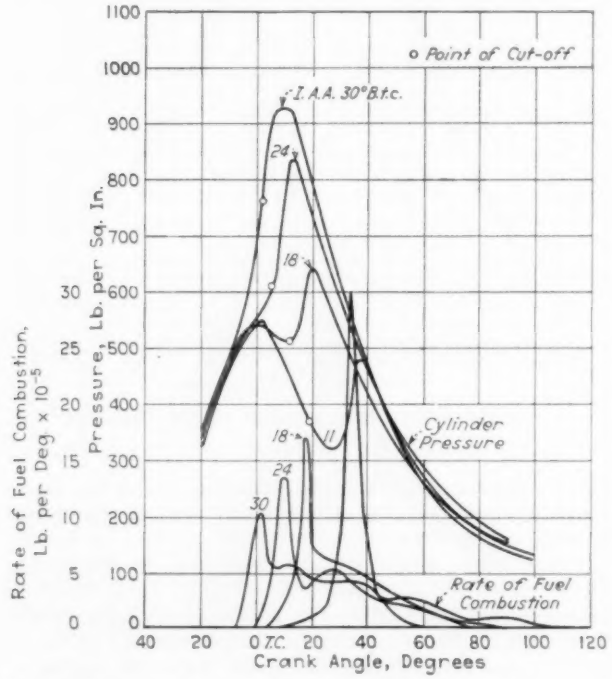


FIG. 8 CHANGE IN RATE OF COMBUSTION WITH ADVANCE OF INJECTION

in that the inlet air pressure is 8.75 in. Hg instead of atmospheric, and the inlet air temperature is 125 deg. fahr. instead of 96 deg. fahr. The pressure curves are differentiated by giving for each curve the apparent start of injection as determined with the oscilloscope when injecting into the atmosphere. The reference point shown by the circle on the pressure curve is the point at which the start of cut-off at the fuel pump affects the rate of injection. This point can be determined more definitely than the start of injection, so it was used as a reference point for determining time lags. The method devised by Schweitzer was used for the analysis of the indicator cards. The curves of rate of combustion in Fig. 8 show that the combustion process changes with injection advance angle and checks the observation of engine performance characteristics in that several knock ranges are passed through with the attendant change in rate of pressure rise.

Another point that is forcibly demonstrated is the importance of the start of ignition for smooth operation, and this, in turn, indicates that perhaps the supercharger air pressure is not responsible for the smooth operation during the supercharging tests, but that probably the increased temperature of the air was more responsible. To supplement this reasoning the calculated temperatures at the end of compression are shown in Fig. 9 for the usual range of initial pressure and temperature and of compression ratios.

The curve of rate of combustion in the case of the double-stem valve shows a remarkable tendency to follow the rate of injection for the first part of the curve, but in no case is the

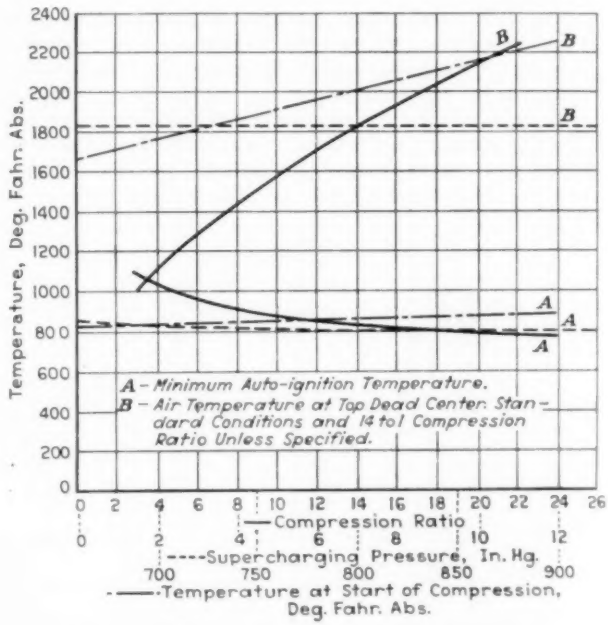


FIG. 9 TEMPERATURES AT END OF COMPRESSION

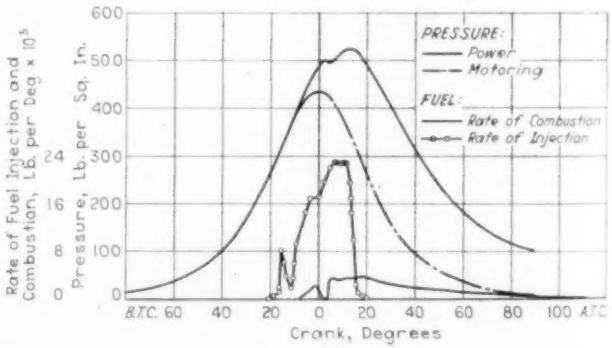


FIG. 10 PRESSURE TIME CARD AND RATES OF INJECTION AND COMBUSTION WITH DOUBLE-STEM VALVE

maximum as great as with a single-stem valve, and the last part of the combustion curve shows a lengthy period of afterburning which was indicated during the engine tests by red flame and black smoke in the exhaust. A typical rate of combustion curve for a double-stem card is shown in Fig. 10. It is possible that the higher temperatures resulting from the combustion of the primary fuel reduced the penetration of the main injection to such an extent that the whole scheme of distribution was disarranged with resulting rich zones which had difficulty in finding sufficient air for combustion within the required time limit.

A number of improvements in method suggest themselves, and further work with the double-stem injection valve may be

done at some future time, but at present it is thought that other simpler means of improving performance offer better opportunities and that perhaps a better understanding of the combustion process with the single-stem valve would be of considerable value in outlining any future programs for the double-stem valve.

The indicator cards presented here were not taken for the purpose of analysis, but were taken along with the performance tests mainly for a check on the pressures involved, and it is hoped that indicator cards taken in the course of a program intended for the sole purpose of studying the combustion process will yield information that will be of even more value for specific applications. This consideration brings up the question of comparison of data from different test units and the application of results obtained from one engine to another similar engine. The results obtained in this work are comparable and consistent within themselves, and the general effects noted during the performance tests will obtain in all quiescent combustion chambers to a greater or less degree.

The two great factors in obtaining improved performance for a compression-ignition fuel-injection engine are of course the control of distribution and control of combustion. For the distribution of the fuel to the air in the quiescent combustion chamber by means of multiple-orifice injection nozzles, these tests give a principle which may be followed in the design of the nozzles and show some of the difficulties encountered. For the control of combustion these tests give indications of the possibilities for control of rate of pressure rise through the control of rate of combustion and show that the rate of fuel injection is not particularly important in influencing the rate of combustion, and therefore the rate of pressure rise for high-speed operation.

This discussion has neglected entirely the effect of various factors on the physical parts of the engine, because in more than 800 hours of engine operation included in these tests there have been no failures or replacements due to the conditions of operation imposed upon the mechanism. Maximum cylinder pressures have usually been less than 800 lb. per sq. in., but in the supercharging tests the pressures encountered were slightly over 1000 lb. per sq. in. without mechanical troubles, so it is believed that an engine with a quiescent combustion chamber is entirely practicable for continued consistent performance.

Bibliography

The test work outlined in this paper is reported in detail in the following publications which give the necessary collateral references:

1 J. A. Spanogle and H. H. Foster, "Performance of a High-Speed, Compression-Ignition Engine Using Multiple-Orifice Fuel-Injection Nozzles," N.A.C.A. Technical Note No. 344, 1930.

2 J. A. Spanogle and H. H. Foster, "Basic Requirements for Fuel-Injection Nozzles in Quiescent Combustion Chambers," N.A.C.A. Technical Note (to be published), 1931.

3 J. A. Spanogle and H. H. Foster, "The Effect of Supercharging on the Performance of a Compression-Ignition Engine," N.A.C.A. Technical Note (to be published), 1931.

4 J. A. Spanogle and E. G. Whitney, "Investigation of the Effectiveness of a Double-Stem Automatic Injection Valve in Controlling the Indicator Card of a Compression-Ignition Engine," N.A.C.A. Technical Report (to be published), 1931.

5 A. M. Rothrock, "Combustion in a High-Speed Compression-Ignition Engine," N.A.C.A. Technical Report (to be published), 1931.

Discussion

C. L. Cummins.[2] The writer's experience in developing a high-speed Diesel engine confirms the author's conclusion that the quiescent combustion chamber is a consistent performer. The combustion chamber of the Cummins engine is not strictly a quiescent one, although it may be considered quiescent so far as the distribution of the main part of the fuel charge is concerned. Air flow is used only for a particular purpose directly under the fuel-injection valve, and the percentage of air involved is small.

The author has neglected to say much about spray penetrations, which are of considerable interest to the engine designer when laying out similar combustion chambers for different sizes of engine. These penetrations are especially important with our injection system, which gives a better atomized and more widely dispersed spray than it is possible to obtain with "solid injection" through a round-hole orifice.

The use of air flow in combustion chambers appears to have some interesting possibilities, but there is almost a complete lack of reliable information upon which to base the utilization of air flow. Experience has shown some of the advantages to be gained, but there have always been accompanying disadvantages which have kept us from using air flow as a means for distributing fuel in a combustion chamber.

Author's Closure

Mr. Cummins is quite right in pointing out that spray penetrations have been neglected, but as this work was done with only minor changes in the combustion chamber shape or dimensions, the question of spray penetration was considered of less importance than the factors which had been investigated at the time the paper was prepared.

In connection with Mr. Cummins' observations on combustion chambers with air flow, at the present time the author quite agrees with him in general, but wants to make it clear that, although this paper deals entirely with the quiescent combustion chamber, the work has not been abandoned on combustion chambers with air flow. It is hoped that further research may discover methods for making the benefits of air flow available without incurring prohibitive penalties.

---

[2] President, Cummins Engine Company, Columbus, Ind.

PME-53-1

Lubrication of Textile Machinery

By L. A. BAUDOIN,[1] NEW YORK, N. Y.

Cotton textiles require absolute cleanliness during manufacture and finishing, and all operations including the lubrication of the equipment are secondary to this requirement. Present designs incorporate modern developments which aid lubrication, and also provide a reserve supply of lubricant with ample protection to the fabric. Frequent application of minimum quantities of lubricant is accepted practice, but "freak" oils are not necessary. A list is included which indicates the frequency of application, cleaning period, and the product required.



LUBRICATION of textile machinery resolves itself to the consideration of groups of machines, each absolutely essential to the finished product, but performing their several duties as independent operations on the fiber, yarn, or cloth. To avoid complications, only cotton textiles are considered in this paper, for the reason that linen, woolen, carpet (combination cotton and wool), jute, silk, rayon, and knit goods each require independent handling from a manufacturing standpoint, and each process differs from the other and all from cotton. There are many similar features, many machines have a close resemblance, but the staple of wool varies from several inches to a foot or more, and silk and rayon are continuous.

Linen has characteristics that are not found in either, and knitting machines differ materially from any cloth-making equipment.

The care of textile machinery is one of the lessons that must be learned by actual experience with the equipment. The care includes lubrication, because the performance required of the intricate machines used in the textile industry can be attained only when friction has been reduced to a minimum. It is well known that this is not an actual achievement, and this paper points out a few of the numerous handicaps.

In the first place, many textile mills, long established, are using machinery designed and built some years before lubrication arrived at the stage of perfection forced by the demands of automotive equipment, and the fact that mill owners have not changed their machinery as rapidly as new developments along lubrication lines were advanced is not a reflection upon them at all, for the reason that in many instances such a move would require the scrapping of thousands of machines which are otherwise suitable for use.

In many instances where a small number of large capacity machines could be replaced to advantage it has been done, and the lubrication features of these new machines are in line with the very best practice today. For instance ball and roller bearings are replacing plain babbitted journals on fan shafts and all load-carrying bearings. Gear sets are well protected from dust and lint and so arranged that the cut gears run in a bath of oil, and in cases where first cost prohibits redesign to incorporate roller bearings, the ones provided are equipped with pressure-gun fittings and reservoirs for grease and provision to shield the bearing from the work so that the lubricant does not come in contact with the cotton fibers.

Throughout the cotton-manufacturing industry lubrication has been a secondary consideration, being forced into the near background by the absolute necessity of keeping oil and grease from spotting the goods. This care starts at the bale breaker and is vigilantly maintained until the cloth is shipped from the finishing plant.

Hand oiling of intricate machine parts requires a conscientiousness bordering on the superhuman to insure that each and every bearing and stud requiring it has an ample oil film. In this case experience alone is no safeguard, and when once passed by, the bearing, stud, or oil hole is soon passed again, and worst of all, the corresponding location on every similar machine is soon absolutely forgotten and neglected until excessive wear causes misalignment or actual breakage, contributing to lowered production. The seriousness of this condition is plainly shown by the accompanying illustrations which show worn parts picked at random. These parts may be readily identified.

Textile mills must have hundreds and in some cases thousands of certain machines, and one manufacturer regrets that the price at which the machines are sold does not allow equipping them with devices for holding an ample supply of oil and regulating its flow to the working parts. Oil holes and open bearings make up the oiling system on these machines. But the sky is clearing, because another manufacturer is now experimenting with one-shot oiling, and has built two large machines equipped with this system.

Passing from the spinning and weaving departments to enter that of cloth finishing, most of the lint and dust is left behind, and many of the machines are equipped with roller bearings or pressure-gun fittings and recesses for excess lubricant. Where gear sets are required they are well protected and contain ample lubricant. However, some of the cloth-handling machines are retained with their characteristic plain bearings.

Cleanliness is an absolute necessity and frequent cleaning periods have always been in order. Lint and waste allowed to accumulate on or near bearings and wearing surfaces cause endless trouble from wear because of the sand and grit present; so all machines, from bale breakers to cards, are brushed daily and thoroughly cleaned each week. At one mill the cans supplying the slivers are all allowed to run out together, and before full cans are replaced the back of the machine is thoroughly cleaned. The fronts of the slivers are cleaned every day.

The grit getting into the bearings wears them out rapidly, but the lint and fly which adhere to the outside of bearings and adjacent parts absorb oil readily and actually dry out the bearing by capillary action.

Mill owners and agents naturally consider cost of production. Operators and machine designers consider lightening the opera-

---

[1] Assistant Supervising Engineer, Sinclair Refining Co. For eight years Mr. Baudoin was connected with various plants as assistant engineer and in 1910 became master mechanic for the Edwards and Loomis Co., Chicago, Ill. While with that company he developed several new material-handling and labor-saving systems for increasing production. He joined the Sinclair Refining Co. as lubrication engineer in 1919.

Contributed by the Lubrication Engineering Sub-Committee of the Petroleum Division and presented at the Textile Sessions of the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

TABLE 1 TEXTILE MACHINERY WITH FREQUENCY OF LUBRICATING AND CLEANING

Equipment	Lubricating frequency	Cleaning periods	Product No.	Remarks
Bale breakers, plain bearings	Every 2 days	Each week	1	
Lattice feed table, plain bearings	Every 2 days		1	
Openers:				
Reserve box automatic feeders:				
Plain bearings	Every day		1	
Ball bearings	Every 6 mo.		8	
Creighton vertical opener:				
Plain bearings	Every day	Each week	1	
Step bearing	Every day		1	The step bearing of this machine is surrounded by water jacket which must be kept filled
Vertical openers, ball bearings	Every 6 mo.	Each week	8	Provided with pressure-gun fittings
Beater type, ball bearings	Every 6 mo.	Each week	8	
Condensers, oil tubes to bearings	Every day		1	
Exhaust fans, ring-oiled bearings	Every month		1	
Distributing conveyors, plain bearings	Every day	Clean boxes every day	1	Gears are well protected from dirt
Gear boxes	Every month		1	
Combination opener:				
Breaker picker	Twice a day	Every 6 mo.	1	
Fan bearings (roller)	Every 6 mo.		8	
Breaker lapper, plain bearings	Every day	Every 6 mo.	1	
Beater roller bearings	Every 6 mo.		8	
Evener cones roller bearings	Every 6 mo.		8	
Lap racks roller bearings	Every 6 mo.		8	
Finishing picker	Twice a day	Every 6 mo.	1	Draft begins
Lapping picker, plain bearings	Every day	Every 6 mo.	1	
Roller bearings	Every 6 mo.		8	
Waste pickers:				
Willow, roller bearings	Every 6 mo.	Every 6 mo.	8	
Plain bearings	Every day		1	
Card and picker waste cleaners	Every day	Every 6 mo.	1	
Roller bearings	Every 6 mo.		8	
Roving and hard waste cleaners	Every day	Every 6 mo.	1	
Carding machines, cylinders	Every day	Every day	1	These bearings and many others are kept filled with grease for safety, oil is used as well every day
Licker-ins	Every week	Every day	1	
Doffers	Every week	Every day	1	
Slow moving parts, bearings and gears	Every day	Every day	1	
Comb boxes	Every week		2	
Sliver, front and back rolls	Every day	As cans run empty clean back of machine	2	
Top rolls, gears	Every day	Fronts each day	1	
Ribbon laps, plain bearings	Every day		1	
If roller bearings	Every 6 mo.		8	
Combers, cylinder shafts	Every day	Twice a year	1	Many plain bearings of this machine are provided with waste pocket to retain supply of oil
Head end gears and cams	Every day		1	
Fan, plain bearings	Every week		8	
Roller bearings	Every 6 mo.		8	
Steel rolls	Every day		1	
Detaching rolls	Once a week		3	These rolls have removable bushings and oil is applied by hand after the bushings are removed; see text
Guide arms	Twice a week		1	
Drawing frames:				
Fast parts and driving gears	Every day	Steel rolls scoured 1 mo.	1	
Lower rolls	Every week	Leather rolls each week	8	
Top rolls	Every day	Clearer washed each hour	1	
Head and foot end gears	Every day		1	
Newer equipment: roller bearings	Every 6 mo.		8	
Slubbers				
Speeders				
1st intermediates				
2d intermediates				
Jack frames				
Fine moving frames				
Bobbing gears and spindles	Every doff		1	
Spindle wells	Every month		5	
Plain bearings and gears	Every day		1	
Bolsters	Every week		1	
Front rolls	Every day	Every week	1	
Middle and back rolls	Every week[1]			
Top rolls	Every week		1	
Bottom rolls, wind back and builder gears	Every 2 wks.		1	
Differential motion	At cleaning period	Every 6 mo.	1	
Spinning:				
Mules:		Inspection once per year		
Spindles	Twice a day		4	
Overhead drives	Every day		1	
Live head, back head	Every day		1	
Cylinder bearings, stud rollers	Every day		1	
Back shaft, fallers	Every day		1	
Carriage wheels, band pulleys—leather rolls	Every day		1	
Ring spinners:				
Spindles, old type	Every 2 wks.		4	
Spindles, new type	Once a month		4	
Cylinder shafts, old	Twice a week		1	
Cylinder shafts, new (ring)	Once a month		1	
Roller bearing	Every 6 months		8	
Head gears, old	Once a week		8	
Head gears, new	Worm runs in bath		1	
Front steel roll	Every day		1	
Back steel roll	Every week		1	
Top and middle rolls	Twice a week		1	
Twisters, spindles, old	Each week		5	Run wet
Spindles, new	Each month		5	
Cylinder and roll bearings	Every 2 days		1	
Rings	Every doff		9	
Head gears, studs	Every day		1	
Fancy twisters, same as above except that they run without water				
Spoolers, spindles, old	Once a week		1	
Spindles, new	Once a month		1	
Cylinder shaft	Once a week		1	
Winders, drive shaft	Once a month		8	
Cam winding motion	Once a month		1	
Universal winders	Once a week		1	Run in oil bath well protected from lint
Reeler, shaft bearings	3 times week		1	
Traverse motion	3 times week		1	
Swing-gate bearing	Once a month		8	
Warper, cylinder	Every week		1	
Beam arbors	Every day		1	
(When roller bearings)	Every 6 mo.		8	
Balling attachment	Every day		1	
Floor stand	Every day		1	
Quilling machines:				
Spindles	Every month		4	
Cylinder shaft, studs, gears	Every day		1	
Universal filling winder shaft	Every week		1	
Cam winding motion	Every week		1	
Slasher cylinder bearings	3 times week		6	Steam rolls
Balance of machine	3 times week		1	
Weaving:[2]				

[1] When front rolls are cleaned.

[2] Copied from manufacturers' bulletin.

TABLE 1 TEXTILE MACHINERY WITH FREQUENCY OF LUBRICATING AND CLEANING *(Continued)*

Equipment	Lubricating frequency	Cleaning periods	Product No.	Remarks
Pick cams and balls				
Pick shaft bearings; crank arms; crank-shaft bearings; rocker shaft boxes	Every day		1	
Cam-shaft bearings, auxiliary shaft boxes; harness cams; idle gear stud; treadle roll pins; filling cams and followers; protector-rod bearings; friction-pulley parts	Every 2 days		1	
Treadles; shuttle feeler bushing; oscillator cam follower; transferer; other shuttle and knock-off parts	Every 3 days		1	
Loose pulley and shipper; lay guides; fork slides; starting-rod bearings; take-up parts; spring shaft; let-off parts; beam and whip roll bearings; harness roll bearings; Lacy top; parallel stud and spring; bobbin disc; pawls and support	Once a week		1	
Cut gears; parallel tongue; temple thread cutter; temple bar; cutter spring and stud; shuttle feeler cam path	Once a week		8	
Occasionally grease bemble	Once a week		8	Very small quantity required
Drop of oil to filling motion shipper bushing	Once a week		1	
L. O. internal gear and pinion, and path of nub of controlling lever link	Once a week		8	
Jacquard attachments	Once a week		1	
Brushers (horizontal and vertical), fast-moving parts	Every day		1	
Slow-moving parts	Every week		1	
Inspection machines	Every day		1	
Folders	Every day		1	
Presses (screw)	Every week		1	
Cloth-finishing machinery				
Singers, plain bearings	Twice a day		1	
Roller bearings	Every 6 mo.		8	
Washing; shafting	Once a week		1	
Bleaching:				
Kier—piling mach. gear box	Every 2 wks.		8	
Squeezers: large bearings	As needed		10	
Other bearings	Once a day		1	
Drier, pressure gun fittings	Once a week		8	
Plain bearings	Once a week		8	
Winders	Twice a day		1	
Shearing machines: plain bearings	Twice a day		1	
Roller bearings	Every 6 mo.		8	
Shear knives (edges)	Twice a day		7	Saturated felt
Float thread shearing machine	Twice a day		1	If pressure fittings provided use grease no. 8
Loop cutters	Twice per day		1	
Unrolling frames	Every day		1	
Railway sewing machines	Every day		4	
Spreader rolls	Every day		1	
Mercerizing frames, wash box	No oil used			
Traveling chain	Caustic soda acts as lubricant			
Diastoforing: frames	3 times day		1	
Scourers	Twice a wk.		10[a]	
Dyeing: jig	Once a week		8	
Roper	Once a week		8	
Drier (pressure fittings)	Once a week		8	
Slop padders	Once a week		8	
Indigo dyeing rolls	Every day		1	
Shafting	Every day		1	
Printing machines: cylinder bearings	Every day		6	
Offset rolls	Every week		10[a]	
Inking rolls, etc.	Every day		1	
Dry cams	Every day		6	
Steamers (pressure fittings)	Once a week		8	
Ager (pressure fittings)	Once a week		8	
Soapers: ropers	Once a week		8	
Soapers: open	Once a week		8	
Stretcher drier, pressure fittings	Each week		8	
Tenter frames: drive ends (sight feed cups)			1	
Traveling chains	3 times week		1	
Gears	Every week		8	
Calenders, main rolls	Twice a week		10[a]	Steam.
Balance of machine	Twice a day		1	
(Some calenders have water-cooled bearings.)				
Cloth folding machines	Every day		1	
Cloth guides (automatic)	Every day		1	
Cloth winders, measurers	Every day		1	
Trade marking machines	Every day		1	
Inspection and trimmer frames	Every day		1	
Inspection perch	Every day		1	

[a] This hard grease may be obtained in block form and applied to the bearings on a perforated metal plate previously fitted to the bearing. The journal heat allows sufficient feed from grease cake through the perforations. Grease should be selected which will not char or plug up perforations.

tion, convenience of handling, cleanliness, and adaption to high speed when any new equipment is contemplated. Lubrication is still secondary and must give way completely to cleanliness.

Lubricant manufacturers must work out the problem with existing equipment, but are insisting that, where necessary, future designs incorporate provision for an adequate supply of the proper oil to suit that particular condition.

Cleanliness is paramount, and so freak oils have found their way into the textile industry to the extent that one of the prime requisites is that it *does not flow* at room temperature. An-

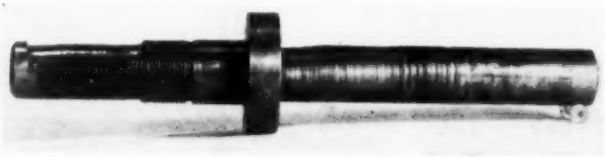


FIG. 1 STUD FROM SPEEDER HORSE HEAD

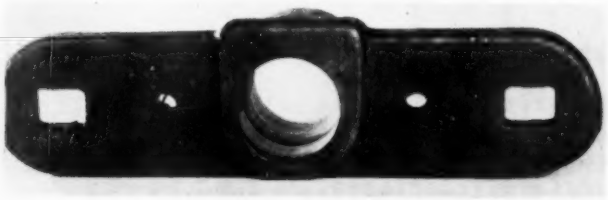


FIG. 2 BOTTOM ROCKER SHAFT FROM LOOM



FIG. 3 GEAR FROM SLASHER

other condition of combination high pressure and hot bearing is met by removing the bearing cap entirely and replacing it with a piece of leaf tallow or suet. That serious trouble has not occurred is simply that the journal speed is slow.

The oils actually required for textile-mill lubrication are comparatively few in number, their characteristics and particular conditions of service being as shown in Table 2.

High-grade straight mineral oils should be chosen where possible for textile-mill service, because when well refined their characteristics do not change in service. They do not tend to form gummy residues nor become rancid. In some special cases such as wet twisters, water-resisting compounds are neces-

TABLE 2

Oil	Flash, minimum	Viscosity: At 100 deg. fahr., maximum	Viscosity: At 210 deg. fahr., minimum	Compound
No. 1	330	265	45	None
No. 2	380	770	59	None
No. 3	405	390	55	High film strength
No. 4	330	78	36	None
No. 5	355	105	38	None
No. 6	525	...	150	None
No. 7	330	105	38	High film strength
Grease				
No. 8				Clean non-drying grease approximately 190 deg. fahr. melting point, having approximately 280 worked penetration. Oil portion composed of highly filtered oil of approximately 300 viscosity.
No. 9				White grease insoluble in water of medium consistency.
No. 10				High melting point of approximately 400 deg. fahr. Oil portion composed of cylinder stock.



FIG. 4 STUDS FROM JACQUARDS AND FANCY LOOMS

sary, and in the case of shearers where the knives run steel to steel a special compound is necessary. Calenders, dry cans, and printing machines running at high temperature and high bearing pressures require a filtered oil of high viscosity. Other bearings subject to high pressure, but slow speed, require a special heavy grease that will feed slowly and not carbonize under the temperature or dry out in service.

The list of textile machinery (Table 1) for manufacturing and finishing cotton cloth is given for completeness. The key numbers given with the lubricant indicate the particular grade to use. The application period is given as a guide only, but conforms to actual experience in large plants manufacturing and finishing cotton cloth. The periods shown for looms were copied verbatim from one manufacturer's bulletin.

The author wishes to thank Messrs. F. K. Hendrickson and S. H. Hillard, of Whiting Machine Company; W. H. Chase, Draper Corp.; L. A. Latham, Crompton & Knowles Loom Works; E. H. Marble, Curtis and Marble Company; C. F. Broughton and W. O. Buzzell, Wamsutta Mills; and Arthur Hirsh, American Print Mills, for their kind cooperation and aid in offering details for this paper.

Discussion

D. V. WATERS.[2] The writer has had no experience with the lubrication of textile machinery. It would seem, however, that

[2] Machine Design Division, Western Electric Co., Chicago, Ill. Mem. A.S.M.E.

experience gained with machinery which insulates wire with cotton or silk could be, to some extent, applied.

The author has mentioned certain places on machinery difficult to lubricate because any excess or leakage of lubricant is detrimental to the product. If loads and speeds are not too high, it is perfectly feasible to use anti-friction bearings made from one of the stainless steels. Such bearings are packed with light grease at assembly and thoroughly sealed against the entrance of foreign matter, after which they are run indefinitely with no attention whatsoever. We have found that these bearings give no trouble during the period between machine overhauls. It is believed that the use of such constructions will, in many cases, eliminate many of the problems connected with getting oil into bearings and its retention thereafter.

Where lubrication facilities must be provided, wick-feed oil systems appear to possess certain advantages, as the oil is filtered by its passage through the wick and a uniform and not too copious amount of oil is fed. Careful proportioning of the length and diameter of wicks, oil-level height, etc., is required in any satisfactory wick-feed oil system.

The company with which the writer is connected has conducted laboratory tests of various commercial feeding systems. These tests have indicated that such systems work best with oils coming within relatively narrow range of viscosities.

C. H. Ramsey.[3] To that portion of the paper having to do with the lubrication of old-fashioned sleeve-type bearings, stud-supported idle gears, sprockets, etc., illustrations of which have been submitted, no reference is made other than to commend the suggestions therein contained.

Earnest emphasis of the development of anti-friction bearings, however, is believed to be very appropriate. There is a constantly increasing and more insistent demand for ball and roller bearing equipped textile machinery, not only because of its contribution to the diminution of power consumption, but also because they lend themselves so admirably to complete enclosures, with ample capacity within the enclosures for lubricant reservoirs.

Complete enclosures for gearing of all types is also being more and more resorted to, not only because of the safety factor, but also because of the cleanliness, protection of the gearing from dirt, grit, etc., and also because the enclosures provide lubricant reservoirs tending toward a more quiet action of the gears and materially preserving their wearing qualities.

The writer thinks that two very important natural forces often used in the machinery field and yet again too often neglected are the action of centrifugal force and the force of gravity. They are utterly and always dependable and have wide possibilities of application in the field of lubrication.

A notable example of this will be found in the writer's patent (1,778,432, October 14, 1930, entitled "Lubricating System"). We have made practical application of this patent in our speed-reducing transmission, and it has worked out in full accord with our expectations. In this example, a disk is mounted on the highest speed shaft, which penetrates the box, which, in action, distributes the lubricant over the entire interior of enclosure, deluging the surface of the enclosure, the action of gravity of course causing it to drain back into the common reservoir in which the disk dips and, in passing, it percolates through all the bearings and gearing, lubricating them thoroughly.

Another familiar detail of textile machinery is the friction-plate method of securing variable speed, which still finds a wide range of usefulness and, indeed, is very efficient and dependable if properly designed and strategically mounted. Years of experience have quite conclusively demonstrated the most useful and dependable type of thrust bearing (to keep the friction disk in contact with the friction wheel) to be two steel pins of glass hardness running against each other. The one objection to this has been the difficulty of keeping them properly lubricated, the heat generated being so great at times as to draw the temper from the steel points and, hence, ruining them. This difficulty has been entirely obviated by enclosing these two points within a housing and constructing in the upper portion of the housing a suitably designed funnel, the lower aperture of which is trained to drip the contents of the funnel on the contacting points of the steel thrust bearing. A disk dipping in the lubricant, the lower half of the enclosure forming a suitable reservoir therefor, deluges the complete interior of the housing with lubricant all the while the machine is in motion. A generous supply of this deluge finds its way into the funnel, providing a continual stream of lubricant for the thrust bearing all the while the machine is in motion. This has worked out admirably in practice. We have made applications of it and have made application for patent thereon.

It is of course obvious that the use of gravity and centrifugal force in the distribution of lubricant entails the use of fluid oil. Experience has indicated to us that fluid oil is superior if it can be properly confined within its enclosures, so as to preclude the possibility of discharge, so that fabric being handled on textile machines, may be kept immaculate.

M. A. Golrick, Jr.[4] The subject of lubrication is generally accepted with a nod of the head and then dismissed from the conscious mind until forcibly reminded by the failure or more often the lack of lubrication at some point. If the subject were of a controversial nature, it would be constantly in mind and freely discussed. It is the lassitude of every one who should be vitally interested that makes lubrication a secondary instead of a primary issue.

The author mentions the difficulties of old machines and the attention required to lubricate by hand these old machines. However, it is believed he could emphasize this condition as many plants continue to use machines which were designed for heavy duty and long life. Due to their inherent designs, these machines are the "perpetual or everlasting type," which seem to live forever, yet are a constant source of trouble. In most cases this is due to the increasing increments of speed which are added one after another. It seems impossible to run many of these machines indefinitely, yet this is apparently the policy that is maintained for many reasons. In most cases the reason is that the cost of new machines is too much for an industry which has labored so long under financial stringency. On the other hand, the constant demand for increased speed does not allow a stable lubrication condition to be maintained.

The schedule presented on frequency of lubrication and cleaning is worthy of special attention, for this forms the basis of an admirable schedule, though the writer would caution against its adoption in whole without careful consideration. In one finishing plant, which had considerable machinery of the type requiring constant lubrication attention, a complete lubrication chart had to be made for each machine. The chart was in minute detail for each machine and showed the total number of points to be lubricated, so that new employees and also old employees would be well informed as to each machine's requirements. In passing, it might be well to mention that a 100 per cent factor of safety was used to set the frequency period, and this proved to be an ample safeguard. This schedule was an absolute necessity in this case, as repeated failures due to lack of lubrication were not corrected until the schedule was adopted.

Now, machinery is generally well designed to require only a

[3] President, Morrison Machine Company, Paterson, N. J. Mem. A.S.M.E.

[4] Dutchess Bleachery, Inc., Wappingers Falls, N. Y.

minimum of attention from the operator, and the author has covered this phase well. His statement that manufacturers must design to meet lubrication requirements is one that is being accepted by most designers. The subject must be constantly kept to the fore by lubricating engineers as their special duty toward industry.

SAMUEL CABOT.[5] To be effective, a lubricant must interpose a smooth, mobile film between the two moving surfaces, thus keeping them apart. Lubricants of high viscosity are required where the film has to withstand great pressures between the moving surfaces. For light-weight parts, low viscosity lubricants are preferred because high viscosity means increased internal friction or friction between the molecules of the lubricant.

An ideal lubricant is one which would have the quality found in high-viscosity liquids of resisting the squeezing-out action caused by the pressure of one bearing on another bearing, and would at the same time have the low internal friction found in liquids of which the molecules are small and thus have low viscosity.

Certain products called colloids which may be introduced into lubricants are attracted by the metal surfaces of the bearings. They are adsorbed to each of the bearings quite strongly in a thin film, said to be the thickness of one molecule. Attached, but less strongly on this film, is a more loosely grouped layer of these, much like minnows around a dead clam. These large molecules are not actually dissolved in the lubricating liquid, but in colloidal solution in it. They are in constant motion and can be seen swarming under an ultra-microscope. When they come close to the metal surface, they are attracted to it, become adsorbed to it, and lose their motion, concentrating largely near the metal surfaces.

These colloids impart to the lubricant some of the ideal qualities of high viscosity and low internal friction. The reason for this is that colloids being on the metal surface and adsorbed by it are very difficult to squeeze out, and as they have a strong affinity for the lubricant, they prevent it from being squeezed out. Thus a lubricant of somewhat lower viscosity may be advantageously used where high viscosities have been the rule. They also tend to smooth any microscopic roughness on the metal bearing surface.

The qualities the writer has been trying to describe may be briefly termed "oiliness or slip." It is what a small amount of the colloid, called soap, gives to water, or mud gives to an asphalt road. The best colloids to use with lubricating oil are those which are most closely related to the oil: They are called iso-colloids, which means that they are of the same family as the liquid they are in. From 5 to 20 per cent of a colloid of this kind added to a lubricant will make it more slippery, less likely to squeeze out of a lower cold test, and will maintain a lubricating film longer after oil feed has been shut off.

[5] Wilbur & Williams Company, Boston, Mass.

PME-53-2

New Pipe-Still Design

By LUIS DE FLOREZ,[1] NEW YORK, N. Y.

Methods and apparatus for applying heat to oil constitute the essential features of modern refinery practice. The tabular oil heater has developed rapidly since adequate turbulence can be obtained in the tubes by high velocity and the surface can be arranged to give comparatively even distribution of heat.

The author describes a new design of still which does away with the conventional brick-lined combustion space and permits absorption of heat to begin immediately after its generation. This has led to an arrangement of heating surface where the flame from the burner is exposed directly to what may be considered as an excess of heating surface. With this arrangement the flame heats the tubes by radiance, and the tubes cool the flame.



USE of the tubular oil heater in refining has increased rapidly in recent years. In its simplest form it consists of a box-like furnace containing a pipe coil through which the oil is pumped continuously, and heated directly by gas or oil burners located in the lower part of the structure. A study of the heat distribution shows that the bulk of the work of the heater is actually accomplished by the comparatively small section of coil at the fire. Under these circumstances, it is evident that the capacity of the heater is limited to the maximum safe loading on this section of the coil, since any endeavor to increase the work of subsequent sections by increasing the fire would result in burning the tubes in the already fully loaded section. For a time it was believed that tube failures were caused by the effect of radiant heat, and designs were made with shielded tubes in an endeavor to avoid the difficulties encountered. Still later, however, it was realized that the transfer of heat by radiance was desirable, if properly controlled, which led to the design of the more modern tubular heaters where the heating elements are generally divided into two sections, the first receiving heat mainly by radiation, while the second is heated by direct contact with the flowing stream of product of combustion.

The problem of transferring heat to oil differs materially from that of transferring of heat to water in view of the wide boiling range of crude oils and distillates and the danger of depositing coke on the inside of the tube if the transfer of heat is too rapid. It is known that, with adequate velocity, transfer rates as high as 10,000 or 12,000 B.t.u. per hr. per sq. ft. of heating surface are possible and these rates are actually being obtained in practice on certain portions of the surface in pipe stills. With the usual design of heaters, however, the average rate of transfer for the whole apparatus is much less, usually of the order of 3500 B.t.u. per sq. ft., since the load is not evenly distributed throughout the heating surface. More even distribution has been sought by the use of recirculating flue gases and the wasteful practice of employing excess air in combustion.

One of the chief aims of the design lies in increasing the average transfer rate for the total surface rather than to increase the maximum safe loads per sq. ft., although this may be found possible later. This condition can be approached by transferring the bulk of the heat, and in some cases all of the heat by radiation, and arranging the absorbing surface in such manner that it will receive as nearly as possible an equal portion of the heat radiated. The transferring of heat from the flame direct to the absorbing surface by increasing the flame temperature, caused by reducing the excess air to a minimum and preheating the air required, permits not only an even distribution of heat to the absorbing surface but is a simpler and more economical method of utilizing fuel.

Perhaps the most radical departure found in the new design is the elimination of the conventional brick-lined combustion space, to permit the absorption of heat to begin immediately after its generation. This has led to a new arrangement of the heating surface where the flame from the burner is exposed directly to what might be considered an excess of heating surface. With this arrangement the flame heats the tubes by radiance, and, conversely, the tubes cool the flame. Thus a balance can be struck between these two factors which results in limiting the maximum temperature to which the tubes may be subjected in service. It is thus possible to operate the heating surface under conditions known to be safe by former practice, and at the same time to burn fuel with high flame-burst temperatures.

In general terms, the vertical tube heater comprises what may be termed a radiant section and an economizer section. The radiant section is constructed in the form of a vertical cylindrical shell, the walls of which are lined with vertically disposed tubes, symmetrically spaced with respect to the center of the cylinder. A burner admits fuel at one end of the vertical chamber, and the products of combustion are discharged at the other end and carried to the economizer section through a suitable flue. The economizer section may consist of a more or less conventional tube-still arrangement for the tubes or may, in some cases, be an air heater which would replace the tube bank. The firing may be effected downward or upward according to the duty intended.

Fig. 1 shows a downshot type with the economizer section disposed vertically and the horizontal flue connection between the two sections continued into the base of the stack to provide an outlet for emergency.

The vertical section is formed by a cylindrical steel shell sup-

[1] Consulting Engineer. Mem. A.S.M.E. Luis de Florez was born in New York City in 1889. He obtained his school education in France and Spain, and was graduated from Massachusetts Institute of Technology, with the degree of B.Sc. in Mechanical Engineering in 1912. Upon graduation he entered the aviation field, in design, construction, and operation of airplanes, with Burgess Company and Curtiss Company at Marblehead, Mass. In 1913 he went with W. A. Hall, as chemical engineer; later he was chief engineer of the Motor Petrol Company. In 1915 he went with the Anglo-Persian Oil Company designing plants for the Russian and British Governments for the manufacture of toluol and benzol from petroleum, returning to America in 1916, designing, erecting, and operating cracking plants for the Texas Company at Bayonne, N. J. During the war he was head of the Instrument Division of the Aviation Section, Bureau of Construction and Repair, U. S. Navy. In 1919 he was engineer for Cochrane, Harper & Co. and consulting engineer for New England Oil Co. and Invincible Oil Co. and in 1923 chief engineer of New England Oil Company. Since 1923 he has been a consulting engineer.

Presented at the Metropolitan Section Meeting, New York, N. Y., April 16 1929, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

ported on columns and provided with a lining of insulating brick and firebrick. The upper portion of the shell is provided with an angle section turned inward which supports the weight of the tube sheet and tubes. This is allowed to slide on the surface of the supporting section to compensate for expansion.

The tube sheet itself is composed of a number of sectors, usually of cast metal, rigidly connected to an inner ring made of boiler plate, uniting them into a continuous circular shape. The tube

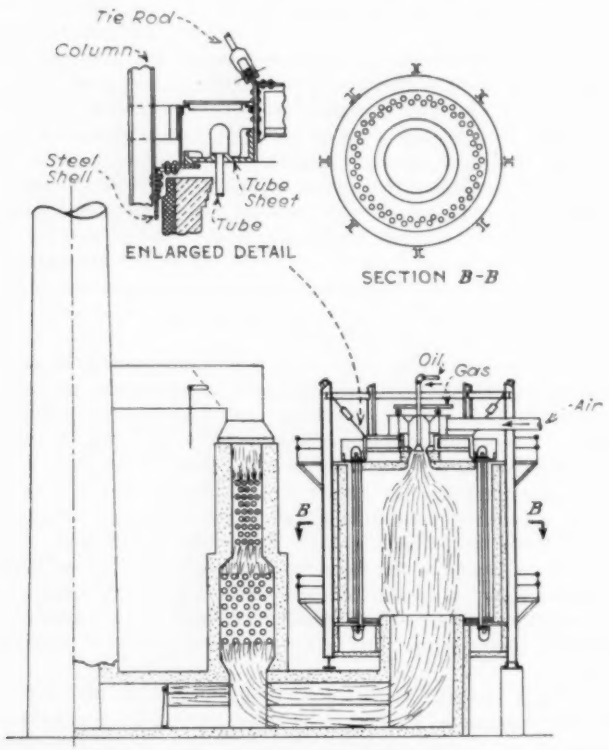


FIG. 1 DOWNSHOT TYPE WITH VERTICAL ECONOMIZER SECTION AND HORIZONTAL FLUE CONNECTION

sheet is perforated to permit the insertion of the tubes which it supports, and is so designed as to be self-supporting. Owing to its circular form it cannot break inwardly without stressing the inner annular ring of metal which joins the casting, and at the same time is free to move laterally as a whole. The burner or group of burners is supported at the center by radial members, and a flat arch is suspended from the radial members to form the top of the combustion chamber. The base of the chamber is provided with a circular opening of smaller diameter than the chamber connected with the flue, which forms the outlet passage for the products of combustion.

A second tube sheet in the form of an annular ring is provided at the base of the chamber with perforations corresponding to the top tube sheet. This lower tube sheet is supported by the shell on the outside, also by the structure formed by the outlet flue on the inner side. In this particular design, the whole of the weight of the tubes is taken by the top tube sheet, the bottom tube sheet acting merely as a support for the lining of the shell and the refractory material for its own protection. The tubes are thus permitted to expand and contract freely in a vertical direction, and their weight tends to keep them straight, eliminating the tendency to sag.

It will be noted in this arrangement that the position of the tubes is such that they are substantially out of the direct path of the products of combustion which tend to channel through the center, at the same time they are in full view of the flame. The return bends or junction boxes connecting one tube to the other are shielded from the fire by the tube plates, and housed in by side plates and removable covers which provide access for cleaning.

The tubes are usually connected in series to form a single coil to insure positive flow of the oil, but they can be divided to provide a multiple flow, if desired. In the series heater, the oil is admitted to the top of the economizer bank, passes downward progressively toward the hotter zone, and from thence is led to the inlet tube of the vertical section around the circle and out.

It will be seen from this general arrangement that all the tubes in the vertical section receive a large amount of radiant heat, while, due to their symmetrical arrangement with respect to the fire, they all receive an approximately even amount of radiation. Of course, as the oil becomes heated and the temperature of the tubes increases, the hotter tubes will absorb somewhat less heat than the cooler tubes, which is advantageous as it relieves the heat load on the tubes which might be expected to give the most trouble. The absorption of heat in the vertical section is sufficiently great to reduce the temperature of the gases delivered to the economizer section to 1200 to 1400 deg. fahr., a temperature at which they can be readily handled with a conventional setting.

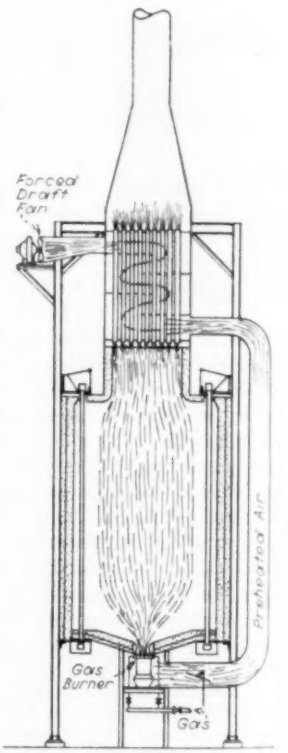


FIG. 2 UPSHOT HEATER WITH ECONOMIZER SECTION REPLACED BY AIR HEATER

With a comparatively large absorbing surface in the vertical section, the products of combustion are cooled very rapidly, making it possible to maintain a high heat gradient between the flame temperature and the average temperature of the gases surrounding the tubes, and further, by proportioning the amount of absorbing surface to the amount of fuel to be burned, it is possible to limit the maximum skin temperatures of the tubes independent of the flame-burst temperatures.

Actually the temperature at a point 10 in. to 12 in. away from the tubes toward the center of the furnace will seldom exceed 1500 deg. fahr., with flame temperatures of 2700 to 3000 deg. fahr., which appears to be a safe limit for tubes carrying oil. In existing installations, the temperatures usually run materially lower than 1500 deg. fahr., and it has been found that the variations in temperature in the length of the chamber are seldom over 100 deg. fahr. Average transfer rates for the whole surface of between 5000 and 8000 B.t.u. per sq. ft. are possible with heaters of this type, depending on the size of the economizer section, which reduces the average.

Another effect resulting from the rapid absorption of heat is evidenced by the reduction of the volume requirements for combustion made possible by the correspondingly rapid contraction of the products of combustion. In the conventional pipe still it is seldom possible to liberate more than 8000 B.t.u. per hr. per cu. ft. of combustion space, and it is probable that 6000 B.t.u. would be a fairer estimate for the average setting. It has

been possible to liberate from 15,000 to 16,000 B.t.u. per hr. per cu. ft. in a tube still of the type described and to continue to do so over long periods of time without endangering the tubes or interfering with combustion. It seems probable that this figure can be materially increased.

Doubtless these figures seem moderate when compared with boiler practice. In oil heating, however, they represent substantial progress.

In oil heating, where tube temperatures are usually higher than in steam practice, and where there is likely to be deterioration of the surface due to oxidation, another advantage results from operation with minimum excess air, apart from the standpoint of economy. The life of the tubes is materially increased by reducing the amount of free oxygen in the gases, and although there always appears to be a slight oxidation at high temperatures, the benefits of running with a minimum of excess air are very marked. In one particular installation, made on a high-temperature cracking plant, tubes have been in service about two years without substantial oxidation when running at oil temperatures of 1050 to 1100 deg. fahr., and where the outside-skin temperature of the tube is materially in excess of this figure.

There has been, to date, very little occasion to effect repairs on brickwork of furnaces built; in fact, the brickwork of the lining, flues, and economizers has shown practically no effects of heating.

In order to provide adequate surface in the combustion chamber, it is usually customary to stagger the tubes to form two rows in such a way that the inner row does not shield the back row from the flame. This usually leads to a staggered arrangement which is very flexible and permits varying the surface with standardized shells and obviates the necessity of making numerous general designs for different requirements in duty. The addition of 1 ft. on the diameter gives roughly 3 ft. on the tube sheet which leads to economy in the large sizes.

Vertical stills are now being built with a range of capacity between 12,000,000 B.t.u. per hr. and 40,000,000 B.t.u. per hr. The larger size stills have a shell diameter of about 20 ft. and a tube length of about 25 ft. in the combustion chamber with the steel work in the economizer in proportion. It is interesting to note that the cost of materials increases at a very moderate rate with capacity owing to the form of the structure, and it is expected that the larger sizes will show a substantial economy in materials per unit of capacity.

Visibility of the tubes, which is important especially in cracking, is provided by peep holes in the side of the shells, and the tubes can be viewed by looking diagonally across the furnace from various points symmetrically disposed in the shell. In the larger furnace top and bottom peep holes are also provided to view the flame.

Access to the tubes for cleaning is provided by removable covers enclosing the top and bottom tube space. The junction boxes connecting the tubes are provided with the usual removable plugs to permit turbining the tubes in place. This operation is carried out from top to bottom, and, due to the disposition of the tubes around the circumference of the combustion chamber, a large number of cleaners can be employed simultaneously, if desired, without interfering with each other. With certain types of junction boxes, dirty tubes can be withdrawn by means of an overhead trolley or crane and replacement effected in a remarkably short time. Rapid cleaning is often important since the furnace is usually connected to a large amount of equipment which must remain idle during cleaning periods and therefore non-productive.

The vertical still controls with great rapidity, and at the same time it appears to be unusually stable. The moderate variations of fuel required to maintain desired outlet temperatures do not seem to affect the percentage of CO_2 to any great extent and the control of the heater can be normally effected without changing the forced-draft pressure. It is interesting to note that if the CO_2 falls off, more load is put on the economizer, and, whereas the outlet temperature of the heater may remain constant, the outlet temperature from the economizer to the vertical section may vary considerably. In this manner changes in combustion conditions may be noticed quickly on the control board of the refinery

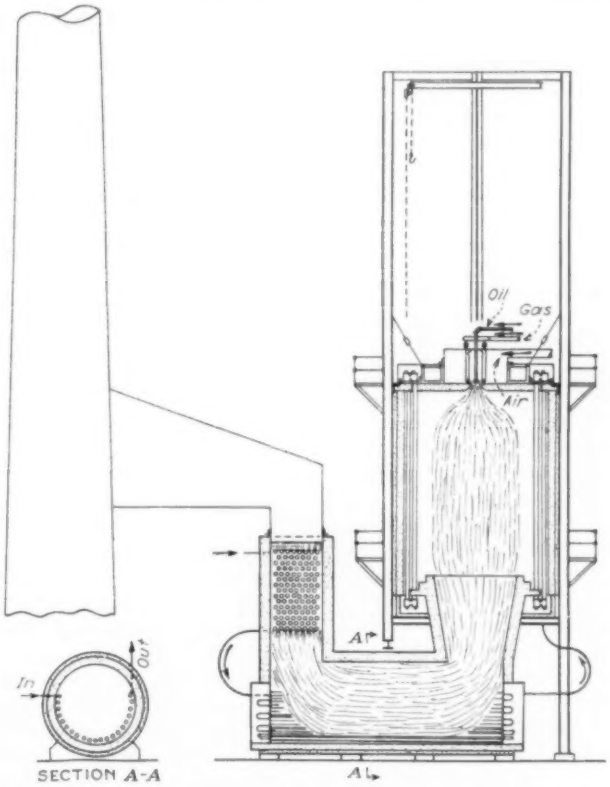


Fig. 3 Recent-Type Heater With Horizontal Flue Forming Secondary Radiant Section

unit and corresponding corrections made. Over two-thirds of the work of the heater is done in the vertical section, the remainder in the economizer. Without the economizer an efficiency of the order of 50 per cent may be expected. Under service conditions an overall efficiency of 70 to 75 per cent can be readily obtained, and it is possible that even better results can be secured when oil-inlet temperatures are not too high.

Fig. 2 illustrates an upshot heater in which the economizer section has been replaced by an air heater, shown more or less diagrammatically above the vertical section. The purpose of the air heater is to reclaim heat from the stack gases and transfer to the air required for combustion. With preheated air it is obvious that higher combustion temperatures will be obtained and therefore more heat will be made available for transfer to the tubes by radiation. This type of heater, many of which are now in operation, is of considerable interest since it carries out more fully the idea of transferring heat from a source at high initial temperatures. It is used in connection with processing where the inlet-oil temperature is of the order of 800 deg. fahr., a temperature which would render a tubular economizer section of little value. In installations of this type average heat-transfer rates for the surface run as high as 10,000 B.t.u. per sq. ft.

The mechanical structure of the heater is very simple, requiring little ground space, and it seems probable that it will lead to

considerable economy in materials. The comparatively short length of travel of the oil should permit high velocities without undue pressure drops, thereby enhancing the transfer rate of heat from metal to oil and reducing the steam required for pumping.

Fig. 3 shows a more recent modification of the type of heater first described. The chief points of interest in this design lie in details of construction, in particular in the use of the horizontal flue to form what may be termed a secondary radiant section.

PME-53-3

Use of Superheated Steam in Oil-Well Drilling

By V. WEAVER SMITH,[1] NEW YORK, N. Y.

Advent of deep drilling for oil has greatly increased the size of the equipment necessary for drilling. The cost of equipment has been mounting very much faster than the return from the wells. Furthermore, the transportation of this heavy equipment is becoming a serious problem. These factors have turned the oil industry to the engineer for assistance in solving the problem of reducing the cost of drilling wells.

Conditions have become so serious that in spite of the fact that the driller pays nothing for his fuel, he is now able to buy power in some cases from public utilities, who pay for this same fuel, more economically than he can produce it. This condition has been brought about because little attention has been paid to decreasing the initial cost and increasing the life of the equipment. Utilities have overcome the fuel cost by more efficient design and operation of the power-plant equipment. The producing industry is in an excellent position to reduce the cost of drilling wells by taking advantage of the improvements that utilities and other industries have made in design of power equipment. By so doing they will recover the advantage of low or no fuel cost, and thereby reduce the cost of drilling.



UP UNTIL a short time ago, as equipment failed or did not seem powerful enough, the tendency was to build it heavier or build it larger. Very little attempt was made to build the equipment more efficient and make it capable of doing more work without increasing its size and weight.

As an example of this the single engine was replaced by a twin engine, and, more recently, the size of the cylinder was increased from 10 in. by 10 in. to 11 in. by 11 in. and finally to 12 in. by 12 in. The size of fluid pumps increased from 10 in. by 4 in. by 12 in. to 15 in. by 7¾ in. by 20 in. In very few instances has any attempt been made to reduce the power required or the weight per unit of work produced. Hence the boiler plant has increased to meet the larger steam demand from two 73-hp., 175-lb. boilers to four 125-hp., 350-lb. boilers necessary to operate a rig.

Steam is the ideal power medium for drilling for many reasons. It gives more flexibility to the ever-changing demands for power over a day's operation. Fuel is invariably available at the location, and water is required for drilling the well. Hence, with all the raw materials necessary for supplying power on the location, the cost of transportation of any other power medium should be overcome very readily by the efficient use of the power medium. Electric power is not available at many locations, particularly in new pools where drilling is often competitive. By standardizing on equipment of one power medium that is usable under all conditions, and developing its efficiency to the highest degree, the initial investment in equipment is reduced, and the problem of replacements is greatly simplified.

The principal reasons for the slow introduction of efficiency into steam power-plant equipment for drilling have been twofold: (1) Most refinements in power-plant equipment have been looked upon as fuel savers and the driller has no fuel cost, (2) until recently the size of the power plant was too small for most refinements to justify their cost. With the heavy power demands of modern drilling and the rapid strides made in power-plant engineering, this condition is now entirely different.

Furthermore, in analyzing the advantages and disadvantages of any piece of equipment it is often necessary to go into detail. For instance, while it is commonly supposed that the reason for using superheaters on locomotives is because of the large saving in fuel and water, it will probably be found on careful analysis that the increase in the capacity of the locomotive and other factors would alone more than justify the investment. And so in considering the use of superheaters for oil-well drilling, a careful analysis of the whole problem must be made. Before considering the advantages it may be well to consider what superheated steam is and how steam is superheated.

Superheated Steam

Steam is superheated when it has been heated above the temperature corresponding to the saturation temperature. The temperature difference between superheated steam and saturated steam at the same pressure is known as the degree of superheat.

Steam that is superheated contains no moisture, and cannot condense until its temperature has been lowered to that of saturated steam at the same pressure. Superheated steam is a more efficient medium than saturated steam for transforming heat energy into mechanical energy and, therefore, fuel into power.

How Steam Is Superheated

Steam is superheated by adding heat to it, after the steam is removed from contact with water. Usually the steam is passed through a vessel located in the path of the furnace gases. Sometimes separate sources of heat are employed. Steam may be superheated or reheated in reheaters or steam resuperheaters while passing from the high- to the low-pressure stages of large turbines, where the heating agent is live steam. Superheaters with separate furnaces, known as separately fired or portable superheaters, sometimes are employed.

Steam can be superheated by throttling. This method, however, is a special instance and is not ordinarily employed, since

---

[1] Engineer, The Superheater Company. Mem. A.S.M.E. The author received his degree of B.S. in Chemical Engineering from Pennsylvania State College in 1914 and the M.S. degree in 1915. After graduation and until 1921 he served in various capacities with the British Chemical Company, Renfrew, Ont., British American Chemical Company, Ridgefield Park, N. J., and Harbor Sales & Manufacturing Company, New York. He then joined the Atlantic Tar & Chemical Works, Ltd., Bayway, N. J., in the capacity of works manager, having charge of rebuilding the plant, until he joined The Superheater Company in 1926.

Presented at the National Petroleum Meeting, Tulsa, Okla., Oct. 6 to 8, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

any degree of obtained superheat is at the expense of reduced pressure.

A Steam Superheater

A steam superheater is an apparatus to increase the temperature of steam without increasing its pressure. It is used generally for raising the temperature of saturated steam direct from a steam boiler. It consists of a system of tubular units or elements with the steam flowing inside and hot gases outside. One end of each tube or unit is connected to a distributor or saturated-steam header, the other end to a collector or superheated-steam header.

A superheater as shown in Fig. 1 for oil-country boilers consists primarily of:

1 A special cast-steel header comprising two compartments, one acting as the distributor for the saturated steam coming from the boiler, and the other for the collection of the steam after it has been superheated, and from which the steam is carried to the main steam line.

2 Detachable units of heavy-gage, seamless, drawn, steel tubing, which are formed by bonding two or more lengths of tubing by forged return bends. These units are connected to the header by means of clamped metal-to-metal ball joints.

3 Connecting pipe between the boiler nozzle and the saturated-steam inlet, safety valve, and bleeder valve.

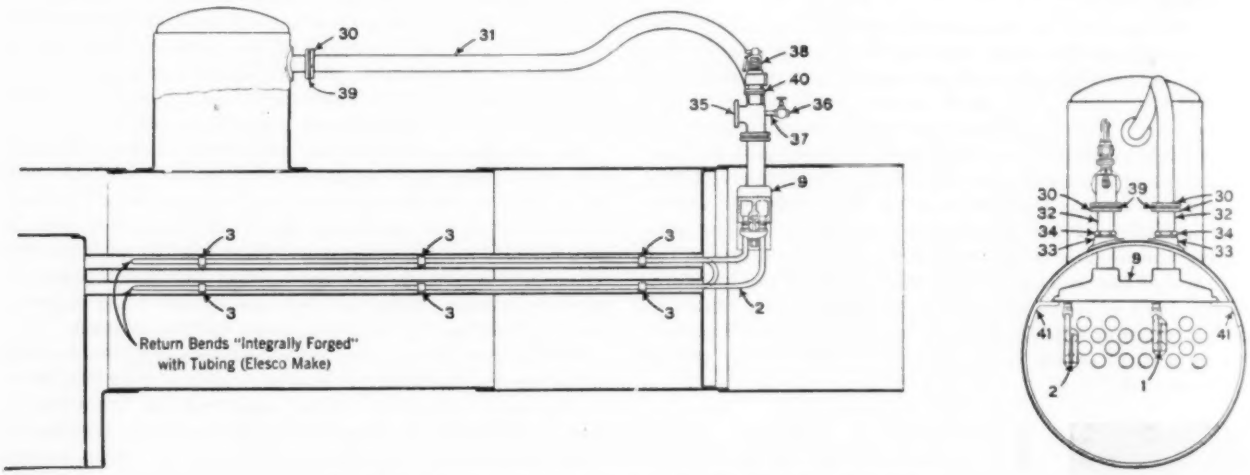


Fig. 1 Superheater Application to Oil-Country Boilers

(1, Unit shape No. 1; 2, Unit shape No. 2; 3, Unit supports; 9, Header; 30, Connecting flanges; 31, Bent connecting pipe; 32, Superheater inlet and outlet pipe nipples; 33, Offset boiler flange; 34, Bushing for list No. 33; 35, Superheater outlet tee; 36, Bleeder valve; 37, Bleeder-valve nipple; 38, Superheater safety valve; 39, Gaskets for 4-in. connections; 40, Gasket for 1½-in. connection; 41, Superheater header brackets.)

Effect of Superheated Steam in Oil-Well Drilling

A careful study of drilling operations was made to determine if superheated steam offered any advantages over saturated steam. This investigation indicated that the use of superheated steam would offer the following advantages:

1 By reducing the steam consumption of the drilling engine and the fluid pump
2 By reducing radiation losses in pipe lines from boilers to the drilling rig
3 By reducing the steam pressure of the boilers for required steam pressure at the engine and pump
4 By increasing the pressure of the fluid pumped to the well
5 By increasing the capacity, speed, and flexibility of the drilling rig
6 By increasing the life of the pumps, engines, and boilers, due to better operating conditions
7 By decreasing weight of boilers required to operate the rig
8 By decreasing the amount of water required for boilers with corresponding reduction in scale formation
9 By reducing the fuel consumption
10 By increasing the boiler efficiency.

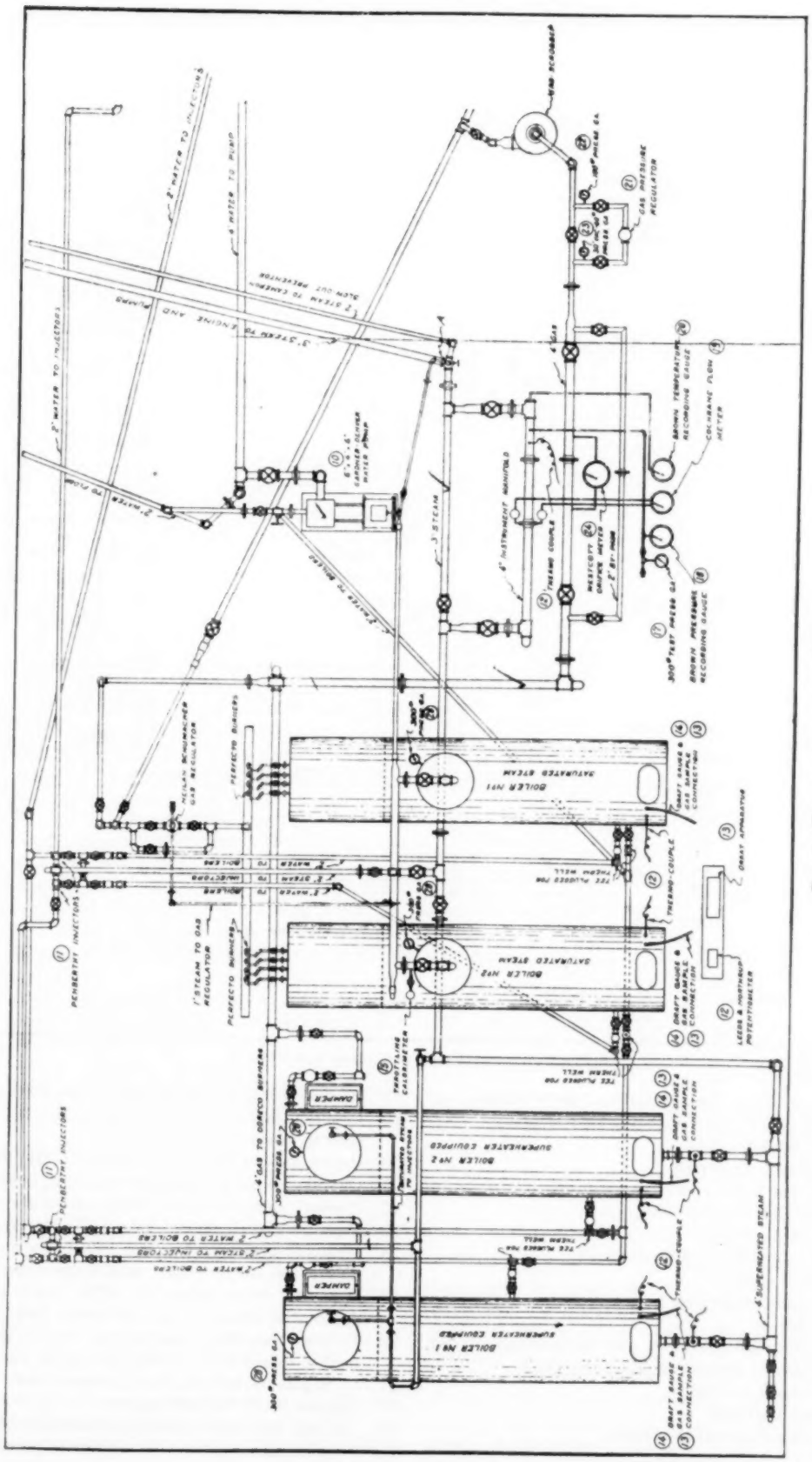
Most of these advantages can be determined theoretically or from experience in other industries. But in order to test in a practical way the use of superheated steam under actual drilling conditions, two boilers were built and equipped with superheaters. The set-up was made as later described in detail so as to approach normal operations as far as possible.

The boilers were set up according to the common practice used in the Gulf Coast fields. The arrangement is shown in Fig. 2. A 3-in. line connected the steam header of the saturated-steam boilers to the rig. The boilers equipped with superheaters were set up in the same manner with a connection from their header, which is in front of the boiler, to the 3-in. line running to the rig. Valves were provided so that either set of boilers could be cut in or out at will without interfering with the operation of the rig. A manifold was provided between the boilers and the rig in which were installed connections for recording pressure, temperature, and the flow of steam from either set of boilers.

A second manifold, as shown in Fig. 3, was provided on the branch line going to the pump in which connections were made for measuring the steam consumption of the pump. Provisions were made for measuring the moisture and temperature of the steam at the boilers, engine, and pumps. Steam gages were provided to register the steam pressure at those points.

Water was ordinarily supplied to the boilers by means of injectors. The water was taken from a common source of supply, namely, two 250-barrel tanks. A 6 × 4 × 6 duplex pump was provided for supplying feedwater in emergencies. No feedwater heaters were used.

Casinghead gas was used for fuel of approximately 1400 B.t.u. quality. This gas was passed through a centrifugal scrubber to remove any oil or water carried over from the separator. The gas was measured by means of an orifice meter. The supply of gas to the boilers was controlled by means of the Doreco boiler governor on the superheater-equipped boilers, and by means of the Neilan-Schumacher spring-type governor on the saturated-steam boilers. Provision was made at the smoke-box of the boilers for measuring the flue-gas temperatures and making flue-gas analyses. The arrangement of the equipment and the location of the instruments are shown on Figs. 2 and 3.


BOILER N°1
SATURATED STEAM
BOILER N°2
SATURATED STEAM
BOILER N°2
SUPERHEATER EQUIPED
BOILER N°1
SUPERHEATER EQUIPED
DAMPER
DAMPER
PENBERTHY INJECTORS
PENBERTHY INJECTORS
PERFECTO BURNERS
PERFECTO BURNERS
THERMO-COUPLE
THERMO-COUPLE
THERMO-COUPLE
DRAFT GAUGE & GAS SAMPLE CONNECTION
DRAFT GAUGE & GAS SAMPLE CONNECTION
DRAFT GAUGE & GAS SAMPLE CONNECTION
DRAFT APPARATUS
LEEDS & NORTHRUP POTENTIOMETER
THROTTLING CALORIMETER
SUPERHEATED STEAM
BROWN PRESSURE RECORDING GAUGE
BROWN TEMPERATURE RECORDING GAUGE
COLORADO FLOW METER
GAS REGULATOR
STEAM TO GAS REGULATOR
GAS TO DORCO BURNERS
2" WATER TO INJECTORS
2" WATER TO PUMP
2" STEAM TO ENGINE AND PUMPS
2" STEAM TO CAMERON SLOW OUT PULSOMETER
2" WATER TO BOILERS
2" STEAM TO INJECTORS
2" WATER TO BOILERS
2" WATER TO PUMP
3" STEAM
4" INSTRUMENT MANIFOLD
THERMO COUPLE
WESTCOTT ORIFICE METER
GARDNER DENVER WATER PUMP
300# TEST PRESS GA
300# PRESS GA
300# PRESS GA
4" GAS
4" GAS
4" GAS

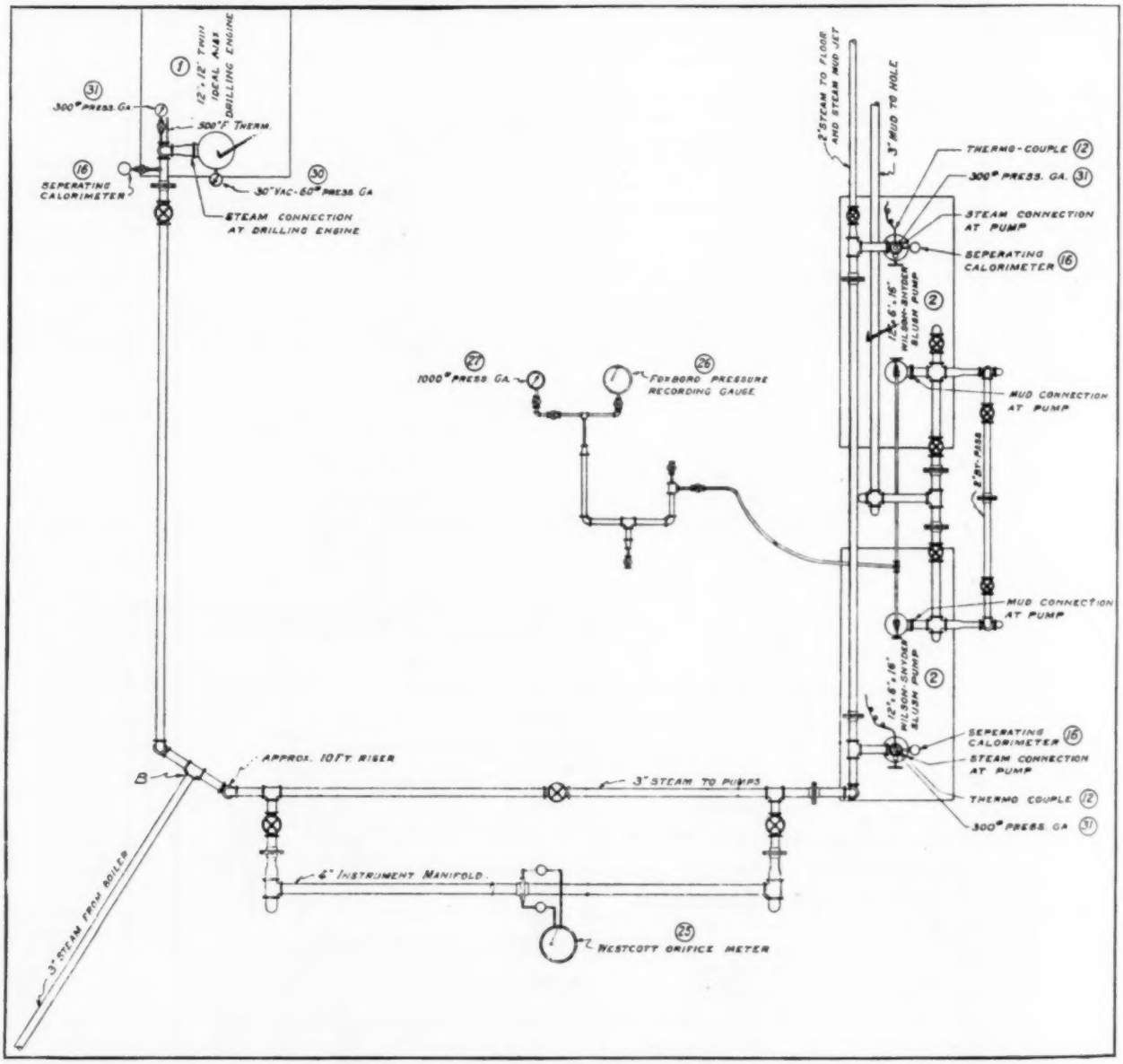


FIG. 3 CONNECTIONS AT ENGINE AND PUMPS

1 Engine, 12 in. by 12 in. Twin Ideal Ajax roller bearing
2 Pumps, two 12 × 6 × 16 Wilson-Snyder
3 Rotary, 20-in. Boykin
4 Drawworks, Beaumont, 4-post
5 Swivel, Ideal, type 6-A
6 Traveling block, 3-sheave Boykin
7 Crown block, Baash-Ross
8 Drill pipe, 6 in. to 4000 ft., 4 in. on down
9 Hook, 8 in. Ideal spring
10 Water pump, Gardner-Denver 6 × 4 × 6
11 Injectors, Penberthy
12 Leeds & Northrup potentiometer and thermocouples 0 deg. to 2000 deg. fahr. for measuring all temperatures
13 Hays Orsat flue-gas analyzer
14 Ellison inclined draft gage
15 Ellison throttling (U-path) calorimeter
16 Schaeffer & Budenberg separating calorimeter No. 1601
17 Ashcroft test pressure gage, 0–300 lb.
18 Brown recording pressure gage, 0–300 lb. No. 85549
19 Cochrane flow meter, orifice 2885 in. dia. 0–12,000 lb. per hr.; used to measure total steam flow
20 Brown gas-actuated thermometer, 0–600 lb. No. 85736
21 Chaplin-Fulton gas-pressure regulator, 0–600 lb.
22 Pressure gage, 0–100 lb. High side of gas regulator
23 Pressure gage, 0–60 lb. Low side of gas regulator
24 Westcott orifice meter, No. 19912, static pressure 50 lb., differential 100 in. water, used as gas meter, $1^1/_2$-in. orifice
25 Westcott orifice meter No. 29731, static pressure 500 lb., differential 100 in., to measure steam flow, $2^3/_4$-in. orifice
26 Foxboro recording pressure gage, No. A-75807, 0–1000 lb.; used to record fluid pressure put up by pump
27 U. S. gage, 0–1000 lb., fluid-pressure check
28 Regular 0–300 lb. boiler gages
29 120 deg. fahr. thermometer for feedwater temperature
30 0–60 lb. gage used for engine cylinder pressure
31 Four 0–300 lb. gages

It is to be noted that all of the equipment used for drilling the well were identical throughout the test with the exception of the boilers and burners for the boilers which were used to provide the superheated and saturated steam respectively. The specifications of the boilers are given in Table 1.

TABLE 1 COMPARATIVE DATA ON SUPERHEATER-EQUIPPED AND SATURATED-STEAM BOILERS USED IN TEST

	Superheater equipped	Saturated steam
Type of boiler	Locomotive, portable	Locomotive, portable
Code	A.S.M.E.	A.S.M.E.
Rating, horsepower	65	90
Working pressure, lb. per sq. in.	180	200
Test pressure, lb. per sq. in.	270	300
Heating surface, evaporative, sq. ft.	649	919
Furnace volume, cu. ft.	74	86
Furnace volume per sq. ft. heating surface, cu. ft.	0.114	0.094
No. tubes, 3 in. o.d.	46	80
No. tubes, 5 in. o.d.	16	None
Length tubes, ft.	12	14
Steam outlet diameter, in.	3	4
Stack diameter, in.	25	25
Stack height, ft.	32	28
Weight of complete boiler, lb. each	18,885	21,460
Burners used on boilers, low-pressure gas	Doreco	Perfecto

Method of Conducting the Test

Since the purpose of the test was to obtain a comparison between superheated steam and saturated steam for the drilling operation, each set of boilers (two each) was operated for alternate periods of 24 hr. throughout the duration of the test. The test was started when the well was at a depth of approximately 1800 ft. and discontinued at a depth of approximately 4000 ft. The test was carried on for 18 days to be sure of obtaining average drilling conditions at the well while both kinds of steam were supplied. Readings of steam pressure, temperature, speed, fluid pressure, moisture in steam, steam and fuel consumption, etc., were taken every half hour throughout the test.

In order to eliminate the advantage of steam pressure of one kind of steam over the other, the automatic fuel regulators were set to obtain the same pressure at the point of measuring the flow of steam on both sets of boilers.

The entire test was conducted under normal drilling conditions. Throughout the test period no interruption to the drilling operation was caused by running the test. The operation of the drilling rig and the firing of both sets of boilers were done by the regular drilling crew.

Data on the boilers, engines and pumps, and at the instrument manifold were collected at half-hour intervals throughout the run. The data collected are shown in Tables 2, 3, 4, and 5.

TABLE 2 DATA TAKEN AT BOILERS[a]

	Saturated steam	Superheated steam
Boiler steam		
Pressure, lb. per sq. in., gage	181	179
Temperature, deg. fahr.	380	569
Superheat, deg. fahr.	....	190
Moisture in steam, per cent	0.5	....
Fuel		
Calorific value, B.t.u. per cu. ft.	1400	1400
Fuel consumed, average cu. ft. per hr.	14,017	12,011
CO_2 in flue gas, per cent	10.0	8.0
Temperature flue gas, deg. fahr.[b]	697	700
Draft at base of stack, inches of water	0.19	0.18
Feedwater temperature, deg. fahr.	81	84

[a] Averages for 18 days.
[b] Saturated-steam boiler operated at average rating of 167 per cent and superheater-equipped boiler at average rating of 228 per cent.

The data thus obtained permit a comparison of the effects of the saturated and superheated steam upon the operation of the pumps and engines, the pressure drop and heat losses in the lines, and thermal efficiencies of the boilers, and other points of interest.

TABLE 3 DATA TAKEN AT INSTRUMENT MANIFOLD COMMON TO BOTH SETS OF BOILERS[a]

	Saturated steam	Superheated steam	Increase per cent	Decrease per cent
Steam flow, average lb. per hr.	9053	8010	..	11.5
Steam pressure, lb. per sq. in., gage	168	168	..	..
Steam temperature, deg. fahr.	375	497	..	..
Superheat, deg. fahr.	....	122	..	..

[a] Averages for 18 days.

TABLE 4 DATA TAKEN AT PUMP AND ENGINE[a]

	Saturated steam	Superheated steam	Increase per cent	Decrease per cent
Steam pressure at pump, lb. per sq. in.	142	151	..	..
Loss in steam pressure between manifold and pump, lb. per sq. in.	26	17	..	34.6
Steam temperature, total deg. fahr.	....	424	..	..
Superheat, deg. fahr.	....	58	..	..
Moisture in steam, per cent	17	....	..	..
Radiation losses in pipe lines from boiler to pump, hp	39.3	18.2	..	53.7
Steam used by pump, lb. per hr.	6180	5320	..	13.9
Per cent of total steam required used by the pump	68.3	66.4	..	..
Fluid pressure developed by pump, average	347	387	11.5	..
Pump speed, strokes per minute	57.4	60.4	..	..
Hydraulic output of pump, average hp. developed	91.4	106.3	16.3	..
Water rate of pump, lb. per hydraulic hp-hr.	67.6	50	..	26.1
Speed of engine, r.p.m.	160	164	..	
Back pressure in exhaust chest of pump, lb. per sq. in.	12	9	..	25.0

[a] Averages for 18 days.

TABLE 5 GENERAL NOTES ON TEST

	Saturated steam	Superheated steam	Increase per cent	Decrease per cent
Horsepower of boilers, rated, total	184	130	..	..
Horsepower of boilers, developed, total	308	297	..	..
Per cent of rating	167	228	..	..
Pounds of water evaporated per hr. per boiler	4526.5	4005	..	11.5
Pounds of water evaporated per hr. per sq. ft. heating surface	4.94	6.17	24.9	..
Cu. ft. gas burned per hr. per boiler	7008.5	6005.5	..	14.3
Furnace volume of boilers, cu. ft.	86	74.4	..	..
Cu. ft. gas burned per hr. per cu. ft. furnace volume	81.5	80.7	..	..
Furnace volume per sq. ft. heating surface, cu. ft.	0.094	0.114	..	..
Furnace volume per horsepower developed, cu. ft.	0.279	0.25	..	..
Calorific value of fuel per cu. ft., B.t.u.	1400	1400	..	..
Heat release per cu. ft. furnace volume per hr., B.t.u.	114,000	112,980	..	..
Temperature flue gas, deg. fahr.[a]	697	700	..	..
Temperature of steam at boiler pressure, deg. fahr.	380	379	..	..
Temperature of steam leaving superheater, deg. fahr.	.....	569	..	..
Difference between steam and gas temperature, deg. fahr.	317	131	..	..
Gas burned per hp-hr. developed, cu. ft.	45.5	40.44	..	11.0
B.t.u. per boiler hp-hr. developed, theoretical (1)	33,479	33,479	..	..
B.t.u. gas consumed per boiler hp-hr., cu. ft. × heat value (2)	63,700	56,616	..	11.0
Boiler efficiency = (1) ÷ (2)	52.8	59.4	12.5	..

[a] Saturated boilers operated at average rating of 167 per cent and superheater-equipped boilers at average rating of 228 per cent.

Results on Pumps and Engines

It can be seen from the foregoing data that the hydraulic horsepower developed, which can be used as a measure of the work done by the steam, was increased with a corresponding reduction in the steam consumption of the rig. Hence it is necessary to calculate the savings in fuel, water, and the increased capacity in terms of the work done to appraise the full benefit derived from the use of superheated steam.

From the foregoing table it is seen that the hydraulic horsepower of the pump increased 16.3 per cent when superheated steam was used; hence:

9053 × 1.163 = 10,528.64 lb. of saturated steam to develop 106.3 hp.
8,010.00 lb. of superheated steam to develop 106.3 hp.

2,518.64 lb. saving in steam, or a decrease of 23.9 per cent in the quantity of steam required for the same work done.

Fig. 4 is a water-rate curve, plotted from the data taken on the pump for the different horsepower developed under changeable conditions. This curve indicates that as the fluid horsepower is decreased, the savings due to superheat increase rapidly. At a load of 60 hydraulic horsepower the water rate is reduced from 112 lb. of saturated steam per hour to 64 lb., or a saving of 43 per cent.

While the water rates may not be true figures, they are comparative since they were determined the same way for both kinds of steam. The hydraulic horsepower developed was figured from the speed, displacement, and pressure developed by the pump. Since there is bound to be some slippage of the pump, both steam rates per hydraulic horsepower should be increased 25 per cent or more to represent actual steam rates.

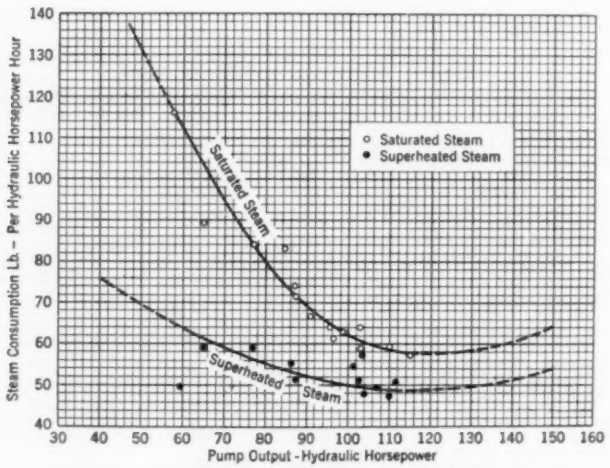


FIG. 4 STEAM CONSUMPTION OF THE MUD PUMPS
(When operated with steam furnished by the saturated-steam boilers and the superheater-equipped boilers.)

The question has often been raised as to how it is possible for there to be such a large saving in steam in a cylinder with superheated steam when no expansion takes place. It is because in uninsulated cylinders the heat radiation is so great. This can be seen from the curve. The slower the pump is running, the greater is the time per stroke or the time the steam is exposed to radiation or cooling.

Likewise, the starting of the piston to move in the opposite direction is a relatively large part of the total time of the stroke. Hence, a pump with a short stroke with saturated or superheated steam is less efficient than a long-stroke pump, all other things being equal.

Since there were no easy means of measuring the work done by the engine without interfering with the operations, no measurements were taken. Since the engine is a reciprocating type, taking steam for the full stroke, superheated steam would be expected to show an equal decrease in the quantity of steam required.

The speed of operation and the flexibility were improved due to the elimination of moisture from the engine. The saturated steam contained 17 per cent of moisture, while the superheated steam still contained some 50 deg. of superheat at the engine. It was observed that it was necessary to pick up several more "thribbles" in low speed with saturated steam than with superheated steam. This reflects the improved operation of superheated steam.

Losses in Steam Lines

The steam line from the saturated boilers to the pumps consisted of 44 ft. of 4-in. standard line pipe and 219 ft. of 3-in. standard line pipe, together with twenty 90-deg. angle fittings. The steam line to the engine consisted of 19 ft. of 4-in. pipe and 210 ft. of 3-in. pipe, together with thirteen 90-deg. angle fittings.

The steam line from the superheater-equipped boilers to the pumps consisted of 50 ft. of 4-in. standard line pipe and 244 ft. of 3-in. standard line pipe, together with twenty-one 90-deg. angle fittings. The steam line to the engine consisted of 25 ft. of 4-in. pipe and 235 ft. of 3-in. pipe, together with fourteen 90-deg. angle fittings.

The lines would have been considerably shorter under normal operating conditions, and would have contained less 90-deg. angle fittings. The longer lines and increased number of fittings were necessary to install the manifolds for test purposes.

The radiation losses in the case of superheated steam amounted to 610,000 B.t.u. per hr. and for saturated steam 1,350,000 B.t.u. per hr. These losses are equivalent to 18.2 boiler horsepower for the superheated steam and 39.3 boiler horsepower for the saturated steam.

The average pressure drop between the instruments on the manifolds at the boiler and the pump was 17 lb. for superheated steam and 26 lb. for saturated steam. This line consisted of 200 ft. of 3-in. standard line pipe with eleven 90-deg. angle fittings.

For safety reasons the use of superheated steam, therefore, permits the boilers to be placed at a greater distance from the rig for the same amount of heat and pressure losses. However, for maximum efficiency, the steam lines should be as short and as free of turns as possible.

Results Obtained on Boilers

No attempt was made to conduct an accurate thermal-efficiency test on the boilers, but readings of the fuel consumption, steam output, flue gas, and feedwater temperatures were taken periodically. The saturated-steam boilers operated at an average output of 9053 lb. of steam per hr. or 308 hp., which is 167 per cent of their rated (92 hp.) capacity. The superheater-equipped boilers were operated at an average output of 8010 lb. per hr. or 297 hp., which is 228 per cent of their rated (65 hp.) capacity. At these average operating conditions, the flue-gas temperatures for the saturated-steam and superheater-equipped boilers were 697 deg. and 700 deg. fahr., respectively.

From the data it can be seen that the fuel required to operate the rig with saturated steam was 14,017 cu. ft. per hr., while with superheated steam only 12,011 cu. ft. were required to operate the rig at 16.3 per cent increased capacity, hence:

14,017 × 1.163 = 16,301 cu. ft. required to do same work with saturated steam
12,011 cu. ft. required to do same work with superheated steam

4,290 cu. ft. fuel saving per hr., equivalent to 26.3 per cent in fuel for same work done.

From the data it can be seen that the average steam required to operate the rig with saturated steam was 9053 lb. per hr. With superheated steam it was 8010 lb. per hr. Since the steam used is equivalent to the water evaporated, the saving in water consumption can be calculated as follows:

9053 × 1.163 = 10,528 lb. water evaporated to do same work with saturated steam

8,010 lb. water (when superheated steam was used)

2,518 lb. water saving, or a decrease of 23.9 per cent in the quantity of water required.

Since scale formation is directly in proportion to the water evaporated, it can be said that there is a reduction of 23.9 per cent in the scale deposit.

The overall thermal efficiencies of the saturated-steam and superheater-equipped boilers were 52.8 per cent and 59.4 per cent, respectively. The saturated-steam boilers had previously been used to drill 9100 ft. of hole and the superheater-equipped boilers had been used to drill 4000 ft. of hole. On a preliminary test these two superheater-equipped boilers developed 393 hp. under maximum firing conditions and without consideration of efficiency. This represents 325 per cent of the rating.

With the reduction in water evaporated to operate the rig, it is obvious that the scale deposit in the boiler is reduced a corresponding amount. This results in less foaming of the boiler, and decreases the amount of blowdown on the boiler.

Since less steam is required for drilling, and more pressure is developed by the mud pump by the same steam pressure when using superheated steam, it is apparent that smaller boilers can be used to perform the work required. This will reduce the difficulty and cost of transporting the boiler in the field, and also reduce the working pressure of the boiler. The superheater-equipped boilers were unloaded at three different locations in running the preliminary and final tests. In transporting them, due to bad roads, they were thrown off the trucks on one trip. During transportation, the superheaters were not removed from the boilers. No leaks or other mechanical troubles developed in the superheaters although the boilers did leak somewhat. This leakage in the boiler was probably due to excess operating pressures when attempting to carry the load at the preliminary test location. This indicates that no trouble in transportation is to be expected with superheater-equipped boilers.

The elimination of excessive moisture by the superheated steam from engine and pump and the better lubrication indicated that the life of this equipment would be longer than with saturated steam. The decrease in scaling troubles of the boilers will increase the life of the tubes and the entire boiler. Lower operating pressures will have the same effect.

It can be seen from the data taken that the capacity of the rig is increased 16.3 per cent by the use of superheated steam. This will result in decreased drilling time in drilling wells. With more holes drilled per rig, the entire investment in drilling equipment can be reduced. When drilling offset wells, the increased capacity due to superheated steam is particularly advantageous.

Operating Conditions When Using Superheated Steam

Neither of the firemen at the well on which the preliminary test was run, or when conducting the regular test, experienced any difficulty in operating superheater boilers. They seemed well satisfied with them. Superheater-equipped boilers are fired in exactly the same manner as saturated-steam boilers except that when firing up, a bleeder at the outlet of the superheater must be left open to permit passage of sufficient steam to protect the superheaters until they are put on the line.

No difficulties of any nature were encountered in operating the rig. No special knowledge is required of the operators in using superheated steam. It is to be noted, however, that a good grade of high-flash lubricating oil should be used for the engine and the pump. With superheater-equipped boilers the steam header is across the front of the boiler. When injectors are used to supply the feedwater, a small saturated-steam header is required from the dome of the boiler as injectors do not operate as readily on superheated steam.

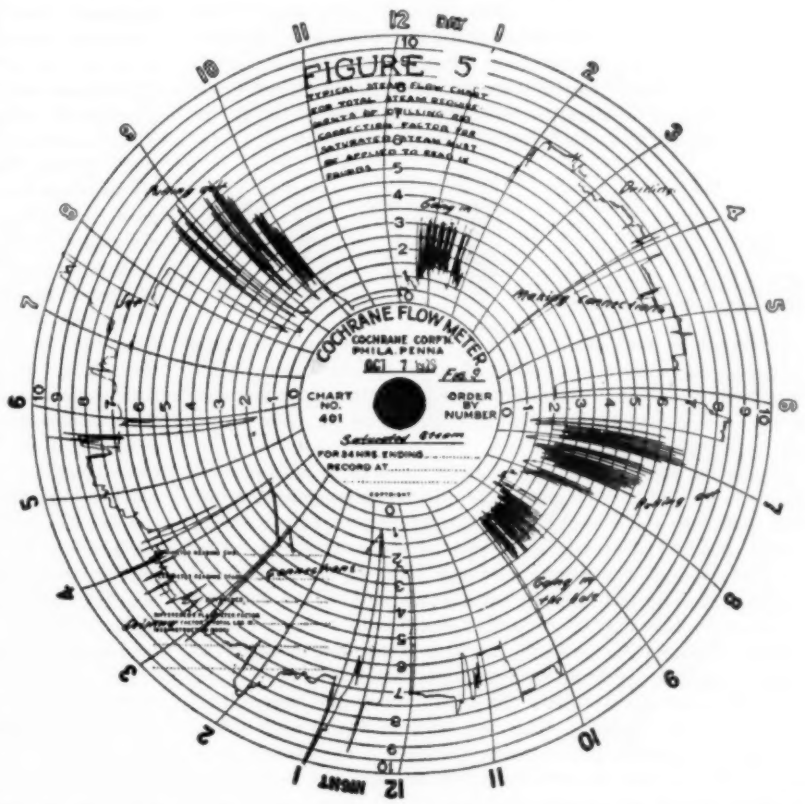


Fig. 5 Typical Steam-Flow Chart for Total Saturated-Steam Requirements of the Rig

(Variations in steam demand for the various operations on the rig are indicated.)

The superheater is of simple construction, consisting of two steam headers and units inserted in large flues in the boiler. These units are clamped to the headers with a ball joint held in place by one bolt. To replace a unit it is necessary only to open one bolt to make the change. Only two types of units are required, so the investment in replacements and number of parts necessary to stock is small.

As pointed out, no mechanical troubles were encountered in the use of the superheaters. The superheat obtained was as high at the end of the run as at the start of the preliminary run. This indicates that the units are remaining clean. In view of this, no trouble is anticipated in the life of the equipment.

It is to be pointed out that the large flues containing the superheater units cannot be swabbed out as readily as the regular tubes. This does not seem to present a serious disadvantage as they can be cleaned readily with a steam lance.

Boilers equipped with superheaters do not contain as much heating surface as saturated-steam boilers of the same size due to the large flues necessary to accommodate the superheater units. However, the increase in boiler efficiency and capacity of the drilling rig, as shown in this report, indicates that this is not a disadvantage.

It can be seen from the results of this test that the use of superheated steam gives a remarkable improvement in the operation of all the major equipment on the drilling rig. It can be seen that in addition to the fuel and water saving that can be made, the increased capacity of all the equipment without increased weight, the increased life due to better operation, increased speed of drilling, and flexibility due to superheating offer many decided advantages.

After this test was made, boilers equipped with superheaters were introduced into Oklahoma, and additional results are now obtainable. It is interesting to compare the operation given in Table 6 on rigs using the same pumps and engines, but using saturated-steam and superheater boilers, all operating at approximately the same depth of hole.

TABLE 6

	Superheater	Superheater	Saturated	Saturated
Rated hp	110	110	125	125
Number, rig	3	3	3	3
Steam pressure	265	230	355	330
Flue-gas temp., deg. fahr.	640	635	910	870
Pump speed, cycles per min.	60	60	60	57

The burners used on these four rigs were the same, so that the flue-gas temperature is a good indication of how much harder it is necessary to fire a saturated boiler to run the rig on an equal basis. These saturated rigs were typical of all the boilers used in the Oklahoma City field.

Operating the rigs with lower steam pressure is an important factor in the life of the boiler. The question is sometimes raised as to why the speed of the pump can be maintained with lower steam pressure when operating with superheated steam. Due to the excessive condensation taking place in an uninsulated cylinder the effective cylinder pressure required is not built up as fast with saturated steam as with superheated steam. Consequently a higher line or boiler pressure is required to effect the required cylinder pressure with saturated steam.

When some operators saw the advantage of the superheated steam at 250 lb. pressure, their first reaction was to increase the pressure further to improve their operation. They found little advantage, for once the effective pressure required was met, any further pressure only increased it slightly due to the higher total steam temperature.

In the Oklahoma City fields it is quite common to see batteries of four 125-hp., 350-lb-pressure, saturated-steam boilers operating a rig. If the results of the test are considered it can be seen that three superheater-equipped boilers would not have to be operated at any higher rating than the four saturated-steam boilers to run the rig.

The comparison of weight of boilers given in Table 7, to be transported from one location to another, shows a decided advantage of the more efficient power plant.

TABLE 7 COMPARISON OF WEIGHT OF BOILERS PER BATTERY TO TRANSPORT

Weight of four 350-lb., 125-hp., saturated-steam boilers, lb.	135,064
Weight of three 250-lb., 125-hp., superheater-equipped boilers, lb.	94,419
Less weight to be transported per well drilled, lb.	40,645
Reduction in per cent in favor of superheaters	30

Boilers are moved to many locations during their life, so that this saving in transportation charges is saved many times over. When one boiler is eliminated from a battery, it is not only the boiler that is saved—all the auxiliary equipment such as burners, blowoff valves, pop valves, check valves, and gages are eliminated from the first cost as well as the cost of replacement, installation, and transportation.

By reducing the quantity of steam required, thereby reducing the water consumption, the equipment serving the boiler plant, such as feedwater pumps, condensers, feedwater heaters, fuel regulators, etc., can be smaller, thereby reducing the investment in this equipment.

The use of condensers to supply pure water to the boilers offers a big improvement by the elimination of scale. Yet condensers are not economical except in special cases because of the large quantities of water required and the high cost of the condensers. Since superheaters reduce the quantity of steam, to condense it will assist in making possible the advantages of condensed water.

In using superheated steam it is highly important that a good grade of lubricating oil be used in the cylinders. Furthermore, greater care must be exercised to see that the lubricators are kept filled at all times. When operating with wet steam, operation without lubrication for short periods may not harm the cylinder walls because the moisture acts as a partial lubricant. This is not the case, however, with dry steam, which has no lubricating properties. However, with the improved lubrication due to dry steam, the life of the cylinders will be greatly increased.

In pumps and engines using piston-type valves no difficulty has been experienced when operating at a total steam temperature of 550 deg. fahr. With slide-valve equipment such as is used in California no troubles will be encountered with a total steam temperature of 500 deg. fahr. With the improvement in metals that are now used on pumps and engines for oil-country practice, these temperatures can even be exceeded without any harm to the equipment. Most steam-pump and engine manufacturers recommend total steam temperatures of 500 deg. fahr. on slide-valve equipment and 650 deg. fahr. on piston-valve equipment.

Since the condensation in uninsulated cylinders is great, it is necessary to have from 100 deg. fahr. to 150 deg. fahr. of superheat at the cylinder to overcome the losses and secure the fullest benefit from the superheater. This, together with the line losses, makes it essential to start with from 200 deg. fahr. to 250 deg. fahr. of superheat at the boiler plant.

In California in one field which is very mountainous, where very long steam lines are used, a separately fired superheater has been tried. The purpose there has been to furnish practically dry steam at the rig, but no superheat in the engine and pumps. With covered lines this does not require a high degree of superheat, and 100 deg. of superheat has proved satisfactory in most cases. While this method reduces the load on the boilers equivalent to the line losses, the use of an additional 100 deg. fahr. of superheat would improve the operation greatly and further reduce the load on the boilers. In this particular field the water is very hard and contains a large amount of solids, so reducing the amount of water necessary to evaporate, offers a particular advantage.

Insulation of steam lines on drilling rigs presents many difficulties. While it is true that savings can be made which will justify its installation, it is always a problem to maintain insulation when frequent set-ups are made in new locations. So in the final analysis the saving is hard to prove and justify. For this reason it would appear that a high degree of superheat becomes a more practical substitute for insulation. The cost of an integral superheater for 200 to 250 deg. is very little more than one for 100 deg. of superheat. Very little additional heat is required for the higher superheat and the fuel necessary to secure it has no value. Line losses with superheated steam are much smaller than with saturated steam so that in a large percentage of the locations all the line losses can be absorbed by

the superheat, and still provide sufficient superheat at the rig to give it the full benefit of superheated steam. This system does not throw any additional load on the boilers for no condensation takes place.

Some thought should be given to the type of superheater best adapted to oil-country practice. Up until the present time the locomotive-firebox type of boiler has proved the most practical for oil-country drilling operations. In this type of boiler it is possible to install economically a superheater for high-degree superheat which will outlast the life of an oil-country boiler. This superheater is automatic in its operation without any attention necessary other than to provide a flow of steam to it during the firing-up period. Being located in the flues of the boiler it is always protected by the water surrounding the flues for all variations in load. Its weight is very little, and it can be moved with the boiler without being disturbed, so that its first cost is practically its last cost.

Separately fired superheaters for high degrees of superheat for the quantity of steam used on drilling operations cannot be built in a portable type unless a large portion of the heat is absorbed by radiation. In Fig. 5 the variation in steam flow over a day's operation is shown. Experience in other industries has demonstrated that it is difficult to protect a superheater of this type with large variations of steam flow. Automatic-firing equipment assists greatly in this protection, but failure for a very short time is usually fatal.

When separately fired superheaters are built in a permanent brick setting it is possible to afford some more protection to the superheating surface for low-degree superheat. The first cost of this type is possibly lower than any other type, and where boilers are used in one location for a long period of time they are probably economical. However, when the additional cost of piping, valves, fittings, automatic control, and the small salvage value of the heating surface and setting are considered, it is probable that when the battery of boilers has to be moved frequently the low first cost is soon dissipated.

It can be seen from the results of the test and from the operation in the Oklahoma City field that the use of superheated steam for oil-country drilling presents an excellent opportunity to reduce the cost of drilling operations.

PME-53-4

Lubrication Research Activities

Fifth Report of A.S.M.E. Special Research Committee on Lubrication[1]

A new general program for experimental work and publications recently formulated by the Committee is outlined in this report, and is followed by a brief statement of results already accomplished. Particular attention has been given to the coordination of data obtained by various investigators on viscosity under high pressure, to the study of pressure distribution in the oil film as applied to both thrust bearings and journal bearings, and to the development of methods for measuring oiliness. From experiments on the flow of liquids through very fine glass and platinum capillaries, it appears that the ordinary laws of viscosity may safely be relied upon down to a film thickness at least as small as one-millionth of an inch. The report concludes with a brief review of recent European investigations.

THIS report summarizes the work of the A.S.M.E. Special Research Committee on Lubrication over a period of three years since the previous report (Ref. 1)[2] was submitted. During the first half of this period, the investigation of the effect of the length-diameter and clearance-diameter ratios on the frictional resistance of full journal bearings was completed at the Bureau of Standards (Ref. 2), a study of oil-ring bearings was conducted by the Westinghouse Electric & Manufacturing Company (Ref. 3), and a series of experiments to test the hydrodynamic theory of lubrication was initiated by the Committee at the Pennsylvania State College. During the latter part of the same period a new general program (Ref. 4) for experimental work and publications was formulated by the Committee, as described hereafter, a substantial beginning was made toward collecting funds to support the experimental work, and accordingly some very interesting results are now being obtained.

Experimental Program

The program outlined is a continuation of the earlier work of the Committee (Ref. 1). The Lubrication Committee is chronologically the oldest one of the many existing Special Research Committees of the Society. It was organized in 1915 to investigate the fundamental problems of lubrication phenomena, to formulate results of investigations previously made, and to keep in touch with contemporary research in this field. It deals with a subject of basic importance in many different branches of engineering, including railway and marine transportation, power generation, and manufacturing industries.

The estimated requirement for effectively conducting the following program over a period of two years is $22,500. This program utilizes personnel and facilities which are already available, otherwise a larger fund would be needed:

1 Continuation of the study of viscosity under high pressure $2500
   (*a*) Preparation of a paper on coordinated results of data obtained in England, Germany, and the United States.
   (*b*) Study of practical applications, including relations to oiliness and seizure phenomena.
   (*c*) Extension to higher pressures and temperatures.
2 Journal bearing investigations $10,000
   (*a*) Study of pressure distribution in the oil film, with variations of clearance and length.
   (*b*) Determination of load-carrying capacity for commercially finished bearings.
   (*c*) Further study of the friction of small-sized bearings (up to 2⅜ in. diam.) in relation to mechanical design and other factors.
   (*d*) Development of apparatus for extension of the friction measurements to larger diameter bearings.
3 Oiliness research $10,000
   (*a*) Further development of four types of oiliness machines (Herschel, Dickinson, McKee, and Kingsbury) and comparison of results obtained from them on the same series of oils and of bearing metals.
   (*b*) Study of fundamental facts regarding effect of speed, load, temperature, and condition of the bearing surfaces upon the coefficient of friction under oiliness conditions.
   (*c*) A beginning on the difficult problem of finding the limiting thickness of film below which the properties of the lubricant are no longer the same as when tested in bulk.
4 Investigations in progress which have already been financed:
   (*a*) An experimental test of the hydrodynamic theory of lubrication at the Pennsylvania State College, using a journal-bearing machine designed by Mr. Kingsbury and loaned to the College for this purpose.
   (*b*) Experiments on pressure distribution in oil films at the Kingsbury Machine Works by means of electrical observations on large-scale models.
   (*c*) Experiments on waste-packed bearings, ring oiling and forced lubrication in the Research Department of the Westinghouse Electric & Manufacturing Company.

Publication Program

Some of the most valuable papers on Lubrication published by the Society are now out of print. The Committee has been asked to compile a representative collection of such papers for reprinting in a single volume, uniform in style with other volumes in the A.S.M.E. Research Publications Series. A number of translations and abstracts of foreign papers may also be included.

Several articles are in preparation for Mechanical Engineering and other periodicals outlining briefly the history and present status of lubrication research, as well as the elements of the mechanics and thermodynamics of lubrication.

The results of research conducted by members of the Committee or in cooperation with the Committee are made available in the form of papers presented for discussion at the regular meetings of the Society.

Status of Experimental Work

The status of work recently begun under the experimental program outlined is briefly as follows:

1 Viscosity Under Pressure. Table 1, prepared in connection with the coordination of numerical data from different sources, indicates the range of variables covered. Table 2 summarizes the data thus far obtained on the apparent solidifying pressures and corresponding temperatures. A number of very striking relationships can be brought out by mathematical and graphical representation. It may be noted, however, that no conclusive evidence has yet been obtained of the solidification of either

---

[1] The personnel of this Committee consists of Alan E. Flowers, *Chairman*, Poughkeepsie, N. Y.; Earle Buckingham, Secretary, Cambridge, Mass.; L. P. Alford, New York, N. Y.; L. J. Bradford, State College, Pa.; H. C. Dickinson, Washington, D. C.; M. D. Hersey, Washington, D. C.; H. A. S. Howarth, Philadelphia, Pa.; G. B. Karelitz, New York, N. Y.; Albert Kingsbury, New York, N. Y., and A. E. Norton, Cambridge, Mass.

Presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] References to literature will be found at the end of this report.

pure castor oil or a pure asphaltic-base petroleum oil. As regards the accuracy of these determinations, it may be stated that excellent agreement has been found between the diversified instruments used for the purpose of viscosity measurement, but that the agreement is not so close under the conditions of plastic flow involved in determining solidifying pressures. This was to be expected and should be further investigated when possible.

The practical applications in view will include a search for correlations with oiliness (i.e., reduced frictional resistance), wear, and seizure of surfaces operating under low speed, heavy load conditions such as roll necks, gear teeth, and cutting tools.

TABLE 1 PRESSURE AND TEMPERATURE LIMITS[3] IN VISCOSITY INVESTIGATIONS

Investigator	Lubricant	Max. press., lb. per sq. in.	Max. temp., deg. cent.
Hersey (Ref. 4)	Lard oil	7,000	20
	Mineral oil 1	3,000	20
Hyde (Ref. 5)	Neatsfoot	18,000	40
	Sperm	16,000	40
	Castor	17,000	40
	Rape seed	16,000	40
	Mineral oil 2	18,000	40
	Mineral oil 3	16,000	40
	Mineral oil 4	17,000	40
	Mineral oil 5	15,000	40
Hersey and Shore (Ref. 6)	Lard oil	50,000	100
	Castor	56,000	140
	Mineral oil 4	34,000	93
	Mineral oil 6	58,000	100
	Mineral oil 7	53,000	100
Bridgman (Ref. 7)	Oleic acid	57,000	75
	Kerosene	171,000	80
Kiesskalt (Ref. 8)	Neatsfoot	9,000	80
	Raw linseed	9,000	80
	Peanut	9,000	80
	Mineral oil 8	14,000	80
	Mineral oil 9	14,000	80
	Mineral oil 4	14,000	80
	Mineral oil 5	14,000	80
	Ref. machine	14,000	80
	Spindle	14,000	80
	St. turbine	14,000	80
	Coal-tar oil	14,000	80
	Voltol	14,000	80
Kleinschmidt (Ref. 9)	Lard oil	102,000	75
	Mineral lard	92,000	75
	Sperm oil	72,000	75
	Castor	135,000	80
	Rape seed	31,000	75
	Mineral oil 10	93,000	75
	Mineral oil 4	50,000	80
	Mineral oil 7	82,000	75

[3] Approximate values only are given in this table. The maximum pressures were not always observed simultaneously with the maximum temperatures. The lower limit of pressure was, in all cases, atmospheric. The lower limit of temperature was not in any case below 20 deg. cent. For description of oils see references cited.

TABLE 2 APPARENT SOLIDIFYING PRESSURES

Lubricant	Temp., deg. cent.	Press., lb. per sq. in.	Ref.
Lard oil	22	24,000	6
	25	38,000	9
	40	60,000	9
	75	102,000	9
Mineral lard	75	92,000	9
Sperm oil (unrefined)	25	17,000	9
	40	30,000	9
	75	72,000	9
Sperm oil (N.P.L.)	25	11,000	9
	40	26,000	9
	75	61,000	9
Castor oil[4]	20	19,000	6
	25	23,000	6
Rape seed	25	31,000	9
Mineral oil 10	25	31,000	9
	75	92,000	9
Oleic acid	30	18,000	7
Mineral oil 7	20	17,000	6
	25	24,000	9
	40	43,000	9

[4] One sample only, hence doubtful.

In preparing for the extension of observations to higher temperatures a study has been made by G. H. S. Snyder of the accuracy of the temperature measurements, especially as affected by time-lag; and the Flowers viscometer (ball-and-slanted tube) has been calibrated with a larger diameter ball to adapt it for lower viscosities. A very simple confirmation of the increase in viscosity of castor oil due to high hydrostatic pressure, and of the corresponding increase in the stiffness of soft cup greases and pressure-gun greases, has been obtained by forcing these materials through a long metal capillary discharging into the free atmosphere. The curve for rate of flow plotted against inlet pressure passes through a maximum, i.e., increasing the inlet pressure beyond a certain point reduces the rate of flow. It seems possible that this method may offer some advantages for observations under extremely high pressures.

2 Journal Bearings. A simple drill-press attachment designed by S. A. McKee of the Bureau of Standards has been used for exploring the pressure distribution in the oil film. The vertical bronze bearing is completely immersed in a bath of the lubricant and loaded by a spring balance in a horizontal position. The bearing contains one small hole suitably connected to a pressure gage. By rotating this bearing the circumferential pressure distribution can be determined at as many points as desired. The working portion of the journal is considerably shorter than the bearing, and is driven from above by a spindle of smaller diameter, so that the longitudinal pressure distribution can readily be determined by elevating or lowering the bearing relative to the journal. The line of application of the load can also be varied at will so as to investigate the effect of eccentric loading. Observations to date have shown a decided departure from the Sommerfeld distribution curves for full journal bearings without side leakage.

It is proposed to investigate the load-carrying capacity of commercially finished journal bearings by determining the location of the minimum points on the diagram in which coefficient of friction is plotted against ZN/P, where Z denotes viscosity in centipoises, N revolutions per minute, and P the bearing pressure, i.e., load per unit of projected area, in pounds per square inch. The minimum point is considered as marking the transition from stable to unstable lubrication. From the results of such tests it should be possible to prepare tables of permissible loads, corresponding to any given speed, lubricant, and temperature, for bearings that are not run in. After running in occurs, the factor of safety is considerably increased. The problem is a statistical one in the sense that a large number of tests must be made to establish reasonable limits within which the minimum point may be expected to lie.

To date the tests have been completed on two sets of white-metal journal bearings of widely differing composition. Although the load capacity as defined above was found to be very nearly the same, it is interesting to note that the bearing with the greater lead content showed a more pronounced running-in effect, i.e., the minimum point could be shifted a given distance toward the origin with a much smaller expenditure of frictional work in foot pounds.

One of the objects of the investigation of the effect of the length-diameter and clearance-diameter ratios recently completed (Ref. 2) was to provide correction factors for reconciling the load-capacity observations that might be made on commercial bearings of unequal lengths and clearances.

3 Oiliness Research. The first machine known to this Committee as being suitable for quantitative measurements of oiliness was designed and constructed by Albert Kingsbury and described, together with numerical data on several different lubricants, in a paper presented at the 1895 annual meeting of the Society (Ref. 10).

This horizontal-axis machine was later installed by Professor Kingsbury at the Worcester Polytechnic Institute, where it is hoped that further tests can be made. The horizontal machine

is particularly adapted for tests on screw threads, thrust bearings, and flat annular surfaces. It should not be confused with the vertical-axis machine developed by Professor Kingsbury at Worcester Polytechnic Institute at a later date (Ref. 11), more commonly known as the Kingsbury oil-testing machine, and in which two separate bearing blocks are pressed against the opposite sides of a revolving journal. Both of these machines were found effective in bringing out characteristic differences between the frictional resistance of different lubricants that cannot be fully accounted for by differences in viscosity. These investigations by Mr. Kingsbury may therefore be considered the first scientific contribution toward the measurement of oiliness.

Further progress, especially in cooperative efforts, required agreement on the definition of oiliness. It was clearly desirable that in comparing any two oils they should possess equal viscosities and be tested under identical conditions; but it was not self-evident whether the viscosities should be equal at atmospheric pressure or under the actual pressure of the test. From an early date W. H. Herschel had advocated the former alternative. Herschel's definition is now generally accepted. Accordingly, oiliness may be defined as the property which causes a difference in friction when two lubricants having the same viscosity at atmospheric pressure, and at the true temperature of the film, are tested under identical external conditions. The lubricant giving the less friction is considered as possessing the greater oiliness.

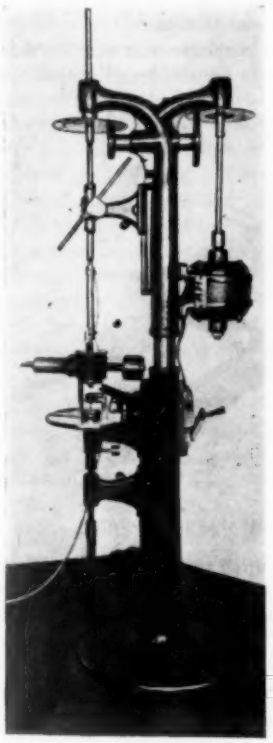


FIG. 1 KINGSBURY OIL-TESTING MACHINE

At least six types of oiliness machines are available for investigation at the Bureau of Standards, including the Herschel oiliness machine (Refs. 12–15, incl.), Dickinson grooved-specimen machine (Refs. 16, 13), Kingsbury oil-testing machine (Refs. 11, 17, 13), McKee journal-bearing machine (Refs. 18, 19, 2), Amsler wear-testing machine (Refs. 20, 21), and a slow-motion inclined plane for static-friction observations.

Among the cardinal requirements of any oiliness measuring machine are: (1) elimination of the influence of viscosity, (2) reproducibility of results, and (3) variability of physical conditions over the full range to be met in service, in so far as it may be found that the property of oiliness depends upon such conditions. It does not, however, appear necessary for the geometrical form of the test surfaces to be the same as those employed in service, any more than it would be necessary for the construction of a viscometer to resemble the construction of an engine for which the oil is being selected. In applying the results of tests on any newly defined property of a material such as oiliness, after the measurements have been completed the task still remains of establishing empirical curves to connect the performance of the machine, taken as ordinate, with the property of the material, taken as abscissa. A report on the logic of oiliness is in preparation in which it is hoped that these questions can be dealt with more fully.

The Herschel oiliness machine appears to be the simplest of those mentioned above as regards construction of the test surfaces. The friction of commercial steel balls is measured against flat annular disks of bearing metal. While the procedure for the use of the Herschel machine has not yet been fully standardized nor its limits of reliability marked out, the following typical results will indicate its applicability to show the effect of changes in chemical composition on the oiliness of a series of lubricants. Six samples of straight mineral oil and six samples of the same oils treated by the addition of a small percentage of oleic acid were obtained from the refiner for oiliness tests. These samples ranged from the lightest to the heaviest grade of automobile oil, and were tested on five types of bearing metal (bronze, brass, babbitt, lead babbitt, and cast iron). Absolute viscosities were then determined at approximately the temperature of the test, as a basis for plotting curves showing the observed coefficients of friction as functions of viscosity. The five pairs of curves thus obtained show a slight systematic decrease in friction with increasing viscosity, and a distinct reduction in friction averaging about 20 per cent, due to addition of the fatty acid.

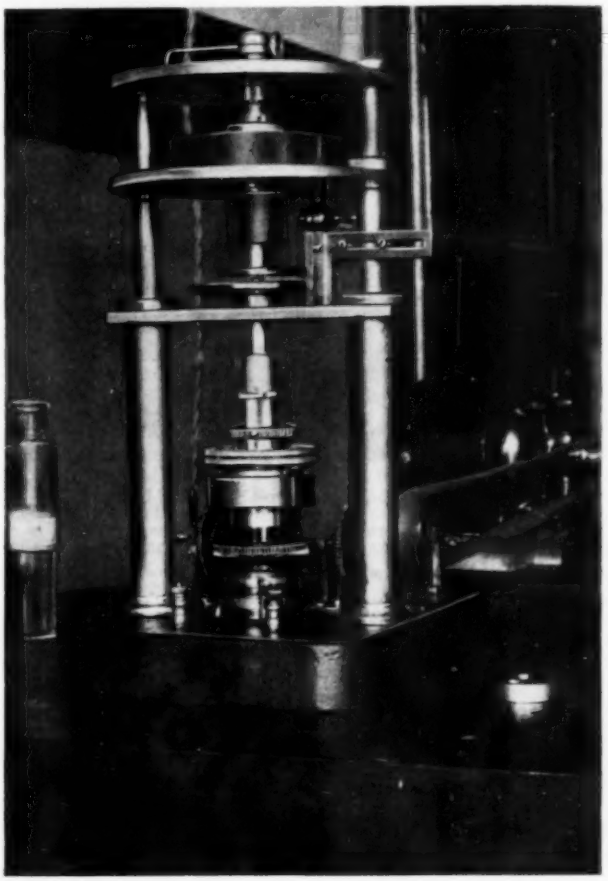


FIG. 2 HERSCHEL OILINESS MACHINE

In the procedure adopted by Mr. Herschel it is intended to eliminate or greatly diminish the influence of viscosity on the performance of the machine by choosing so low a speed for the test that the coefficient of friction, f, will be at or near its maximum value. The existence of such a maximum has been found experimentally, and it lies far below the speed corresponding to the more familiar minimum point. So far as concerns the intrusion of purely hydrodynamic action, it may be inferred that when f passes through a maximum on the speed diagram it must like-

wise do so on the ZN/P diagram, and that therefore $\partial f/\partial Z = 0$ when $\partial f/\partial N = 0$.

A collection of lubricants tested on the Herschel machine are now awaiting their turn on the other machines to ascertain whether they will stand in the same qualitative order of sequence. A good degree of correlation has already been found between the results of static friction observations by W. C. Wilharm of the

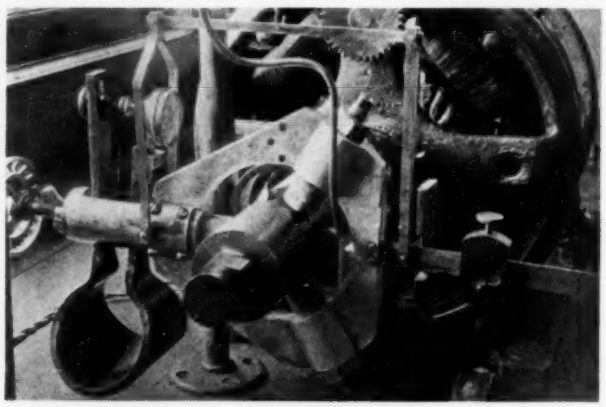


FIG. 3 DICKINSON GROOVED-CYLINDER MACHINE

Westinghouse Electric & Manufacturing Company and the results obtained by Mr. Herschel on the same lubricants, using the same pair of bearing metals (steel and brass).

G. H. S. Snyder has conducted a large number of experiments for the Committee on the Dickinson grooved-specimen machine at the Bureau of Standards, using brass, steel, and white-metal alloys for the bearing metals, in conjunction with lard oil, castor oil, high- and low-viscosity mineral oils, and oleic-acid mixtures. These observations have served to map out roughly the magnitudes of the friction values to be dealt with, and the degree of scattering to be expected upon repeating the tests using successive grooves. This machine was intended for seizure experiments rather than for oiliness measurements in the strict sense, and was designed primarily for testing cutting fluids under conditions of rapid abrasion. Three steel balls, or balls of any desired tool material, are forced against the rotating cylinder of work material in which grooves have already been machined with the same radius as that of the balls, while the load is read on a dial-indicator gage.

A study of conditions at the left of the minimum point (i.e., at low values of ZN/P) was undertaken for the Committee several years ago by C. W. Staples, using the Kingsbury oil-testing machine. This investigation was begun at the Carnegie Institute of Technology through the courtesy of Professor W. Trinks, and afterward repeated and continued at the Bureau of Standards.

In the original Pittsburgh experiments sperm, lard, mineral lard, heavy, medium, and very light mineral oils, and oleic-acid mixtures were tested in conjunction with brass bearing blocks and a steel journal. These bearing blocks were constructed, as in Mr. Kingsbury's tests (Ref. 11), with only a pair of narrow ridges on each block making actual contact on the journal. These ridges were well separated, in order, so far as possible, to prevent the usual tilting action of partial bearings from occurring, in the hope of avoiding the formation of any true fluid film. All this, of course, is for the purpose of eliminating viscosity effects and obtaining boundary lubrication alone, isolated from hydrodynamic action.

Bath lubrication was employed in order to control the temperature of the film and to standardize the amount of lubricant reaching the surfaces. Bearing pressures were brought up to the limit of the capacity of the machine, or about 10,000 lb. per sq. in. as a maximum. This resulted in a noticeable amount of wear, and a moderately wide scattering of observations.

The coefficients of friction, when plotted against ZN/P, showed a very wide divergence between the several oils. Now, if it were not for the existence of oiliness, i.e., some property of the lubricant which can influence friction, other than viscosity, all the curves would be expected to coincide (Ref. 22) provided the rate of wear is not too great. Accordingly, the differences between the ordinates of the curves can be taken as an indication of real differences of oiliness.

The curves for the heavy and medium mineral oils fall high up on the chart, while those for the fatty oils (lard, mineral lard, and sperm) fall considerably lower down, as might have been expected. A surprising fact was, however, that the curve for the very light mineral oil (a paraffin-base spindle oil) intersected the curves for the fatty oils at a sharp angle, so that over a limited range of ZN/P the oiliness of this mineral oil appeared to be superior to that of the fatty oils.

This anomalous result was the occasion for repeating the greater part of the work after a new machine of the same type had been constructed by Mr. Staples in Washington. The absolute values, to which little if any significance can be attached, differed considerably from those obtained in Pittsburgh. The relative positions of the curves for the respective oils were almost exactly repeated, including the anomalous intersection point.

One advantage of the Kingsbury machine over any type in which spherical or cylindrical contacts are employed is that the

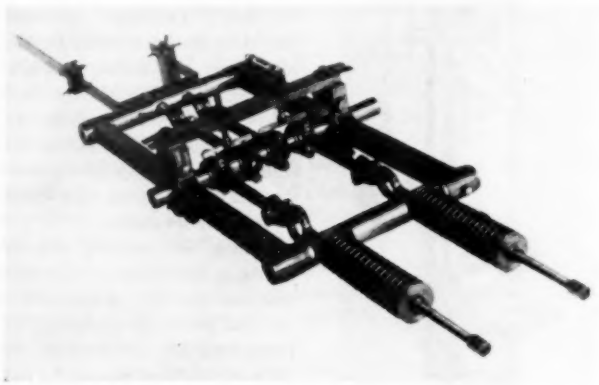


FIG. 4 MCKEE JOURNAL-BEARING MACHINE

total bearing area is a readily measured quantity unaffected by the elastic constants of the metal surfaces. Calculations of the effective film thickness can, therefore, be made, considering the viscosity in the film the same as that of the bath, and working back from the observed friction torque. From such calculations it appears that the film thickness under the heavier loads employed in the foregoing experiments was of the order of only one or at most a small number of molecular diameters.

While the McKee journal-bearing machine was not originally intended for oiliness measurements and requires a long period of careful running in before reproducible results are obtained, it has the advantage that the data will be obtained on actual bearings. The nature of the results that may be expected can be inferred from McKee's investigation of the effect of kerosene upon the oiliness of lubricating oils (Ref. 23) conducted with an earlier form of journal-bearing machine. Observations on the relative oiliness of a series of lubricants using the McKee machine with a set of well run-in white-metal bearings have recently been begun, for the purpose of a comparison with the other oiliness machines.

Another form of journal-bearing test, from which data may be obtained relating to oiliness and seizure, was described by H. C. Mougey in a recent paper (Ref. 57).

The Amsler machine which is in use at the Bureau of Standards for wear tests in connection with metallurgical investigations offers the possibility of oiliness measurements based on the frictional resistance of disks that are in contact circumferentially, and to which any desired differential motion can be given. It has been suggested that film formation might be successfully prevented by rotating the two disks in opposite directions. On account of the friction developed in the measuring mechanism itself, however, this machine is better suited for conditions under which the friction between the specimens is very high, rather than for tests in which the surfaces are to be preserved unchanged.

Another form of rotating-cylinder and stationary-specimen test, from which data may be obtained relating to oiliness and wear or "scuffing," was described by E. Wooler, Timken Roller Bearing Co., before the S.A.E. at Cleveland, November 17, 1930.

The inclined-plane apparatus will be held in reserve until further experience has been reported by the Westinghouse Electric & Manufacturing Company and the Bell Telephone Laboratories, Inc., both of whom are now investigating the problem of static friction measurement. It has been found that to obtain reproducible results requires an extreme degree of protection from vibrations.

With respect to the problem of ascertaining the minimum thickness of film below which the properties of a lubricant are no longer the same as in bulk, it has generally been considered for practical purposes that Albert Kingsbury's experiments with the tapered plug (Ref. 24) established a superior limit of 0.000025 in. The ordinary laws of motion of viscous fluids could therefore be relied upon down to films at least as thin as that amount.

From an investigation conducted by Ronald Bulkley at the Bureau of Standards and at Johns Hopkins University, it would appear that the limit has now been pushed down to, at most, no more than about one millionth of an inch (25$\mu\mu$). Dr. Bulkley's experiments were carried out on extremely fine capillaries of glass and of platinum in conjunction with mineral oil, oleic-acid mixtures, water, and other liquids of widely different chemical composition. The metal capillaries were skilfully drawn down to an internal diameter of about 0.0008 in. (200$\mu\mu$) and the glass capillaries to less than 0.0004 in. (100$\mu\mu$). The details of these experiments have been submitted for publication in the Bureau of Standards Journal of Research. It was concluded that no plastic adsorbed film could have been formed on the walls of the capillaries greater in thickness than the above stated limits without being detected in the course of these experiments as a departure from Poiseuille's law. Poiseuille's original experiments on the viscosity of water included measurements with glass capillaries down to 0.014 mm. (14μ) diameter without finding measurable effects that could be attributed to wall films.

As a preliminary step in the foregoing investigation it was necessary to check up very definitely on the curious phenomenon known as "clogging of capillaries" discovered by Wilson and Barnard in tubes as large as 0.3 mm. (over 1/100 in.), and attributed by them to the gradual building up of plastic adsorbed films of astonishing thickness. (See Ref. 25.) It may be concluded from Dr. Bulkley's experiments that the clogging effects previously mentioned were due solely to foreign particles such as lint and dust which could be eliminated by appropriate methods of filtering, and should not be attributed to the adsorption of any chemical constituent of the liquid.

Further evidence tending to disprove the supposed existence of extremely thick adsorbed films of a plastic consistency (Ref. 26) has recently been found by Ronald Bulkley and F. G. Bitner in connection with surface-tension measurements. (See Ref. 27.) In general, therefore, it appears that there is not so much valid experimental evidence in favor of extremely thick adsorbed films as was at one time assumed, and that the views of Langmuir (Ref. 28) and of Woog (Refs. 29, 30) to the effect that such films are generally monomolecular, or at most only a few molecules thick, are receiving important confirmation.

4 Other Investigations on the Program of the Committee. Professor Bradford at Pennsylvania State College has carried out a number of experiments in which careful measurements have been made of the total pressure on the bearing and the pressure values at different angular positions and has covered a range up to relatively high pressures as compared with standard bearing practice. This work it is expected will continue for at least another year.

A paper is in preparation by Mr. Karelitz entitled "The Oil Flow in a Perfectly Lubricated Bearing," which describes the results of experiments recently completed.

The investigation by Mr. Kingsbury designated as 4(*b*) in the above outline of the Committee's Experimental Program is reported in a paper being presented for discussion at this meeting under the title "On Problems in the Theory of Fluid-Film Lubrication, with an Experimental Method of Solution."

Status of Publications

Fourteen of the earlier publications no longer available have been selected for reprinting as Volume I of Collected Papers on Lubrication Research. The list includes Albert Kingsbury's papers on the friction of screws, on the experiments with an air-lubricated journal, on the new oil-testing machine, on the tests of large shaft bearings, and the first report of the Sub-Committee on Lubrication,[5] of which Mr. Kingsbury served for a long period as Chairman. (Refs. 10, 31, 11, 32, and 24, respectively.) It includes the paper by Professor A. W. Duff[6] on empirical formulas for viscosity as a function of temperature (Ref. 33), and two papers by Dr. A. E. Flowers, one on viscosity measurement and a new viscosimeter, the other on a cylinder friction and lubrication testing apparatus. (Refs. 34 and 35, respectively.) The remaining papers are by M. D. Hersey on the laws of lubrication of horizontal journal bearings (experimental), on the laws of lubrication of journal bearings (mathematical), and on the theory of the torsion and the rolling-ball viscosimeters and their use in measuring the effect of pressure on viscosity (Refs. 36, 22, and 4, respectively); by H. A. S. Howarth on slow-speed and other tests of Kingsbury thrust bearings (Ref. 37); by H. A. S. Howarth and N. Ogden on friction tests of propeller thrust bearings (Ref. 38); and by Commander L. J. Linsley, U. S. N., on the investigation of critical bearing pressures causing rupture in lubricating films (Ref. 39).

While no steps have been taken toward the publication of translated matter, a considerable number of the more recent French and German research papers have been translated and carefully edited, with photostats of illustrations, for the use of members of the Committee and contributors to the Lubrication Research Fund. For services in this connection acknowledgment is made to Winslow H. Herschel, Dr. Glenn F. Rouse, and Miss H. P. Bell.

Five articles dealing in some manner with the history and present status of lubrication research were prepared for publication in response to specific requests. (Refs. 13 and 40–43, incl.)

In connection with the proposed work on the elements of the mechanics and thermodynamics of lubrication, consideration has been given to questions of nomenclature, definitions, and units, and further applications have been made of the theory of dimen-

[5] The Special Research Committees of the Society were at that time designated as Sub-Committees of the Research Committee.

[6] Member of the committee until 1927 and adviser to the committee since that date.

sions, or dimensional analysis. (Refs. 44, 45.) The theory has been extended to include, as dependent variables (e.g., ordinates of a curve): the hydrostatic pressure in the film, the volume of oil flowing through a bearing due to its own pumping action, and the angle from the line of action of the load to any characteristic point of the film, such as the point of nearest approach and the points of maximum and minimum pressure; and as independent variables (e.g., abscissas of a curve): the inertia, compressibility, and capillarity of a lubricant; its pressure-coefficient of viscosity; gravity, as affecting delivery by oil rings or otherwise; and the static pressure of the oil supply in forced-lubrication systems. The use of the f vs. ZN/P diagram under oiliness conditions has been fully considered. In the extension of dimensional analysis to cover any of the foregoing practical conditions, including oiliness (except in the ideal limiting case where the effect of viscosity totally disappears) it is only necessary to introduce the appropriate additional dimensionless variables, without disturbing ZN/P or its equivalent. This variable, designated by the less convenient notation $\mu n/p$ in the earlier papers (Refs. 36, 22, and 46) can still be retained as the chief argument of a table or the base line of a diagram. As a further simplification the single letter g (for generalized variable) has been proposed in place of ZN/P, so that the customary method of plotting coefficient-of-friction observations can be designated more briefly as the "fg diagram."

A beginning has been made on the mathematical treatment of plastic lubricants, also on problems concerned with the rise of temperature in the oil film. A relation from which it appears, under certain conditions, that the excess of temperature in the oil film relative to the adjacent metal will be proportional to the square of the surface speed and independent of the film thickness was discussed at a meeting of the Committee in December, 1915, but not published. The same or an equivalent formula was independently discovered by Donald Bratt of the Brooklyn Edison Company (unpublished communication, 1927); and A. G. M. Michell (Ref. 47). A solution for the effect of a uniform change of temperature upon the coefficient of friction of a full journal bearing has been published to illustrate a new method of calculation that might be further applied. (Ref. 48, pp. 152–154.[7])

Eight papers on lubrication have been presented for discussion at the meetings of the Society, either by members of the Committee or upon the recommendation of the Committee, during the period elapsed since the publication of the last report, viz.: "Journal Running Positions," by H. A. S. Howarth (Ref. 49); "Friction of Journal Bearings as Influenced by Clearance and Length," by S. A. and T. R. McKee (Ref. 2); "Service Tests on Diesel-Engine Lubricating Oil," also "The Purification of Diesel-Engine Lubricating Oil," by A. E. Flowers and M. A. Dietrich (Refs. 50 and 51); "Grooving Bearings in Machines," also "Performance of Oil-Ring Bearings," by G. B. Karelitz (Refs. 52 and 53); "Progress in Fluid-Film Lubrication," by A. G. M. Michell (Ref. 47); and "A Study of Tin-Base Bearing Metals—II," by G. B. Karelitz and O. W. Ellis (Ref. 53).

Cooperation With Other Committees

The Special Research Committee on Lubrication cooperates with a number of other Committees working in closely related fields. At the present time contact is made with the Special Research Committee on Cutting of Metals through M. D. Hersey; with the Special Research Committee on Worm Gears through Earle Buckingham; with the Research Committee of the Society of Automotive Engineers through H. C. Dickinson; and with Committee D-2 of the American Society for Testing Materials, Petroleum Products and Methods of Testing, through A. E. Flowers, official representative of the A.S.M.E. on Committee D-2.

The following publications recently issued through the medium of various other Committees are of interest in connection with lubrication research: "Cooling and Lubrication of Cutting Tools" (Ref. 54); "Present Practice in the Use of Cutting Fluids," by S. A. McKee (Ref. 55); "Worm Gears—a Study and Review of Existing Data" (Ref. 56); "The Significant Properties of Automotive Lubricants," by H. C. Mougey (Ref. 57); "Progress on Efficiency Tests of Electric Street-Car Trucks," by S. A. McKee (Ref. 58); "Bearing Metals and Bearings," by W. M. Corse (Ref. 59).

Recent Foreign Investigations

At the time of the organization of our Committee 15 years ago the amount of activity in European countries in the field of experimental lubrication research was relatively insignificant. Today, the situation appears to be reversed. No one who has followed the work of such investigators as Hanocq in Belgium, Hardy and Stanton in England, Trillat and Woog in France, Kiesskalt and the Vieweg brothers in Germany, to name only a few, can fail to be impressed with the high standard of talent that is being concentrated upon lubrication problems, the extent to which such investigations are encouraged and promoted by the industries, the engineering societies, and the national government organizations, and the consequent rapid progress that is being made.

References 60–113, inc., will indicate the general character of the investigations in progress in European countries during the past three years. English, French, and German publications are separately grouped and arranged alphabetically by authors. The subject matter of the investigations listed herein includes hydrodynamic theory; the development of testing technique for friction and film thickness measurements; experiments on boundary lubrication, seizure, and high-temperature operation of bearings; and researches on oiliness (onctuosité, Schmierfähigkeit), including adsorption and molecular orientation.

Acknowledgments

Specific acknowledgments have been indicated wherever necessary in the foregoing pages. For cooperation in conducting the work leading up to this report a general acknowledgment is due to the U. S. Bureau of Standards and other organizations represented on the Committee, also to the numerous contributors to the Lubrication Research Fund; and to the Research Associates of this Committee in Washington, D. C., Miss H. P. Bell, Dr. Ronald Bulkley, and Messrs. T. R. McKee and G. H. S. Snyder.

References

1 Progress in Lubrication Research. Fourth Report of A.S.M.E. Special Research Committee on Lubrication. Trans. A.S.M.E., 1928, APM-50-4, pp. 1–5; discussion continued in *Mech. Eng.*, vol. 50, 1928, pp. 682–683.

2 McKee, S. A., and McKee, T. R. Frictional of Journal Bearings as Influenced by Clearance and Length. Trans. A.S.M.E., 1929, APM-51-15, pp. 161–171.

3 Karelitz, G. B. Performance of Oil-Ring Bearings. Trans. A.S.M.E., 1930, APM-52-5, pp. 57–70.

4 Hersey, M. D. Theory of the Torsion and Rolling-Ball Viscosimeters and Their Use in Determining the Effect of Pressure on Viscosity. *J. Wash. Acad. Sci.*, vol. 6, 1916, pp. 525–530.

5 Hyde, J. H. On the Viscosities and Compressibilities of Liquids at High Pressures. Proc. Roy. Soc. A., vol. 97, 1920, pp. 240–259. Appendixes 7, 8, 9, Report of the Lubricants and Lubrication Inquiry Committee, Dept. Sci. Ind. Res., London, 1920, pp. 80–90.

6 Hersey, M. D., and Shore, H. Viscosity of Lubricants Under Pressure. *Mech. Eng.*, vol. 50, 1928, pp. 221–232.

[7] A misprint occurs in Equation [74] of the reference cited. There should be a dot over the C on the left-hand side. C itself denotes the bearing clearance, while $\dot{C}$ represents the fractional change of clearance per unit rise of temperature.

7 Bridgman, P. W. The Effect of Pressure on the Viscosity of Forty-Three Pure Liquids. Proc. Am. Acad. Arts Sci., vol. 61, 1926, pp. 57–99.

8 Kiesskalt, S. Untersuchungen über den Einfluss des Druckes auf die Zähigkeit von Oelen und seine Bedeutung für die Schmiertechnik. Forschungsarbeiten aus dem Gebiete des Ingenieurwesens, no. 291. Berlin: V.D.I., 1927.

9 Appendix No. 1, pp. 2–5 of Reference 1 preceding.

10 Kingsbury, A. Experiments on the Friction of Screws. Trans. A.S.M.E., vol. 17, 1895–1896, pp. 96–116.

11 Kingsbury, A. A New Oil Testing Machine and Some of Its Results. Trans. A.S.M.E., vol. 24, 1903, pp. 143–160.

12 Herschel, W. H. Proc. A.S.T.M., vol. 28, pt. II, 1928, pp. 687–689.

13 Hersey, M. D. Fundamental Action of Lubricants. *Am. Mach.*, vol. 70, June 13 and 20, 1929, pp. 919, 975.

14 The Herschel Oiliness Machine. Bull. No. 112, American Instrument Co., Inc., Washington, D. C., 1929.

15 Vieweg, V. Die Messung der Schmierfähigkeit von Ölen. Techn. Mech. u. Thermodyn., V.D.I., 1930, vol. 1, no. 3, pp. 101–105.

16 Marvin, C. F., Jr. A Machine for Comparing the Lubricating Properties of Oils at High Pressures. *Jour. S.A.E.*, vol. 17, 1925, pp. 287–289.

17 Jervis, W. K. (Pseud.). Friction of Lard and Machine Oils. *Am. Mach.*, vol. 30, pt. I, May 9, 1907, p. 655.

18 McKee, S. A. Performance Characteristics of Journal Bearings When an Abrasive Is in the Lubricant. *Jour. S.A.E.*, vol. 20, Sec. 1, 1927, pp. 3–6; Trans. S.A.E., vol. 22, pt. I, 1927, pp. 73–77.

19 McKee, S. A. The Effect of Running-In on Journal Bearing Performance. *Mech. Eng.*, vol. 49, 1927, pp. 1335–1340; vol. 50, 1927, pp. 528–533.

20 French, H. J. Wear Testing of Metals. Proc. A.S.T.M., vol. 27, pt. II, 1927, pp. 212–234. Cf. also vol. 28, pt. II, 1928, pp. 298–325; vol. 29, pt. II, 1929, pp. 450–486, and *Bur. Standards Jour. Research*, vol. 1, Sept., 1928, RP. No. 13, pp. 343–421.

21 Amsler Wear-Testing Machine. Circular issued by Herman A. Holz, New York, N. Y.

22 Hersey, M. D. On the Laws of Lubrication of Journal Bearings. Trans. A.S.M.E., vol. 37, 1915, pp. 167–202.

23 McKee, S. A. The Effect of the Addition of Kerosene on the Oiliness of Lubricating Oils. *Jour. S.A.E.*, vol. 19, pp. 356–360; Trans. S.A.E., vol. 21, pt. II, 1926, pp. 248–261.

24 Report of A.S.M.E. Sub-Committee on Lubrication. *Mech. Eng.*, vol. 41, 1919, p. 537.

25 Wilson, R. E., and Barnard, D. P., 4th. Methods of Measuring the Property of Oiliness. Trans. S.A.E., vol. 17, pt. I, 1922, pp. 238–281; *Jour. Ind. Eng. Chem.*, vol. 14, 1922, p. 682.

26 Wilson, R. E., and Ries, E. D. Surface Films as Plastic Solids. Colloid Symposium Monograph, vol. I, 1923, pp. 145–173.

27 Bulkley, R., and Bitner, F. G. Surface Tension of Soap Solutions and Its Relation to the Thickness of Adsorbed Films. *Bur. Standards Jour. Research*, vol. 5, October, 1930, RP. 241.

28 Langmuir, I. The Mechanism of the Surface Phenomena of Flotation. Trans. Faraday Soc., vol. 15, 1919–1920, pp. 62–73.

29 Woog, P. Contribution a l'Étude du Graissage. Paris: Delagrave, 1926.

30 Woog, P. Frottement—Graissage—Onctuosité. *La Nature*, vol. 55, 1927, pp. 160–163, 193–199.

31 Kingsbury, A. Experiments With an Air Lubricated Journal. *Jour. Am. Soc. Naval Engs.*, vol. 9, 1897, pp. 1–9.

32 Kingsbury, A. Tests of Large Shaft Bearings. Trans. A.S.M.E., vol. 27, 1906, pp. 425–432.

33 Duff, W. A. Empirical Formula for Viscosity as a Function of Temperature. *Phys. Rev.*, vol. 4, 1897, pp. 404–410.

34 Flowers, A. E. Viscosity Measurement and a New Viscosimeter. Proc. A.S.T.M., vol. 14, pt. II, 1914, pp. 565–616.

35 Flowers, A. E. A Cylinder Friction and Lubrication Testing Apparatus. Proc. A.S.T.M., vol. 15, pt. II, 1915, pp. 400–411.

36 Hersey, M. D. The Laws of Lubrication of Horizontal Journal Bearings. *Jour. Wash. Acad. Sci.*, vol. 4, 1914, pp. 542–552.

37 Howarth, H. A. S. Slow Speed and Other Tests of Kingsbury Thrust Bearings. Trans. A.S.M.E., vol. 41, 1919, pp. 685–707.

38 Howarth, H. A. S., and Ogden, N. Friction Tests of Propeller Thrust Bearings. *Jour. Am. Soc. Naval Engs.*, vol. 34, 1922, pp. 1–15.

39 Linsley, L. J. Investigation of Critical Bearing Pressures Causing Rupture in Lubricating Films. Trans. A.S.M.E., vol. 46, 1924, pp. 855–876.

40 Hersey, M. D. Work of the Bureau of Standards on Friction and Lubrication. *United States Daily*, June 14, 1928, p. 11; *Mech. Eng.*, vol. 50, August, 1928, p. 594.

41 Hersey, M. D. Some Lubrication Developments. Trans. A.S.M.E., vol. 51, 1929, MSP-51-21, pp. 159–161.

42 Fundamental Research on Lubrication. *Jour. Frank. Inst.*, vol. 208, 1929, pp. 422–426.

43 Lubrication Research. Synopsis of address by M. D. Hersey before the Detroit Section, A.S.M.E. *Mech. Eng.*, vol. 52, August, 1930, pp. 772–774.

44 Buckingham, Edgar. Model Experiments and the Forms of Empirical Equations. Trans. A.S.M.E., vol. 37, 1915, pp. 263–296.

45 Bridgman, P. W. Dimensional Analysis. New Haven: *Yale Univ. Press*, 1922.

46 Hersey, M. D. Problems of Lubrication Research. *Jour. Am. Soc. Naval Engrs.*, vol. 35, 1923, pp. 648–674.

47 Michell, A. G. M. Progress in Fluid Film Lubrication. Trans. A.S.M.E., MSP-51-21, 1929, pp. 153–163.

48 Hersey, M. D. Effect of Small Changes in Temperature on the Properties of Bodies. *Bur. Stands. Jour. Research*, vol. 4, 1930, RP. No. 137, pp. 137–156.

49 Howarth, H. A. S. Journal Running Positions. Trans. A.S.M.E., APM-51-3, 1929, pp. 21–33.

50 Flowers, A. E., and Dietrich, M. A. Service Characteristics of Diesel Engine Lubricating Oil. Trans. A.S.M.E., vol. 52, no. 8, 1930, OGP-52-2.

51 Flowers, A. E., and Dietrich, M. A. The Purification of Diesel Engine Lubricating Oil. Presented at the National Oil & Gas Power Meeting of A.S.M.E., State College, Pa. June 12 to 14, 1930.

52 Karelitz, G. B. Grooving Bearings in Machines. Trans. A.S.M.E., MSP-51-5, 1929, pp. 21–29.

53 Karelitz, G. B., and Ellis, O. W. A Study of Tin Base Bearing Metals—II. Trans. A.S.M.E., MSP-52-7, 1930, pp. 87–99.

54 Cooling and Lubrication of Cutting Tools—Progress Report No. 1 of the Sub-Committee on Cutting Fluids of the A.S.M.E. Special Research Committee on Cutting of Metals. Trans. A.S.M.E., MSP-51-8, 1929, pp. 47–58.

55 McKee, S. A. Present Practice in the Use of Cutting Fluids. Progress Report No. 2 of Sub-Committee on Cutting Fluids, A.S.M.E. Special Research Committee on Cutting of Metals. Trans. A.S.M.E., MSP-52-8, 1930, pp. 103–108.

56 Worm Gears—A Study and Review of Existing Data—Progress Report No. 2 of the A.S.M.E. Special Research Committee on Worm Gears. *Mech. Eng.*, vol. 51, no. 3, March, 1929, pp. 210–217.

57 Mougey, H. C. The Significant Properties of Automotive Lubricants. Presented at Detroit Meeting of A.S.T.M., March 19, 1930.

58 McKee, S. A. Progress on Efficiency Tests of Electric Street-Car Trucks. Proc. Am. Elect. Ry. Eng. Assoc., 1929, pp. 554–575. For abstract see *AERA*, Dec., 1929, pp. 742–744.

59 Corse, W. M. Bearing Metals and Bearings. (Am. Chem. Soc., Monograph Series.) New York, Chemical Catalog Co., 1930.

60 Boswall, R. O. The Theory of Film Lubrication. London, Longmans, 1928.

61 Charnock, G. F. Bearings for Line-Shafting. Proc. Inst. of Mech. Engrs., No. 5, 1929, pp. 909–922.

62 Deeley, R. M. Lubrication and Lubricants. *The Engineer*, Oct. 18, 1929, pp. 405–407.

63 Goodman, J. An Experimental Determination of the Distribution and Thickness of the Oil-Film in a Flooded Cylindrical Bearing. Proc. Inst. Civil Engrs., vol. 226, pt. 2, 1929, pp. 242–288.

64 Hardy,[8] W. B., Nottage, M., and Jakeman, C. The Analysis of Commercial Lubricating Oils by Physical Methods. Lubrication Research Technical Paper No. 1. London: Dept. Sci. & Ind. Res., 1930.

65 Hardy, W. B., and Nottage, M. Studies in Adhesion, II. Proc. Roy. Soc. A, vol. 118, 1928, pp. 209–229.

66 Hardy, W. B., and Nottage, M. Studies in Adhesion, III. Proc. Roy. Soc. A, vol. 118, 1928, pp. 607–616.

67 Hodgkinson, F. Journal Bearing Practice. Proc. Inst. Mech. Engrs., Nov. 15, 1929, pp. 843–904. *Engineering*, vol. 128, Nov. 22, 1929, pp. 690–693.

68 Southcombe, J. E. Some Contributions to the Theory and Practice of Lubrication. Trans. North-East Coast Inst. Engineers and Shipbuilders, vol. 44, 1928, pp. 207–228.

69 Stanton, T. E. The Lubrication of Surfaces Under High Loads and Temperatures. *Engineering*, vol. 124, Sept. 2, 1927, pp. 312–313.

70 Stanton, T. E. Lubrication of Cylindrical Bearings. *Engineering*, vol. 126, July 20, 1928, p. 85.

71 Wells, J. H. Kinetic Boundary Friction. *The Engineer*, vol. 147, Apr. 26, 1929, pp. 454–456.

[8] See also Ref. 88.

72 Brillié, H. Nouvelles Recherches, Expérimentales et Théoriques, sur le Film d'Huile et le Graissage des Machines. *Le Génie Civil*, vol. 96, Nos. 21, 22, May 24, 31, 1930, pp. 504–507; 528–530, 534–535; Bull. Technique du Bureau Veritas, June, 1928–Jan., 1929, pp. 105, 130, 151, 170, 198, 215, 237.

73 Brillié, H. Exposé Sommaire de la Theorie de Film d'Huile and Contribution á l'Étude des Coussinets. Bulletin de L'Association Technique Maritime et Aéronautique, No. 33, June, 1929, pp. 671 and 695. See also Brillié, H., and Robb, A. M. Oil Films and Bearings. Trans. Inst. Naval Arch., vol. 71, 1929.

74 Dantin, Ch. Le Graissage Rationnel et les Roulements sur Huile. *Le Génie Civil*, vol. 95, 1929, pp. 34 and 63.

75 Hanocq, Ch. Les Recherches Expérimentales du Laboratoire de Construction des Machines sur le "Frottement Fluide." *Rev. Univ. d. Mines*, vol. 2, 1929, Aug. 1, Sept. 1, and 15, Nos. 3, 5, 6. Reprinted (23 pp.), Liége: H. Vaillant-Carmanne, 1929.

76 Marcelin, A. Les Vernis Superficiels sur l'Eau et les Dimensions Moléculaires. *Comptes Rendus*, vol. 189, July 29, 1929, pp. 236–238.

77 Tilman, J. Considérations sur le Problème du Graissage des Mécanismes. *Rev. Univ. d. Mines*, vol. 3, 1930, Jan. 15, pp. 48–54.

78 Trillat,[9] J. Les Nouvelles Theories et la Technique du Graissage. *Science et Indust.*, No. 158, 1927; No. 169, 1928. Reprinted, 27 pp.

79 Woog,[10] P. Observations sur les Mesures du Coefficient de Frottement des Huiles de Graissage. Internat. Cong. Test. Mats., Amsterdam, vol. 2, 1927, pp. 409–414.

80 Woog, P. De l'Extension des Lubrifiants sur les Surfaces Solides. *Comptes Rendus*, vol. 186, Jan. 9, 1928, pp. 71–73.

81 Woog, P. De l'Extension des Lubrifiants sur les Surfaces Solides. Influences Moléculaires. Role de la Photolyse. *Comptes Rendus*, vol. 189, 1929, pp. 977–979.

82 Blom, A. V. Neuere Theorien über den Aufbau des Schmierölfilmes. Bericht No. 27, der Eidg. Materialprüfungsanstalt E.T.H., 1928, Über die Prüfung von Ölen und Schmiermitteln, pp. 21–23.

83 Cuypers, P. Schmierungsprobleme bei Uhren und Feingeraten. *Deutschen Uhrmacher-Zeitung*, Nos. 35, 36, 37, 1927, pp. 3–11, in reprint.

84 Duffing, G. Die Schmiermittelreibung bei Gleitflächen von endlicher Breite. Vol. 5, Auerbach-Hort, Handbuch der physikalischen und technischen Mechanik, Leipzig: J. A. Barth, 1930.

85 Eisenschitz, R., Rabinowitsch, B., and Weissenberg, K. Zur Analysendes Formänderungswiderstandes. Mittl. der deutschen Materialprüfungsanstalten, vol. IX, Arbeiten aus dem Kaiser Wilhelm-Inst. f. Metallforschung und dem Staatlichen Materialprüfungsamt zu Berlin-Dahlem, Berlin, 1929, pp. 91–94.

86 Gottwein, K. Kühlen und Schmieren bei der Metallbearbeitung. Berlin: V.D.I.-Buchhandlung, 1928, 2d ed., 89 pp.

87 Gottwein, K. Zur Wirkung der Kuhl- und Schmiermittel beim Abdrehen von Stahl. Berichte über betriebswissenschaftliche Arbeiten, vol. 3, 1930, V.D.I. (35 pp.).

88 Hardy, W. B. Statische Reibung und Adsorption. *Koll. Zeit.*, vol. 51, Apr., 1930, pp. 6–9.

89 Herttrich, H. Versuchsanlagen für Maschinen und Maschinenteile. *Maschinenbau*, vol. 8, 1929, pp. 48–53; 119–122.

90 Karplus, H. Der Aufbau der Schmierschicht und die Kolloidgraphitschmierung. *Petroleum Zeit.*, 25, 1929, pp. 375–386.

91 Köhler, W. Die Reibungsverhältnisse des Gleitlagers bei unvollkommener Schmierung. Archiv. f. d. Eisenhüttenwesen, vol. 2, 1928, pp. 159–168.

92 Kyropoulos, S. Untersuchungen über die physikalischen Grundlagen des Schmiervorganges, inbesondere im Explosionsmotor. *Zeit. für Technische Physik*, vol. 10, 1929, pp. 2–18.

93 Kiesskalt,[11] S. Bedeutung der hydrodynamischen Lagerreibungstheorie für die Praxis. *Zeit. d.V.d. Ing.*, vol. 71, 1927, pp. 218–222.

94 Kiesskalt, S. An der Grundlagen der halbflüssigen Reibung. *Zeit. für technische Physik*, vol. 9, 1928, pp. 207–212.

95 Kiesskalt, S. Berechnung der Lagertemperatur. *Petroleum*, vol. 24, No. 27, Sept., 1928, pp. 1234–1235.

96 Kiesskalt, S. Neuere Ergebnisse über die Druckzähigkeit von Olen. *Zeit. d.V.d. Ing.*, vol. 73, 1929, pp. 1502–1503.

97 Kiesskalt, S. Der Druckabhängigkeit der Viskosität. *Petroleum*, vol. 26, No. 11, Mar. 12, 1930, pp. 1–3.

98 Kiesskalt, S. Die Erscheinungen der Lagerreibung- und Schmierung. Wien-Harms, Handbuch der Experimental-Physik, vol. 4, pt. II, Leipzig, 1930.

99 Reichel, W. Bestimmung des für den Walzvorgang günstigsten Kuhl- und Schmiermittels. Chap. IX of Dissertation. Breslau, July, 1930.

100 Schlesinger, G. Wechselwirkung zwischen Werkstoff und Werkzeug. *Werkstatt Tech.*, 1927, p. 611.

101 Schneider, E. Versuche über die Reibung in Gleit- und Rollen-lagern. *Petroleum*, vol. 26, 1930, Nos. 7 and 11, pp. 221–236, 337–348.

102 Stäger, H. Untersuchungen an Dampfturbinen-Oelen. Int. Cong. Test. Mat. Amsterdam, 1927, pp. 457–468.

103 Talmud, D. Über die Orientierung der Moleküle an fester Grenzfläche und über die Reichweite der orientierenden Kräfte. *Zeits. f. Phys. Chem.*, vol. 142, July, 1929, Abt. A. 3, pp. 233–236.

104 Trillat, J. J. Moderne Anschauungen über die Reibung. *Metallwirtschaft*, vol. 7, 1928, pp. 101–108.

105 Vieweg, R., and Vieweg, V. Ueber die Bildung des Oelfilms in Schwinglagern. *Maschinenbau*, vol. 6, 1927, pp. 234–235.

106 Vieweg,[12] V., and Kluge, J. Ueber Messungen der Schmierfähigkeit von Oelen in Lagern. Archiv. f. d. Eisenhuttenwesen, vol. 2, 1929, pp. 1–7.

107 Vieweg, V., and Vieweg, R. Über Lager Versuche. *Maschinenbau*, vol. 8, 1929, pp. 48–53; 119–122.

108 Voitländer, R. Untersuchungen an einem neuen Apparat zur Beurteilung der Schmierfähigkeit von Oelen. Mitteilungen des Hydraulisches Institut der Technischen Hochschule, München, Heft 3, 1929, pp. 145–159.

109 Walger and Schneider. Der Einfluss von Graphit auf die Reibung von Gleitlagern. Berichte über betriebswissenschaftliche Arbeiten, V.D.I., vol. 3, 1930.

110 Wallichs, A., and Krekeler, K. Untersuchung der Kühlöle bei der Zahnradbearbeitung. *Zeit. d.V.d. Ing.*, vol. 73, 1929, pp. 643–647.

111 Weber, L. I., and Lewin, G. Ueber Wandschichtdicken. Koll. Zeit., vol. 50, Mar., 1930, pp. 197–199.

112 Wellner, E. Zur Berechnung von Gleittragzapfen. *Zeit. d.V.d. Ing.*, 72, 1928, pp. 435–440.

113 Wolff, R. Über die Schmierschicht in Gleitlagern und ihr Messung durch Interferenz. *Forschungsarbeiten*, No. 308, 1928, V.D.I.

[9] See also Ref. 104.

[10] See also Ref. 30.

[11] See also Ref. 8.

Discussion

C. G. A. Schmidt, Jr.[13] In reading the paper, the writer fails to find any reference to research activities in connection with the lubrication of anti-friction bearings, and it seems that this phase of the question should receive careful consideration and investigation.

For the past three years this company has designed and marketed a series of roller-bearing rock crushers of the overhead eccentric-shaft type. This shaft operates the moving jaw. This jaw is mounted on the eccentric shaft with spherical roller bearings, and the shaft is supported in the crusher frame in spherical roller bearings. A machine of this type is subjected to unusually severe shock loads, practically of indeterminate quantity, subjecting the bearings to the most severe type of service.

Another serious phase is the presence of a great amount of gritty dust, naturally resulting from the crushing of stone, gravel, and similar materials. For this reason it has been considered the best practice to use grease as a lubricant rather than oil. Fresh grease can be pumped into the bearings every day, thereby forcing out the used grease with any gritty substance that may have accumulated at the outer ends of the bearings. The usual speed of these machines is approximately 300 revolutions per minute, and operating temperatures are expected not to exceed 100 to 125 deg.

The determination of the proper grease lubricants for use under such conditions should be a worthy contribution by your committee.

[12] See also Ref. 15.

[13] Chief Engineer, American Road Machinery Company, Kennett Square, Pa. Mem. A.S.M.E.

LOUIS ILLMER.[14] This report shows intensified activity on part of the Special Research Committee on Lubrication. It is noted that the so-called ZN/P relation is accepted as established and infallible. After carefully analyzing many authoritative friction tests (see "Friction Coefficient Research," S.A.E. Journal, June, 1930), the writer has been unable to find that such relationship does actually prevail as determined over wide-range operating conditions. The mere fact that a straight-line ZN/P plot may be obtained under certain working conditions affords no real proof that a more accurate basis for coordinating friction coefficients may not be afforded by a different and more comprehensive groundwork.

Other aspects of the present issue have been elaborated upon in the published discussion of the writer's paper. The accompanying curves clearly show the deviation from the ZN/P relation to become particularly marked, when the per cent of actual contact area is kept at a relatively low value. The net degree or effective contact ratio is a matter that apparently has been neglected by prior investigators.

It would appear, therefore, that the constancy of the alleged ZN/P ratio is taken too much for granted; possibly this is one reason for having to call upon the corrective "oiliness" function in order to reconcile the present findings. It might be expected, however, that the effect of oiliness should come into play only when the bearing load approaches the seizing pressure. The recent fashion of reporting experimental friction determinations in terms of the composite ZN/P relation alone is thought to be wholly unjustified from a scientific standpoint and makes the results practically useless for test-analysis purposes. It is the writer's opinion that the criticized ZN/P method has been made into a fetish and is unlikely to lead to a complete and satisfying solution of the friction-coefficient problem.

The writer has been unable to locate any reasonably simple and readily understandable treatment of the hydrodynamic theory of lubrication that convincingly proves up the expectation of the ZN/P relationship, especially one applicable to relatively low contact factors. As matters now stand, engineers are still unable to predetermine friction coefficient for many of their common but more-important operating conditions. Hence it may be expedient to wait with the further formulation of a complete theory of lubrication until the laboriously acquired experimental data of others have first been more thoroughly analyzed. A large and valuable mass of such work is already available, and it is not seen wherein continued experimentation without a more concrete objective will allow fully grasping the significance of past test findings.

There is apparently still need for new concepts regarding the inherently complex principles that underly friction behavior. In the present endeavor, it should clearly be kept in mind that the conventional coefficient is not a physical constant in a rational sense, but merely represents a convenient resultant measure of bearing efficiency. The factors that control such behavior center in a number of different variables, of which each such variable exerts an influence at its own distinctive rate, and these combine to produce a composite result. It is believed that after the effect of each such factor is more accurately known, a propitious time will have arrived for definitely formulating a theoretical basis for friction behavior that may or may not justify the use of the ZN/P relation.

R. MACDONALD.[15] The writer would draw the attention of this committee to the necessity of research on troublesome noise. He refers particularly to bearing noise, affected by lubrication.

Since the installation of motor appliances in offices, hospitals, apartments, and private homes, the demand for quiet operation has become more and more imperative. It has been found by experiment in the Westinghouse Laboratories that unnecessary noise may be prevented by lubrication.

There are many instances in industry where noise may be absorbed, and as there seems to be no precedent, much might be accomplished through research on this subject.

D. V. WATERS.[16] In view of the increasing use of anti-friction bearings in machine construction, some information relating to their lubrication is suggested as a fit subject for committee investigation. Some information in regard to the use of greases and blended oils also would be valuable.

Oiliness is defined as "the property which causes a difference in friction when two lubricants having the same viscosity at atmospheric pressure, and at the true temperature of the film, are tested under identical external conditions. The lubricant giving the lesser friction is then considered as possessing the greater oiliness." This is taken to mean, not that two oils, to be comparable in respect to oiliness, must have the same viscosity at the same temperature, but that to compare the oiliness of the two oils, the tests must be made at such temperatures as will make the viscosities the same. The present knowledge of this quality, what it is, how it should be measured and in what units, and what it means in regard to performance of lubricants, is not such as to make it of much immediate value from the practical standpoint, but oiliness research seems to promise considerable future help and should be followed rather closely by those interested in lubrication.

The reference to other research agencies, and especially the appended list of publications on the subject, should prove helpful to those studying the subject.

C. M. LARSON.[17] The question of evaluating film strengths has been one on which our company has been doing considerable research work. The film-strength testing machine (Fig. 5) and the following procedure are being used for testing various products.

The machine consists essentially of a steel pin A, which is turned by suitable means within three steel bushings at a constant speed, running in a bath of oil. There is a housing which acts as a support for the bushings and is substantial enough to keep them rigidly in absolute alignment, at the same time providing ample space for the lubricant being tested. This housing also bears projections for the purpose of providing a fulcrum pin upon which a substantial lever is rotatively mounted. This lever is provided with a counterweight and two poises so located that ratios of 40 to 1 and 10 to 1 may be obtained.

The pin A is exactly 1 in. in diameter and 3 in. long (1 in. by 3 in. ± 0.005), with a hexagonal head $^1/_2$ in. long to provide means for turning.

The bushings completely envelop the pin, the internal diameter being 1 in. plus 0.007 in. (1.007 in.) to allow ample space for viscous lubricants. The two end bushings are held rigidly in a suitable retainer, and the center bushing is held from turning in a suitable link, but is free to move vertically with the link which transmits the load from the lever to the pin.

Disks weighing 25 lb. each are provided, and when these are applied as at C, or upon the 40-to-1 poise, a load of 1000 lb. is imposed upon the projected area of the pin within the center

[14] Development Engineer, Brewer-Titchener Corporation, Cortland, N. Y. Mem. A.S.M.E.

[15] Small Motor Engineer, Westinghouse Elec. & Mfg. Co., Springfield, Mass.

[16] Development Engineer, Western Electric Company, Chicago, Ill. Mem. A.S.M.E.

[17] Supervising Enginer, Sinclair Refining Company, New York, N. Y. Mem. A.S.M.E.

bushing. (This is exactly 1 sq. in.) When the disks are applied as at *B*, or upon the 10-to-1 poise, a load of 250 lb. is imposed upon the same area.

The pin is driven through a flexible coupling link (located within the oil well to facilitate removal of pins) by means of a constant-speed motor and reduction gear at exactly 100 rev. per min. The motor must be of ample capacity so that no variation in speed occurs even though chattering starts, because it has been found that slight variations in speed under the predetermined rate result in actual seizure and damage to the machine.

A voltmeter and an ammeter are provided, and the variation in the current supplied indicates the performance.

The weights are applied one every 10 sec. until the oil film rup tures, which is indicated by the ammeter reading. At this point the load is removed with the motor still operating and the pin

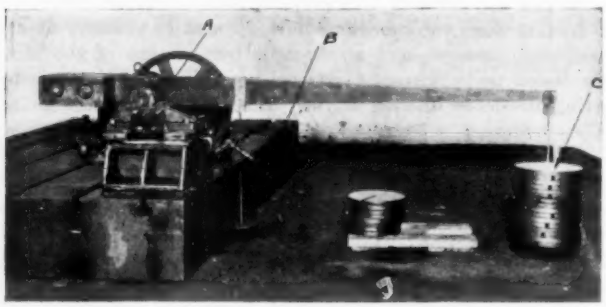


FIG. 5 FILM-STRENGTH TESTING MACHINE

turning, so that a true estimate may be made of performance by comparing the scratches (if any) which show upon the surfaces of the pins.

It is important to note that the 40-to-1-ratio poise is used when examining heavy viscous products such as gear lubricants, etc., and that a combination of the 40-to-1 and the 10-to-1 poises is used when examining oils of comparatively low viscosity. The results are plotted on cross-section paper, applied pressure versus meter reading in amperes.

This machine has the advantage of determining quickly the actual load in pounds per square inch that a lubricant may be expected to sustain without rupture, rapidly and with a minimum amount of oil.

It has been found that petroleum lubricating oils vary in film strength from 3000 lb. per sq. in. up to 12,000 lb. per sq. in. before seizure of the steel shafting and the steel bearing occurs. It also has been found that lubricating oils of the same viscosity made from different crudes have different film strengths. For instance, by taking three oils having a viscosity of S.A.E. 30, it will be found that a naphthene or Texas base oil has a film strength of around 3800 lb. per sq. in., a Mid-Continent 30 pour-point oil a film strength of about 4400 lb. per sq. in., whereas a Pennsylvania oil of 30 pour point has a film strength of approximately 4700 lb. per sq. in.

When the Mid-Continent oil is dewaxed to zero-pour, the film strength is raised to 5100 lb. per sq. in., whereas the Pennsylvania product when dewaxed has a film strength of 5800 lb. per sq. in.

By adding free fatty acid to an oil the film strength is raised approximately 5 per cent. It is possible to take an oil of 100 viscosity at 100 deg. fahr., which normally has a film strength of 3100 lb. per sq. in., and by adding 5 per cent of compound, to raise the film strength to 26,000 lb. per sq. in., and the viscosity of this blended product would not be more than 128 sec. at 100 deg. fahr.

High-film-strength lubricants are being developed for hypoid gears on rear axles, for roll necks, and for heavily loaded journals.

In testing used oils from automotive-engine crankcases, the film-strength testing machine shows definitely that the film strength decreases as the oil decomposes through oxidation and carbonization.

ALPHONSE A. ALDER.[18] The paper makes frequent reference to two physical properties, viscosity and oiliness. Since the terms used in this work will receive wide publicity and will find their way into future literature, the writer would like to suggest to the committee to give due thought to the choice of nomenclature. The well-known physical property of capillarity does not appear in the report. It is the writer's belief that this property is the basic property underlying both viscosity and oiliness. The wetting of surfaces by water, oils, etc., and the non-wetting phenomena of mercury on glass also can be explained by the property of capillarity.

If oiliness and viscosity are shrouded in mystery, research should be directed along these lines so that their phenomena may be coordinated properly with classical physics. The evident question to be answered is, What makes a substance a lubricant?

CLOSURE

The question of including the lubrication of ball and roller bearings within the scope of this Committee may be worthy of further consideration, as suggested both by Mr. Schmidt and by Mr. Waters, but has always hitherto been decided in the negative. The function of oil or grease in such bearings is not primarily one of lubrication, but rather for the purpose of protecting the surfaces and keeping foreign matter out of the bearings.

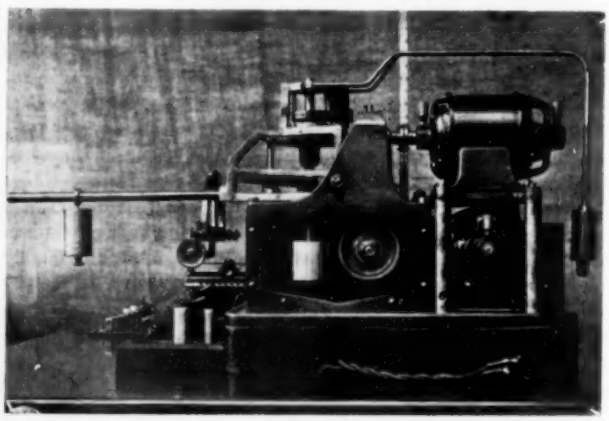


FIG. 6 VOITLÄNDER OILINESS MACHINE

From Mr. Illmer's discussion of the ZN/P diagram we are left in doubt whether he has thoroughly examined the original papers of 1914 and 1915 in which this method of analysis was first outlined. No claim has ever been made in publications of the A.S.M.E. Committee that the plot would necessarily be a straight line, but it is believed to offer a convenient method for separating the effects of viscosity from the unknown factors that remain to be investigated.

The determination of critical loads at which seizure occurs, as described by Mr. Larson, has much in common with the procedure described by H. C. Mougey of the General Motors Research Corporation, for seizure tests on automobile rear-axle differential pins (Ref. 57).

The investigation of conditions at seizure was the purpose in

[18] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

mind in the Dickinson machine as described by C. F. Marvin, Jr. (Ref. 16). All such methods are of great interest. The Committee hopes to ascertain whether any correlation exists between the critical loads so determined, and the solidifying pressure determined hydrostatically, as stated in Table 2 of the present report.

The Committee is indebted to Dr. Rolf Voitländer of the Hydraulisches Institut Technische Hochschule, Munich, for sending a brief description and a photograph, reproduced herewith as Fig. 6, of his improved form of oil-testing apparatus, developed as a continuation of his previous investigation (see Ref. 108). This apparatus will be fully described in the next "Mitteilungen des Hydraulischen Instituts der Technischen Hochschule," München, Heft 4, which is to appear shortly. The previously published work dealt with a comparison of different oils using a given combination of bearing metals. The more recent work with the newest apparatus, as conducted by W. Bürner and reported by Dr. Voitländer, includes also an intercomparison of bearing metals, and very satisfactory results are reported as being obtained.

PME-53-5

Repairing Oil Trunk Lines by Arc Welding

By ALBERT N. HORNE,[1] BARTLESVILLE, OKLA.

Repairs to oil trunk lines should be made in the field without interruption to the service, and the methods of fixing up split joints, leaking couplings, and line breaks by the use of arc welding are described. Pipe line that has been corroded or that has become badly pitted also is patched and repaired by this means. Data are given to show that patch welding pays.



OIL TRUNK LINES are usually operated very nearly continuously and at such capacity factor or on such time schedule that any interruption of service is detrimental to efficiency and may mean loss of revenue. If the interruption is a line break or leak, considerable loss of oil and attendant damages may result therefrom. In years past the general practice in the oil industry has been to have large storage-tank "farms" in the field, along trunk lines, and at the refineries. This system was the result of a policy of getting the oil out of the ground as quickly as possible when a productive area was found and of storing it as a reserve over periods of lean production if the immediate supply were greater than current demand. Storage along a trunk line was partly with the idea of having an intermediate buffer to take care of oil movements in case of line outage. Each trunk-line pump station was equipped with considerable storage for the same purpose, and also to provide two or more working tanks in order to enable a check on receipts and pumpings over each section of line between stations. By receiving the oil in one tank and pumping it out of another tank, losses or shortages were quickly detected and leaks or line breaks more readily located.

Modern methods are tending to place the location and production of crude oil on such a basis of certainty that the storage of large volumes of crude for reserves is not warranted, and modern trunk lines are so constructed that the hazards of operation, such as outages due to split joints, leaking couplings, and line breaks, are practically eliminated. Protection of underground lines against the ravages of corrosion has progressed remarkably in the last few years.

These factors, together with possibilities of unit production of crude, are causing a definite trend toward smaller storage-tank farms. The development of centrifugal pumping equipment makes possible the total elimination of tankage at relay pump stations. Thus the trunk line itself must more nearly than ever operate close to 100 per cent as to capacity and time factors.

These conditions therefore have caused the development of more efficient methods of making emergency repairs and of reconditioning the older lines without taking them out of service for any lengthy period of time.

On the old screw-coupled line the method of repairing a break or a split was to cut out a section of the pipe with pipe cutters, rethread with hand dies, and then couple up with a companion flange. The smaller pit holes were repaired with saddle clamps, and leaky couplings were calked or collar clamps were applied.

With increasing use of welding in pipe-line construction, breaks are repaired by draining the line, cutting out the section, plugging the open ends of the pipe by tamping full of mud to hold back the oil and vapors, and then welding in a section of pipe. On a screw-coupled line this method is probably no more rapid than the screw-flange method, due to having to wait some time under certain conditions to drain the line sufficiently to prevent blowing the mud plug, but it does permit of making the repair with a lesser number of men, as only a bell-hole for the welder is required, whereas to screw-couple it requires that considerable line be stripped and raised from the ditch to facilitate flange-coupling.

In order to keep oil moving through the line continuously and at the same time to recondition a stretch of line economically, both in summer and in winter, the arc-welding machine has been successfully put to use.

A typical method, on an 8-in. loaded line, is to uncover the pipe, raise it from the ditch and place it on skids over the ditch, and then run a traveling cleaning machine over it. The pipe is then carefully inspected, and all serious pit holes, defective couplings, and badly corroded areas are marked with chalk.

A gasoline-engine-driven arc-welding machine mounted on wheels is then pulled alongside the line where the repairs are to be made. The machine is equipped with a long cable so as to work upon as much pipe as possible at one set-up. The metal electrodes are ordinary mild-steel general-purpose rods, from $^3/_{32}$ to $^3/_{16}$ in. in diameter and 14 in. long.

Where the pits are not too deep or too numerous, the repair is made simply by filling the depressions with the arc-weld deposit. On a line under pressure, a rough estimate for safety is a pit that is very little more than half through the pipe wall and not over $1^1/_2$ in. in diameter. No special cleaning of the pit is necessary nor is any special technique required for welding. The striking of the arc expels with explosive force the small amount of oxides and corrosion deposit, and this same material seems to act in the capacity of a flux. The bond is always good, as evidenced by numerous specimens cut out, cross-sectioned, and carefully examined. The actual welding time of the average pit, up to $^1/_2$ in. in diameter, is 15 to 30 sec.

In nearly four years' continuous recondition by one gang, with two welding machines, only two or three "burn throughs" on loaded lines have occurred, and these were not serious.

If the pits are nearly through the pipe wall or are grouped so closely that individual fills are not economical, the next step in the operation is to apply a patch or "half sole."

This patch is made by cutting with the oxyacetylene torch a piece of the desired length and width from the same size of pipe as that under repair. The patch is then placed on a joint of pipe and hammered with a few sledge blows, which shape it

---

[1] General Superintendent, Empire Pipeline Company. Mem. A.S.M.E. Mr. Horne was born Nov. 4, 1894, in Mississippi. He worked in the yellow-pine lumber districts between schooling, in sawmills, planing mills, log camps, office, commissary, etc. He is a graduate of Mississippi A. and M. College, B.S.E.E., 1917. He was with the Tank Corps, U. S. Army, 1918; student engineer with the Empire Companies in 1919 and in various capacities until 1928; General Superintendent of the Empire Pipeline Company since 1928. Mr. Horne is on the A.S.M.E. Research Committee on Automatic Pump Stations and a member of the Reserve Officers Association of the United States and Captain, Infantry Reserve.

Presented at the First National Petroleum Meeting, Bartlesville, Okla., Oct. 6–8, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.



FIG. 1 TYPICAL CORROSION AT LEAKY COLLAR

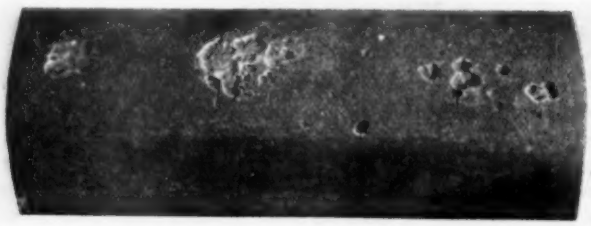


FIG. 2 TYPICAL PITTED PIPE

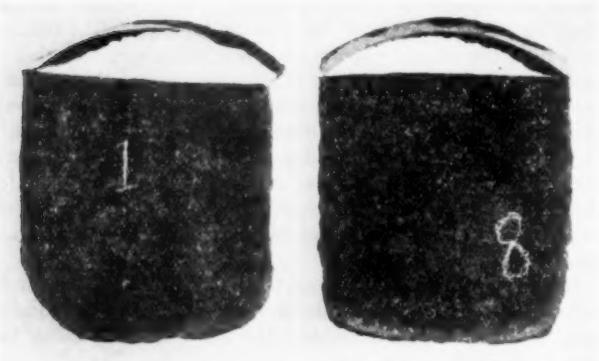


FIG. 3 RUSTED AND PITTED PIPE, WITH CROSS-SECTION

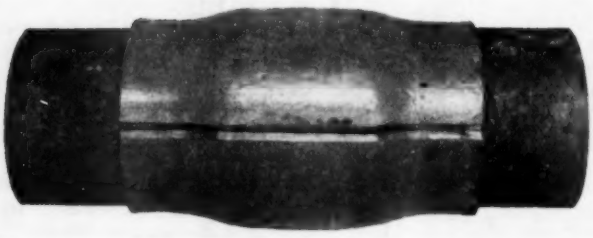


FIG. 4 SPLIT SLEEVE IN PLACE, READY FOR WELDING



FIG. 5 SPLIT SLEEVE IN PLACE, READY FOR WELDING



FIG. 6 SPLIT SLEEVE WELDED INTO PLACE



FIG. 7 PITTED PIPE ARC-WELDED IN THE FIELD

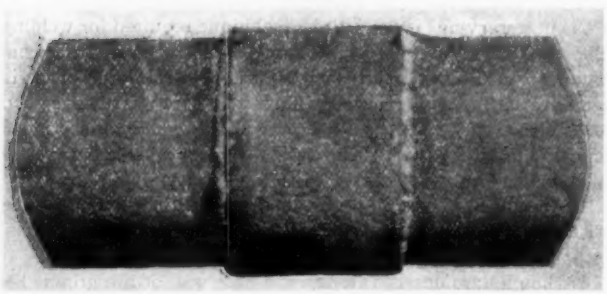


FIG. 8 COLLAR "ROPED" BY ARC-WELDING

nearly enough to the outside diameter on which it is to be applied. A good source of supply is straight junk pipe which may be badly corroded on only one side. For an extended job of reconditioning, patches are cut at one central point in lengths of 6, 5, 4, 3, 2, 1, and $^1/_2$ ft., then distributed to the welding machines.

A patch is clamped in place over a badly corroded area, usually the bottom side of the pipe, and is tacked and then welded to the pipe with one heavy bead. No special cleaning or grinding is necessary on this, and these patches have been subjected directly to pressures far exceeding operating pressures, without failure. It is sometimes necessary to patch the full length of a joint, or even several in succession. This is done by joining up the required lengths of patches.

In some instances, such as creek crossings or areas where the pitting or corrosion covers the entire pipe area, it has been necessary to encase one or more joints entirely. This is done by splitting a joint of 10-in. pipe, then placing it over the pipe, the seams being welded and the ends swaged down and welded to the line.

Leaky couplings that can be calked dry or couplings that appear to be weak due to threads not being run up fully are "roped," by welding each end of the coupling to the pipe. This method does not weaken the joint. Several full-joint roped couplings and one regular screw coupling properly set up were shipped to one of the pipe mills for pull tests, and results show that the roping added considerably to the strength of the screw coupling.

If the coupling cannot be calked dry or is corroded excessively, the repair is made by applying a special split sleeve designed for the purpose. This sleeve is the equivalent of a double-end swage, split in half, with a recess fitting fairly snugly over the coupling and of sufficient length to enclose any exposed threads. The two halves are clamped in place, the seams are welded, and then the ends are welded to the pipe.

These couplings have been directly subjected to over 2000 lb. pressure without failure, and bend and pull tests were entirely satisfactory.

These same split sleeves are being used instead of straps to reinforce welded joints on river crossings. They are welded in place, and the recess or annular space is filled with molten lead through holes cut for the purpose.

Emergency repairs are being made on operating lines by arc-welding patches over holes in place of saddle clamps, and methods are being worked out to repair sizable collar leaks, splits, and cracked or broken welds. Arc-welding machines for emergency-repair gangs are being mounted on regular maintenance-gang trucks and are driven from the truck motor. Machines are on the market which have take-offs for lights and for operating cleaning brushes and grinding wheels.

The economic limit of repairing a line by the various patch methods in the worst places is a cost per foot equal to the cost per foot of cutting out and replacing with new pipe. If continuous operation of a line is a factor, this limit may be exceeded, within reason.

Accurate cost data on recondition work have been kept, and results show that patch welding pays.

A section of 8384 feet of pipe in very bad condition may be cited to give an idea of costs, the repairs being summarized as follows:

Sleeves	23
Feet of weld bead for sleeves	165
Collars roped	72
Feet of weld bead for collars	324
Pits filled, feet of weld bead	168
Patches, square feet	5275
Feet of weld bead for patches	2105
Casing, 10-in., lineal feet	80
Lineal feet 8-in. line reconditioned	8384
Labor	$ 712.26
Material	767.20
Equipment, operation, depreciation, etc	352.75
Total	$1832.21
Cost per lineal foot	$0.2185

PME-53-6a

Evaluation of Steam-Turbine Oil Stability

By C. M. LARSON,[1] NEW YORK, N. Y.

The Navy work-factor method, although sufficient in itself to indicate the rate of decomposition, loss through evaporation, and the corrosive tendencies in steam-turbine oils, fails to measure definitely the resistance to emulsion. Considering demulsibility at 130 deg. fahr. of samples taken every 25 hr. during the 100-hr. Navy work-factor run, however, the emulsion tendencies are recorded, and this demulsibility change is a measure of the stability of steam-turbine oils.



IT IS a well-known fact that various turbine oils in the presence of different combinations of heat, air, and water deteriorate at varying rates. The usual measure of stability is determined by four things: (1) How fast the turbine oil decomposes, (2) resistance to emulsion, (3) loss through evaporation, and (4) whether the oil becomes corrosive. The practical test by placing the oils in the steam turbine and recording the changes in characteristics or analyses, as shown in Fig. 1, is tedious work. It takes months to evaluate turbine oils in actual service. Then, too, due to the variation in conditions under which turbines are operated, it never has been possible to make a direct comparison of the sludge and emulsion percentage increases between two oils while in service or even two lots of the same oil.

In order to simplify the procedure and have a compact test for steam-turbine oils, a Navy work-factor machine, Fig. 2, was built, the work-factor evaluation method being required for Navy approval of lubricating oils when bidding on U.S. Navy contracts. This machine has been used very successfully for evaluating lubricating oils. However, for turbine oils, Navy grading by the formulas, Table 1, did not seem complete, so that a scheme of including the demulsibility at 130 deg. fahr. (Method 320.31, Technical Paper 323-B) on the oil when new, the 25-hr., 50-hr., 75-hr., and 100-hr. periods was added. The constants of the machine are as follows:

Motor speed	3500 r.p.m.
Gear-reducer ratio	1.75 to 1
Journal speed	2000 r.p.m.
Journal diameter (D)	3.82 in.
Journal circumference	12 in.
Bearing length (L)	6 in.
Bearing chord (effective, C)	3.5 in.
Projected area (effective)	21 sq. in.
Load per sq. in.	150 lb.
Quantity of oil for test	5 quarts
Oil pump delivery	1.5 g.p.m.
Oil pressure to bearing (P)	15 lb. per sq. in.
Length of run (continuous)	100 hr.

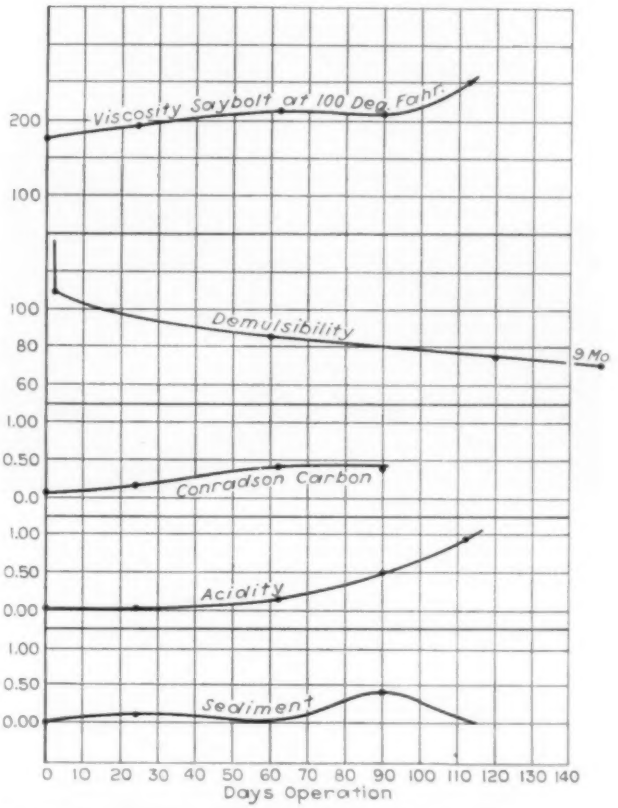


FIG. 1 CHANGE OF CHARACTERISTICS OF TURBINE OIL USED IN 10,000-KVA. STEAM TURBINE

TABLE 1 NAVY ENDURANCE TEST WORK-VALUE FORMULAS

Formula No.	Symbol	General formula	Turbine oils Formula for oils 2110-2190
1	V'_1	$1.00 - \frac{S_f - S_0}{S_0} \times \frac{1}{C}$	$1.00 - \frac{S_f - S_0}{S_0} \times 5$
	V'_1	At 100 deg. fahr.	Work value based on Saybolt viscosity
	V''_1	At 130 deg. fahr.	Use same formula as for V'_1
	$V''_1{}'$	At 210 deg. fahr.	Use same formula as for V'_1
	V_1	$\frac{V'_1{}' + V''_1 + V''_1{}'}{3}$	Average viscosity work value
2	V_2	$\frac{R_0 - C_0}{R_0}$	$\frac{0.55 - C_0}{0.55}$
3	V_3	$\frac{R_f - C_f}{R_f - C_0}$	$\frac{1.05 - C_f}{1.05 - C_0}$
	V_{23}	$\frac{V_2 + V_3}{2}$	Average Conradson carbon work value
4	V'_4	$\frac{R_f - C_f}{R_f - C_0}$	$\frac{1.01 - C_f}{1.01 - C_0}$
	V'_4		Neutralization work value
4	V''_4	$\frac{R_f - C_f}{R_f - C_0}$	$\frac{1.01 - C_f}{1.01 - C_0}$
	V''_4		Precipitation work value
	V_a	$\frac{V_1 + V_{23} + V'_4 + V''_4}{4}$	Average work value

Legend:
S_0 = Initial Saybolt viscosity
S_f = Final Saybolt viscosity
R_0 = Initial rejection point
R_f = Final rejection point
C_0 = Initial content
C_f = Final content

[1] Supervising Engineer, Sinclair Refining Co. Mem. A.S.M.E. Received degree of B.S. in M.E. from Armour Institute of Technology in 1913. Became Lubrication Engineer, The Texas Co., and advanced to Chief Engineer, Chicago District, in charge of lubrication service engineers, comparative tests, lubrication equipment installation, and development of new products. Served as Wing Lubrication Engineer, 1st Lieut., Air Service, U. S. Army, and joined the Sinclair Refining Co. in 1919.

Contributed by the Petroleum Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

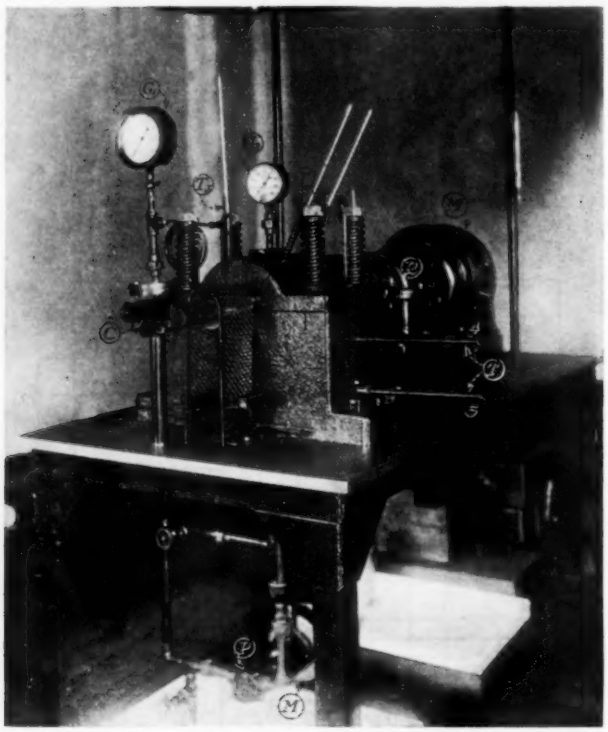
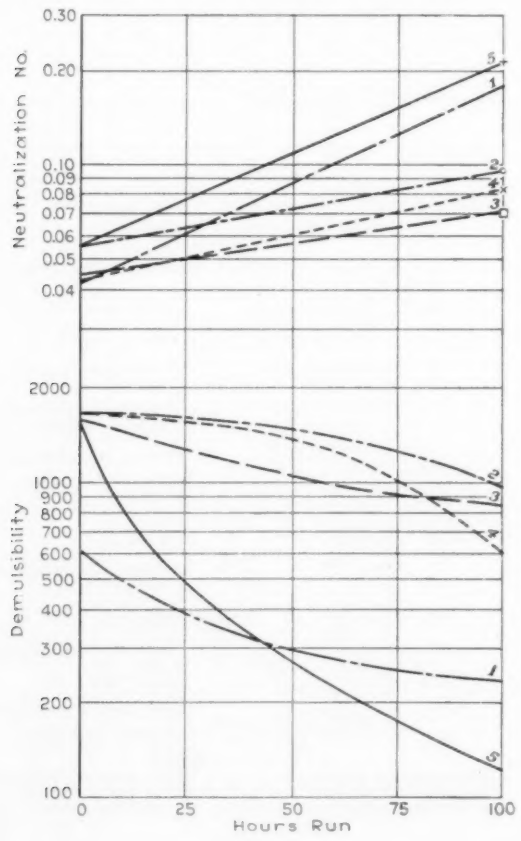


FIG. 2 NAVY WORK-FACTOR MACHINE

(T_1, T_2, and T_3, bearing thermometers; T_4, sump thermometer; T_5, oil-line thermometer; G_1, oil-line pressure gage; G_2, film pressure gage; C, revolution counter; P, Viking motor-driven pump; R, D. O. James speed reducer; M, induction motor.)



By referring to Fig. 3 it will be noted that there are plotted the neutralization numbers of the oil when new and at the end of the 100-hr. Navy work-factor run. These are the numbers that enter into the work-factor evaluation. The lower set of curves are those plotted from the demulsibility numbers determined from the 25-hr. period samples of the same oils whose neutralization numbers are plotted in the top set of curves. In general there is some similarity in decomposition of the oils as based on the neutralization numbers and judged by the demulsibility changes. That is, oils 1 and 5 that have the greatest increase in acidity fall off the most in demulsibility, whereas oils 2, 3, and 4 are the reverse. However, when oils 2 and 3 are

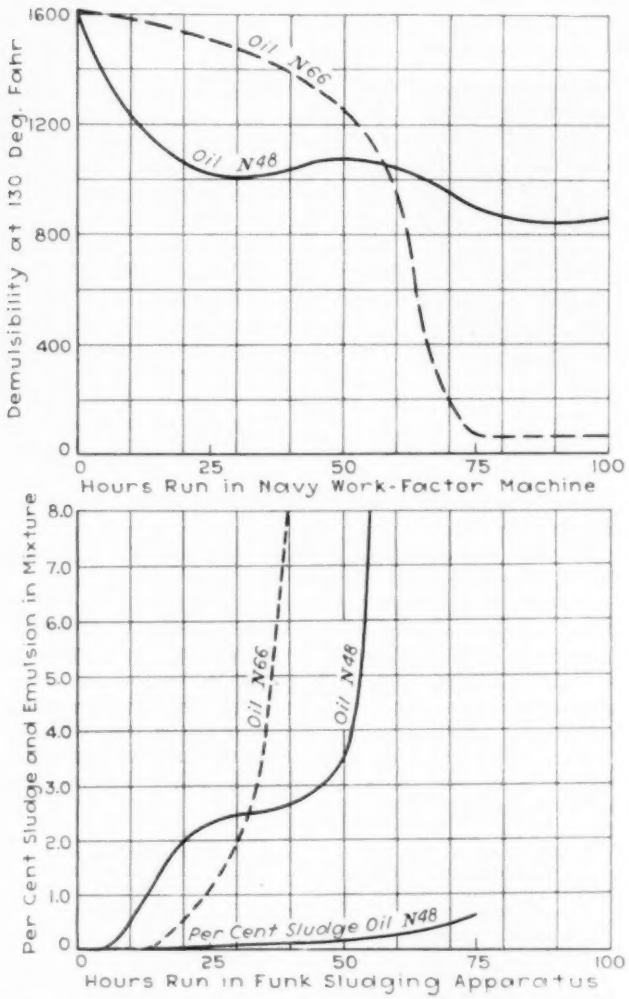


FIG. 4 RESULTS OF TEST OF OILS N-48 AND N-66 IN FUNK SLUDGING-TEST APPARATUS AND NAVY WORK-FACTOR MACHINE

classified by the demulsibility determined by the 25-hr. samples, oil 2 appears to be better than 3 throughout the entire 100-hr. range, and yet making a comparison of the two oils based on the neutralization numbers alone, 3 is better than 2. The same variations are shown by oils 1 and 5.

Now, by studying two turbine oils of exactly the same source and produced under the same refinery methods, with the exception of a slightly different degree of filtration, the value of using the demulsibility determinations of the 25-hr. period samples in addition to the work-factor numbers can be seen. The analyses

FIG. 3 (LEFT) DEMULSIBILITY AND NEUTRALIZATION NUMBERS OF OILS WHEN NEW AND AT THE END OF A 100-HOUR RUN

TABLE 2 TURBINE OIL N-48—CONTROL LABORATORY TEST DATA

Hours	Gravity	Flash	Fire	Viscosity (Saybolt) 100	130	210	Pour	Color	Ash	Carbon	Acid No.	Precipitation No.
0	24.8	430	500	408	177	54.5	20	3½ plus	None	0.09	0.043	None
100	24.6	430	495	414	179	55	20	7 plus	None	0.22	0.07	None

Demulsibility on new sample at 130 deg. fahr.—1620
Demulsibility on used sample at 130 deg. fahr.—840 (Used 100 hr.)

NAVY WORK VALUES

Viscosity (Saybolt) 100	130	210	Average	Conradson carbon Original	Final	Average	Acid No.	Precipitation No.	Work value (avg.)
V'_1	V''_1	V''_1'	V_1	V_2	V_3	V_{23}	V'_4	V''_4	Va
0.925	0.945	0.986	0.952	0.836	0.865	0.850	0.972	1.000	0.944

Demulsibility on 25-hr. sample at 130 deg. fahr.—1020
Demulsibility on 50-hr. sample at 130 deg. fahr.—1080
Demulsibility on 75-hr. sample at 130 deg. fahr.— 900
Demulsibility on 100-hr. sample at 130 deg. fahr.— 840

TABLE 3 TURBINE OIL N-66—CONTROL-LABORATORY TEST DATA

Hours	Gravity	Flash	Fire	Viscosity (Saybolt) 100	130	210	Pour	Color	Ash	Carbon	Acid No.	Precipitation No.
0	25.4	435	510	409	178	54	10	3 plus	None	0.06	0.028	None
100	25.3	430	510	413	180	54	10	6 —	None	0.10	0.042	None

Demulsibility at 130 deg. fahr. on new oil—1620
Demulsibility at 130 deg. fahr. on 25-hr. samp.—1500
Demulsibility at 130 deg. fahr. on 50-hr. samp.—1260
Demulsibility at 130 deg. fahr. on 75-hr. samp.— 60
Demulsibility at 130 deg. fahr. on 100-hr. samp.— 60

NAVY WORK VALUES

Viscosity (Saybolt) 100	130	210	Average	Conradson carbon Original	Final	Average	Acid No.	Precipitation No.	Work value (avg.)
V'_1	V''_1	V''_1'	V_1	V_2	V_3	V_{23}	V'_4	V''_4	Va
0.950	0.945	1.000	0.965	0.891	0.959	0.925	0.986	1.000	0.974

of the new oil and 100-hr. period samples, the Navy work values, and the demulsibility determinations are given in Tables 2 and 3.

Under the Navy work-factor method of evaluation turbine oil N-48 would be classed as having a value of 0.944 (considering the acid number separately 0.972), whereas turbine oil N-66 has a higher work evaluation of 0.974 and an acid number rating of 0.986; so that by work-factor evaluation, turbine oil N-66 is a superior product to oil N-48. However, by determining the demulsibility numbers of the 25-hr. period samples, Tables 2 and 3, N-48 is shown to be superior to turbine oil N-66.

Further to prove the superiority of turbine oil N-48 over N-66, the two oils were checked in the Funk sludging-test apparatus. The results of both the demulsibility numbers and the Funk sludging test proved that turbine oil N-48 outranked N-66, as shown in Fig. 4.

From the foregoing it is rightly concluded that the Navy work-factor method, although sufficient in itself to indicate the rate of decomposition, loss through evaporation, and the corrosive tendencies in steam-turbine oil, fails to measure definitely the resistance to emulsion. By taking into consideration the demulsibility at 130 deg. fahr. of samples taken every 25 hr. during the 100-hr. Navy work-factor run, the emulsion tendencies can be recorded as well, and this demulsibility change is a real measure of the stability of steam-turbine oil.

Discussion

M. A. DIETRICH.[2] In using the increase in neutralization numbers of oils as measures of rates of deterioration, useful information is frequently obtained by determining how much of this acidity is water-soluble. Even though an oil may have a high acidity it does not necessarily follow that the oil is corrosive. The nature of the acidity is important.

[2] Process Laboratory, De Laval Separator Company, Poughkeepsie, N. Y.

[NOTE: Joint discussion of this paper and that of James G. O'Neill by E. H. Thompson, Nevin E. Funk, and G. B. Karelitz will be found following the O'Neill paper.]

PME-53-6b

The Work Factor of Lubricating Oil

By JAMES G. O'NEILL,[1] ANNAPOLIS, MD.

The usefulness of the standard laboratory tests in judging the quality of lubricating oils in the past has been limited, due to widely different crudes from which lubricating oils are produced and to the varied processes used in their refinement. The new method proposed for evaluating lubricating oils seeks to extend the application of the standard laboratory tests in that their results will indicate the service that will be obtained from the oils in actual operation. Increases in viscosity, neutralization number, precipitation number, and per cent of carbon residue of a lubricating oil when subjected to an endurance test, which simulates service conditions as found in a forced-feed lubrication system, are expressed numerically by means of formulas, and the average of these results is called the work factor of the oil. A description of the method and apparatus of the endurance test is given, and the concordance of results obtained is shown when different machines are used to test the same oil. The work factors of turbine, Diesel-engine, and aviation oils are discussed with relation to their performance in actual service. The Navy method for purchase of lubricating oils employing the work factor is discussed, as also are the advantages derived from the use of the work factor.



LUBRICATING OILS for turbines, Diesel engines, and aviation engines may show by the standard tests to have properties that are in accordance with oils that are known to be satisfactory, yet they may be very unsuitable. Specifications for lubricating oils embodying the standard tests are of value when they cover a narrow range of oils made from a crude and by processes of refinement which are known to yield satisfactory oils, but when such specifications are used for oils made from widely different crudes or by widely different methods of refinement, their value is very doubtful. Lubricating oils, when obtained from various refiners having different crudes or different process of refinement, must, in order to insure safe and economical operation of machinery, be subjected to some test which will give true information of their suitability and the quality of the service to be obtained from them; in other words, the test must tell whether or not an oil is satisfactory for use and also properly evaluate satisfactory oils, in order that the consumer may be able to decide what oils are the most economical to purchase.

The suitability and evaluation of lubricating oils are believed to be very well accomplished by means of the work-factor evaluation method. This method is the result of the accumulation of information over a long period of time of the behavior of oils in service and their behavior when subjected to an endurance test which simulates service conditions. In reality it is an accelerated service test for lubricating oils carried out in a rugged machine, equipped with a test journal, which operates at a fixed speed and bearing pressure continuously for 100 hours without means for heating or cooling the oil. It is not the purpose of the endurance test to replace the standard tests for lubricants, but rather to increase the usefulness of these tests in determining the suitability of lubricating oils and in evaluating them.

Apparatus and Method of Test

The apparatus, as shown in Figs. 1, 2, 3, and 4, consists of a machine steel journal *K* 6 in. long and 3.82 in. in diameter, which revolves in a split babbitt-metal bearing, the lower half of which is embedded in the base *A* and the upper half in the bearing cap *U*. The journal is directly connected to a flexible coupling *L* on the shaft of a $3^1/_2$-hp. motor *M*. The bearing pressure is applied to the bearing cap by means of four calibrated springs *D*. The revolutions of the journal are obtained from revolution counter *J*. The bearing cap is fitted with a pressure gage *H* and a mercury glass thermometer *F* for the purpose of ascertaining the oil-film pressure and the temperature of the bearing. The temperature of the oil to the bearing is obtained from mercury glass thermometer *E* inserted in a thermometer well in the inlet oil line *V* to the bearing. The oil to the bearing is supplied from the oil reservoir in the base *A* through the pump *O*. The oil reservoir is equipped with a gage glass *B*, filling connection *C*, and a drain plug *P*, in the oil line *R*. The driving motor *M* is calibrated and equipped with voltmeter and ammeter.

Five quarts of the oil to be tested are poured in the oil reservoir, through the filling connection. The oil is pumped from the reservoir to the oil groove in the bearing. The journal is then started, and the bearing pressure is equally and gradually applied by the springs until the nominal pressure of 150 lb. per sq. in. is obtained, when the speed of the journal is raised to 2000 r.p.m. The test is then started and run continuously for 100 hr., during which time hourly readings are taken of all the instruments. During the test all oils of the ultra-heavy grade (S.U.V. at 130 deg. fahr. 300–340) and heavier are fed at a pressure of 15 lb. and lighter oils at a pressure of 10 lb. During the test the leakage loss of oil amounts to about 5.0 per cent, which leaves sufficient oil for the test. It is essential that no new oil be added during the test. The oil is examined before and after the test in order to determine the nature and extent of the changes in the oil during the test.

Work Factor

The foregoing test was developed in 1920, some of the results obtained being published in 1921.[2] Since that time a large

[1] Chief Chemist, United States Naval Engineering Experiment Station, Annapolis, Md., and technical adviser to the Bureau of Engineering, Navy Department, Washington, D. C., on lubrication. Mr. O'Neill was graduated from Cornell University in 1903. He has been associated as chemist with the Carborundum Company, the Cayuga Lake Cement Company, the Empire Coke Company, and the Semet-Solvay Company. He was the first to devise and operate a successful liquid purification process for coal gas, the description of the process being presented to the Eighth International Congress of Applied Chemistry.

Contributed by the Petroleum Division and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] "Endurance Test of Force Feed Oils," by J. G. O'Neill, Associate, Journal of the American Society of Naval Engineers, vol. 33, no. 2, May, 1921.

number of tests have been conducted on oils using this method. The results of these tests have been used in determining the suitability of oils for naval use. All oils showing abnormal

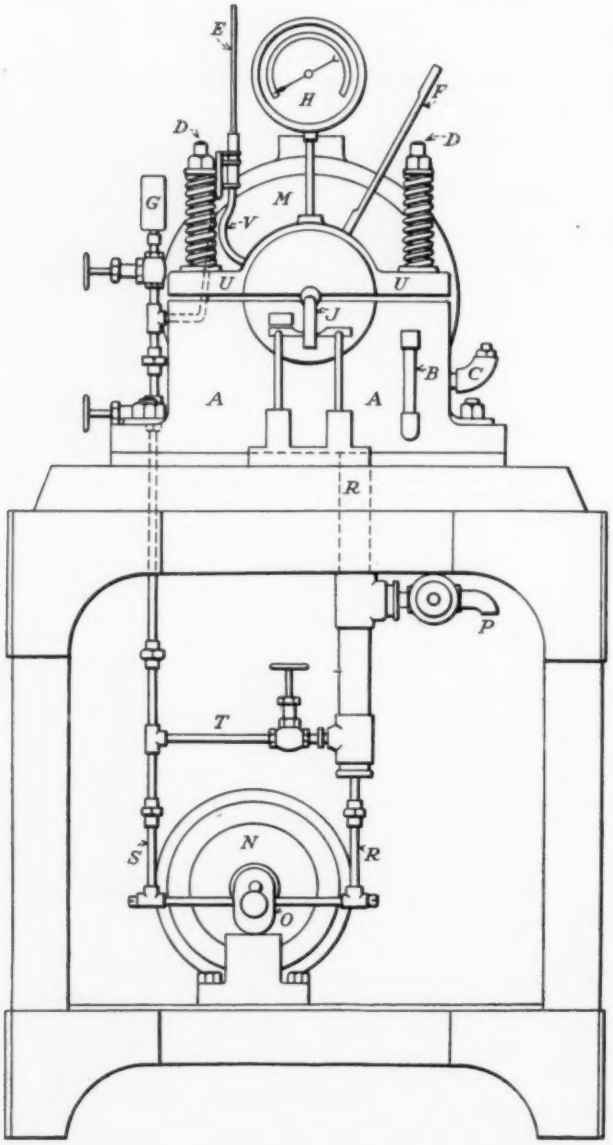


FIG. 1 DIAGRAM OF FRONT VIEW OF ENDURANCE OIL-TESTING MACHINE

increases, when subjected to the test, in viscosity, acidity, asphaltic matter, and carbon residue, were rejected. These tests showed that acceptable oils did not increase beyond the following limits in these properties, when subjected to the test:

Viscosity	All oils below aviation oils in viscosity, per cent	20
	Aviation oils, per cent	25
Neutralization number		1.01
Precipitation number		1.01
Carbon residue, per cent	Oils of viscosity 190 sec. at 130 deg. fahr. and below	1.05
	Oils of viscosity 190 to 340 sec. at 130 deg. fahr.	2.05
	Aviation oils	3.05

In 1928, Keleher and McGeary[3] proposed a work-factor method of evaluating lubricating oil for forced-feed and motor-cylinder use, based upon the oil's behavior in the endurance test as regards increases in viscosity, neutralization number, precipitation number, and per cent of carbon residue. Since the publication of this article, slight changes have been made in the method of calculation of the work factor, and it is considered advisable to give a brief description of the method. Table 1 gives the Navy's classification of the oils, and Table 2 gives the limitations of the necessary characteristics for determining the work factor of these oils.

TABLE 1 U. S. NAVY CLASSIFICATION OF OILS

Navy symbol	Commercial designation	Viscosity at 130 deg. fahr.	S.U.V. at 210 deg. fahr.
1075	Refrigeration oil	65–85	.....
2110	Light forced-feed and motor-cylinder oil	100–115	.....
2135	Medium forced-feed and motor-cylinder oil	125–145	.....
2190	Heavy forced-feed and motor-cylinder oil	180–200	.....
2250	Extra-heavy forced-feed and motor-cylinder oil	240–260	.....
2310	Ultra-heavy forced-feed and motor-cylinder oil	300–340	.....
3080	Winter aviation oil	.....	75–85
3100	Summer aviation oil	.....	90–100
3120	Heavy aviation oil	.....	115–125

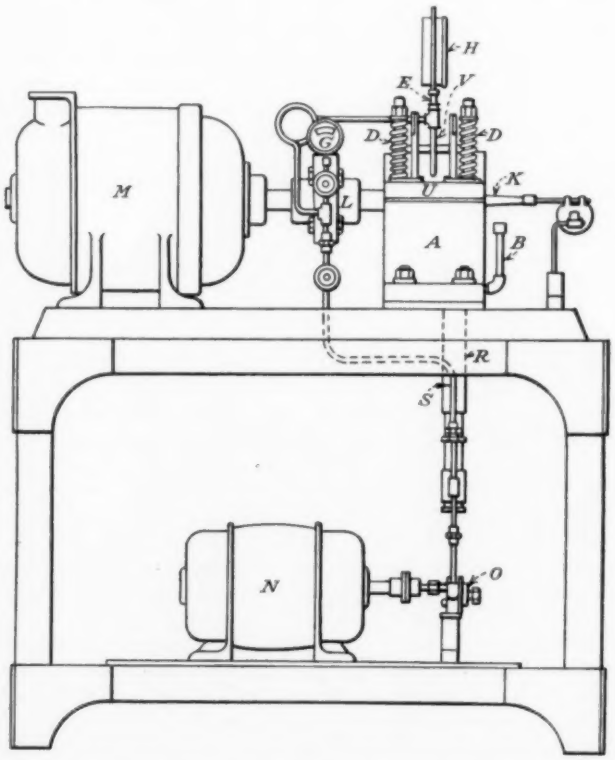


FIG. 2 DIAGRAM OF SIDE VIEW OF ENDURANCE OIL-TESTING MACHINE

The viscosities of the new and used oils are taken at three temperatures, 100, 130, and 210 deg. fahr., and the work value for the oil at each of the temperatures is calculated; the average of the three work values is the viscosity work value.

The formula used for oils 1075 to 2310, inclusive, is as follows:

$$\text{Work value} = 1.00 - \frac{\text{Final viscosity} - \text{initial viscosity}}{\text{Initial viscosity}} \times 5$$

The formula used for oils 3080 to 3120, inclusive, is:

$$\text{Work value} = 1.00 - \frac{\text{Final viscosity} - \text{initial viscosity}}{\text{Initial viscosity}} \times 4$$

[3] "How the Navy Selects Its Lubricants," by Com. T. J. Keleher, U.S.N., and F. M. McGeary, *Manufacturing Industries*, August, 1928.

The neutralization number and precipitation number work values are each determined by the following formula:

$$\text{Work value} = \frac{\text{Final rejection point} - \text{Final content or content of used oil}}{\text{Final rejection point} - \text{Initial content or content of new oil}}$$

TABLE 2 LIMITING VALUES FOR WORK-FACTOR DETERMINATION

		Unused oil; initial rejection point	Used oil after endurance test; final rejection point
S.U.V. viscosities at 100, 130, and 210 deg. fahr. of oils 1075 to 2310, incl.		..	In excess of 20% increase
S.U.V. viscosities at 100, 130, and 210 deg. fahr. of oils 3080 to 3120, incl.		..	In excess of 25% increase
Neutralization number		0.11	1.01
Precipitation number		0.05	1.01
Carbon residue, per cent	Oils 1075 to 2190, incl.	0.55	1.05
	Oils 2250 and 2310	1.05	2.05
	Oils 3080 to 3120, incl.	1.55	3.05

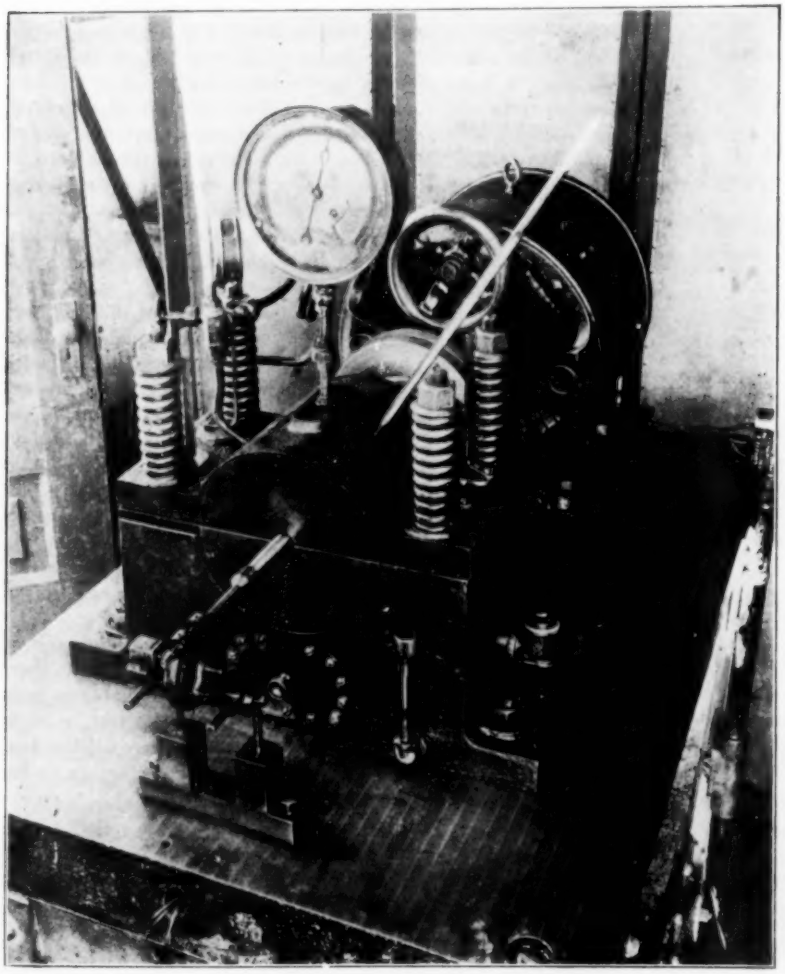


FIG. 3 PHOTOGRAPH OF FRONT VIEW OF ENDURANCE OIL-TESTING MACHINE

The carbon residue work value is calculated from two formulas—one used for the oil when new and the other for the oil after the test; the average of the two being the carbon residue work value.

The formulas are as follows:

$$\text{Work value for new oil} = \frac{\text{Initial rejection point} - \text{Initial content}}{\text{Initial rejection point}}$$

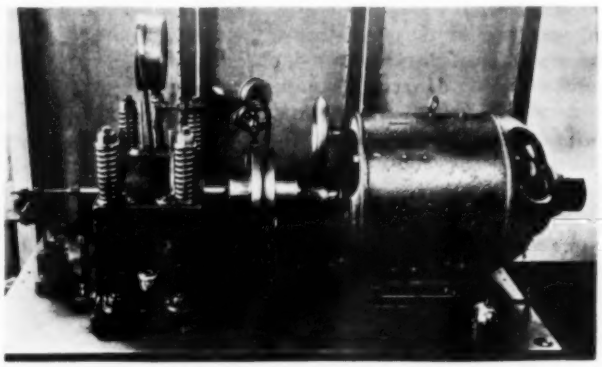


FIG. 4 PHOTOGRAPH OF SIDE VIEW OF ENDURANCE OIL-TESTING MACHINE

$$\text{Work value for used oil} = \frac{\text{Final rejection point} - \text{Final content}}{\text{Final rejection point} - \text{Initial content}}$$

To obtain the average work value of the oil the foregoing results are assembled and expressed as follows:

Work-value viscosity at 100 deg. fahr. ....
Work-value viscosity at 130 deg. fahr. ....
Work-value viscosity at 210 deg. fahr. ....

(*A*) Viscosity work value = $\frac{\text{Total of above work values}}{3}$

(*B*) Neutralization number, work value ....
(*C*) Precipitation number, work value ....
Work value, carbon residue, new oil ....
Work value, carbon residue, used oil ....

(*D*) Carbon residue, work value = $\frac{\text{New and used oil, work values}}{2}$

Total (*A*), (*B*), (*C*), and (*D*) ....

Average work factor of oil = $\frac{\text{Total of work values}}{4}$

The highest work factor possible for an oil is 1.00. Where any of the foregoing work values is less than zero, a zero rating is given. An oil having an average work factor less than 0.50 is not considered suitable for use, nor is any oil which, when subjected to the test, exceeds any of the rejection points shown in Table 2.

CONDITIONS AFFECTING WORK-FACTOR DETERMINATION

In a previous paper,[4] there were discussed the work factors of lubricating oils made from different crudes, the lack of tempera-

[4] "Evaluation of Lubricating Oil by the Work Factor Method," by J. G. O'Neill and F. M. McGeary, *National Petroleum News*, June 11, 1930.

ture control, reproducibility of results on different machines, and the consumption of lubricating oils in service which have different work factors. Since this was published a lubricating oil was subjected to endurance tests in four different machines, all operating at the same time. The work factor for the oil from each machine was determined, and it was found that the results from all four machines varied less than 5 per cent. These tests were run during a hot summer month. Previously, the work factor of the same oil had been determined during a cool fall month. It was found that the average work factor for the four runs during the summertime was 0.73 per cent greater than the work factor of the oil as determined in the fall month. Table 3 gives the results of the endurance tests of the oils, and in Table 4 are shown the calculated work factors of the oils and their properties after being subjected to the endurance test.

TABLE 3 RESULTS OF ENDURANCE TEST
(Oil, Navy Symbol 2110)

Test conducted in	Oct., 1929	July, 1930			
Run on machine No.	3	1	2	3	4
Hours of test	100	100	100	100	100
Average speed of journal, r.p.m.	2034	2008	2035	2020	1955
Average room temperature, deg. fahr.	75	86	86	86	86
Average bearing temperature, deg. fahr.	181	197	181	186	185
Average difference in temperature in deg. fahr. (bearing-room)	106	111	95	100	99
Average oil inlet temperature, deg. fahr.	143	156	161	133	143
Oil pressure to bearing, lb. per sq. in. gage	10	10	10	10	10
Bearing pressure, lb. per sq. in.	150	150	150	150	150
Oil film pressure, lb. per sq. in. gage	240	278	252	259	296
Coefficient of friction	0.0040	0.0040	0.0039	0.0036	0.0037
Condition of bearing at end of test	Bearing in good condition after all tests. No indications of scoring, corrosion, or formation of gummy deposits.				

TABLE 4 PROPERTIES OF OIL AFTER ENDURANCE TEST AND THEIR WORK FACTORS
(Oil, Navy Symbol 2110)

Test conducted in		Oct., 1929	July, 1930			
Run on machine No.	New oil	3	1	2	3	4
A.P.I.	26.07	25.55	25.37	25.55	25.37	25.37
Flash point, open cup, deg. fahr.	335	340	340	340	340	340
Fire point, open cup, deg. fahr.	380	385	385	380	385	385
Pour point, deg. fahr.	5	5	5	5	5	5
Viscosity, S.U.V. at						
100 deg. fahr.	220	253	250	247	244	247
130 deg. fahr.	109	120	120	120	119	119
210 deg. fahr.	46	48	48	48	48	48
		(Dil.)	(Dil.)	(Dil.)	(Dil.)	(Dil.)
Color, N.P.A.	1.5	3.0	3.5	3.0	3.0	3.5
Neutralization No.	0.06	0.16	0.19	0.19	0.19	0.19
Precipitation No.	None	None	None	None	None	None
Carbon residue, per cent	0.08	0.12	0.13	0.11	0.10	0.13
Work Factor Determination						
Work value viscosity at						
100 deg. fahr.	...	0.250	0.318	0.387	0.455	0.389
130 deg. fahr.	...	0.495	0.495	0.495	0.542	0.542
210 deg. fahr.	...	0.783	0.783	0.783	0.783	0.783
(*A*) Viscosity work value	...	0.509	0.532	0.555	0.593	0.571
(*B*) Neutralization No., work value	...	0.894	0.863	0.863	0.863	0.863
(*C*) Precipitation No., work value	...	1.000	1.000	1.000	1.000	1.000
Carbon residue work value:						
New oil	...	0.854	0.854	0.854	0.854	0.854
Used oil	...	0.958	0.948	0.969	0.979	0.948
(*D*) Carbon residue work value	...	0.906	0.901	0.911	0.916	0.901
Total (*A*), (*B*), (*C*), and (*D*)	...	3.309	3.296	3.329	3.372	3.335
Average work factor	...	0.827	0.824	0.832	0.843	0.834

In order to apply the bearing pressure, no clearance can be allowed between the test journal and the bearing. Therefore, the metals of the two members are separated during operation only by the thickness of the oil film. This means that the bearing must be in good condition and run in by means of bearing grinding compound. Immediate seizing will take place between the journal and bearing if the oil pressure ceases for an instant. Good mechanical condition of the endurance machine is necessary if proper results are to be obtained. When starting the test, care must be taken that the bearing pressure is gradually and uniformly applied.

The endurance test offers a convenient means for study of various problems of lubrication, such as the effect on the oil of different bearing metals, the depreciation of the oil with excessive or scant flow of oil to the bearings, the effect of excessive vibration of the journals, and, possibly, the effect of different gases on lubricating oil during operation. It is believed that the solution of these problems will be greatly aided by means of the endurance test and the work factor.

Work Factor of Turbine Oils

Turbine oils, as manufactured in the United States, show work factors from almost 1.0 to well below the allowable minimum of 0.50. The Navy, like most other large consumers of lubricants, purchases, in widely separated sections, lubricating oils which are made from different crudes and by different processes of refinement. In addition, there are many "freak" oils placed on the market, for which manufacturers make marvelous claims. It is therefore necessary that definite information of the suitability and value be obtained of all oils offered for use. It is not practicable to submit all such oils to extended service tests; first, on account of the possibility of casualties, and, second, because of the time and expense involved. Before an oil is placed aboard ship for a service test, there must be confidence that it will give satisfactory service. If an oil during its endurance test increases abnormally in viscosity, in per cent of carbon residue, in acidity, and in "sludging" matter, it is but a matter of good judgment not to submit such an oil to a service test, but to reject it.

In the early days of the endurance test, year of 1924, a heavy turbine oil was submitted to be tested for naval use. The standard tests for physical and chemical properties indicated the oil to be of such excellent quality that it was sent aboard ship before being subjected to the endurance test. The endurance test of the oil was then run, and it was found to have a work factor of 0.32. The report from the ship stated that the oil became thick and heavy, seriously reducing the flow of oil through the gravity flow boxes, also depositing a heavy sludge, the oil purifier separating out 50 gal. of sludge out of a total run-through of 600 gal. A sample of the sludge was analyzed and found to contain 28 per cent oil, 38 per cent water, and 34 per cent asphaltic matter.

The results obtained from the use of the endurance test in the testing of turbine oils have been so satisfactory that there is a growing conviction that this test is more reliable in indicating the quality of an oil than a single service test. Conditions may arise during a service test which may have a deleterious effect on the oil, for which the oil is not responsible; in an endurance test this is not so possible.

Diesel-Engine Lubricating Oil

The lubricating oil in use in Diesel engines undergoes the following changes:

With engine in good condition, piston rings tight, and combustion satisfactory:

Increase in specific gravity. Flash and fire points and pour point not much affected.

Increase in viscosity, neutralization number, precipitation number, and per cent of carbon residue, to practically the same extent as encountered in a lubricating oil in a force-feed turbine lubrication system.

The oil is darkened more than oil used in a turbine system, but not as greatly as oil used in gasoline engines.

Very little contamination from the fuel.
Demulsifying properties depreciated.

With engine in poor condition, piston rings loose, or combustion unsatisfactory:

Specific gravity decreased.
Flash and fire points considerably decreased.
Pour point decreased.
Viscosity decreased.
Neutralization number, precipitation number, and per cent of carbon residue increases with period of use.
The oil is darkened.
Considerable contamination from the fuel.
Demulsifying properties considerably depreciated.

In view of the foregoing, a high-work-factor oil should prove more stable and be a better lubricant than a low-work-factor oil for use in Diesel engines. This has proved to be true, less troubles are experienced in the use of the higher work factor oils, and oils which have passed satisfactory endurance tests have also passed satisfactory service tests in Diesel engines.

Work Factor of Aviation Oils

The Navy has completed a remarkably successful year in aviation. It is believed that the lubricating oil used in the aviation engines contributed somewhat to this success. In the beginning, aviation was handicapped by a lack of knowledge of the proper oil to use for lubrication. The motor industry, while much older, offered little assistance of value. Even such a simple matter as the proper viscosity oil to be used required a long time for its solution. The advance in aviation engine design from the low-compression water-cooled motor to the high-compression water-cooled motor and then to the high-compression air-cooled motor added to the complications. Lubricating oils which appeared to give good service in the water-cooled motor were not satisfactory for the air-cooled motor. Why such oils proved to be unsatisfactory no one knew. An oil subjected to the standard laboratory tests did not differentiate sharply between good and poor lubricating oils. A great deal of idle conversation was indulged in about high flash points, low per cents of carbon residue, and so on, but such did not advance the art, and consequently many things happened to motors which were difficult to explain.

Since the endurance test demonstrated its value in examining turbine and Diesel engine oils, it was thought that it might possibly be of value in determining the quality of aviation oils. Aviation oils in service are contaminated with the gasoline fuel, but this contamination does not generally exceed 3 per cent by volume. The gasoline absorbed by the oil reduces slightly the viscosity and fire point, and very markedly the flash point, closed cup, of the oil. The neutralization number of the oil is not greatly increased due to the short period of use, but the precipitation number and per cent of carbon residue very often increase greatly, dependent on the quality of the oil.

The endurance test of the aviation oils and the calculated work factor showed that oils which had at one time been considered satisfactory, but had been discarded on account of general poor performance and fouling of the engine, had work factors from 0.50 to as low as 0.10, and, on the other hand, lubricating oils which gave excellent performance and would run indefinitely in the engine with scarcely a sign of carbon deposits on the top of the pistons had work factors of approximately 0.90. The high work factor could not be explained by consideration of carbon residue content of the oil alone, for some of the oils which had very low work factors showed less carbon residue content in the original oil than some of the oils which had high work factors. In fact, it was impossible, and is still impossible, to differentiate between good and poor aviation oils by means of the ordinary standard tests, but it is possible to do so by determination of their work factors. The work-factor method of qualifying aviation lubricating oils has so accurately demonstrated its value by a great number of practical tests that it is scarcely necessary to subject them to service tests in order to prove their suitability. Any one familiar with the operation of high-compression aviation engines recognizes the necessity of maintaining the pistons and cylinders free of carbon deposits; a very slight amount of carbon deposit on these parts greatly increases the detonating properties of the gasoline. While all are aware of the necessity of using gasoline which will withstand the high compression without detonation, very few take into consideration the fact that the use of a lubricating oil which produces carbon deposits on pistons or cylinders is just as objectionable as the use of an unsuitable gasoline. The necessity for the use of high-work-factor aviation oils will become more apparent with the increased use of aviation engines of higher compression ratio.

Purchase of Oils on Work-Factor Basis

Lubricating oils are purchased by the Navy on the service cost per gallon, using the following formula:

$$\text{Service cost per gallon} = \frac{\text{Cost per gallon}}{\text{Work factor}}$$

the low service cost per gallon being considered the low bid. This method of purchase has many advantages, viz.:

(*a*) It creates a means to compare quality with price and allows selection of the most economical oil from the consumer's standpoint.

(*b*) It eliminates the personal element in the decision as to acceptability or non-acceptability of a brand of oil.

(*c*) It is an incentive to refiners to produce oils of higher quality, with some hope of remuneration for the better quality of oils.

(*d*) It offers a ready means of evaluating oils, when considering the factors of safety, upkeep and repair of engines, and length of service; the higher work factor oils increasing the safety of operation and length of service and decreasing the cost of upkeep and repairs.

Conclusion

The object of lubrication is to substitute liquid friction for solid friction in operating machinery. To do this satisfactorily, it is necessary that the liquid possess low internal friction and be capable of long use with minimum change in its properties. The very nature of mineral oils in being resistant to chemical action makes them valuable for lubrication, but this resistance also makes it difficult to evolve satisfactory tests for their quality. The stability of an oil must then be recognized as of great value, but the stability must be determined in a manner which is in agreement with its application; not at too high temperatures or in highly concentrated oxygen atmosphere, but under pressures, temperatures, and in atmospheres which have some relation to service conditions. The results obtained from lubricating oil in service have been the governing factors from which the work-factor evaluation method has been evolved. The changes which take place in the oil in the endurance test and the actual service obtained from the oil are co-related, and consequently form the basis for the work-factor evaluation method.

Discussion

E. H. Thompson.[5] The Petroleum Division is fortunate in obtaining research information such as Messrs. O'Neill, Larson,

[5] Lubricating Engineer, Baltimore Copper Smelting and Rolling Company, Baltimore, Md. Mem. A.S.M.E.

and Hersey have given. The pioneer is the milestone along the pathway of progress. His work frequently stands out above that of the many who profit thereby.

The industrial lubrication engineer can improve operation and reduce costs, but in any one given application his progress is limited by commercial costs and other requirements.

Lubrication engineering, however, is a going concern, and is not necessarily stopped permanently by such conditions. These will eventually yield to further improvement by means of the tools which research from time to time gives us. The measurement of oil-film strength is likely to become such a tool.

NEVIN E. FUNK.[6] The oils used in steam turbines are subjected to the deteriorating influences produced by heat, air, and contact with various metals and with water. Heat, air, and contact with some metals cause changes in the character of oil by oxidization, thus producing some harmful and some harmless combinations from the oil structure.

The harmless combinations remain dissolved in the oil, showing their presence by darkening the oil and increasing its acidity, but do not make sludges or emulsions when brought in contact with water.

The harmful combinations not only darken and acidify the oil, but also are changed from a liquid to the form of semi-solid sludges and emulsions by contact with water, or to sludges even when water is not present, if they are of a certain type and have been formed in sufficient amount.

These sludges and emulsions are suspended in the oil and deposited throughout the lubricating system of the turbine, clogging the system and reducing the flow of oil, covering the cooling coils and reducing the heat transfer, thus increasing the oil temperature and accelerating the deterioration process.

The rate at which sludge and emulsions are produced from an oil has a greater bearing on the practical usefulness of that oil than most of its other characteristics; therefore any test that does not in some manner take into consideration the behavior of serviced oil in the presence of moisture cannot accurately determine the service value of the oil.

The writer agrees with Mr. O'Neill's conclusion that endurance tests made under conditions approaching those of operation are best, and he is therefore led to the conclusion that, for the testing of turbine oils, the elimination of so important a factor as moisture from the test apparatus is very apt to lead to erroneous classifications of lubricating oils.

Mr. Larson in his paper has pointed to this fact and has shown that a material classification error results if the emulsifying tendency of the oil is neglected.

G. B. KARELITZ.[7] Any oil user as well as oil manufacturer would like to have some formula or specification that would enable him to determine by laboratory tests the quality of a product as a lubricant. Any work done along these lines is of high value to the industry, particularly when the amount of work and time necessary for a reliable investigation in this field are taken into consideration.

The oil-testing machine described by Mr. O'Neill has been built apparently with a main object of testing turbine oils. It does not take into consideration the quality of oil which is called oiliness. It is true that some investigators found the performance of oils in internal-combustion engines depending on their viscosity, but there are a number of indications that the lubricating quality of an oil in a reciprocating engine depends on oiliness to a marked degree. The machine must necessarily serve, then, as a good controlling apparatus to differentiate between good oils submitted by oil manufacturers from the standpoint of their durability. Is there not a danger that oils may pass the test and show a service cost per gallon lower than others without having the proper oiliness? The paper by Mr. Larson seems to substantiate such a possibility.

The work test is a further step in oil testing, but there is apparently a long way to go before a reliable specification for purchasing lubricants on the basis of laboratory tests alone can be agreed upon.

A. E. FLOWERS.[8] It should be borne in mind that the work factor as described by Mr. O'Neill represents a quite arbitrary choice of factors to be observed and an entirely arbitrary method of combining results of several different tests into a composite number. Since this method is actually employed by the Navy it is very desirable that information should be made available as to the method of test, but a heavy burden of proof lies upon the author to justify his selections for the method of carrying out the test and evaluating the results.

M. A. DIETRICH.[9] Have any tests been made and are figures available which show any definite relation between the work factor and sludge deposition of lubricating oils in Diesel-engine service? It is difficult to see how this test can measure the resistance to breakdown of oils under such temperature conditions as are encountered in Diesel-engine use.

In steam-turbine lubrication the rate of breakdown and formation of sludge is materially affected by the presence or absence of water. The Navy work test will apparently not measure the effect of steam leaks on the rate of deterioration of an oil.

What were the considerations taken into account in the adoption of the final rejection points for neutralization number and carbon residue? The question of what constitutes a maximum upper limit for neutralization number is subject to considerable dispute.

The test is undoubtedly a considerable improvement over the present arbitrary laboratory methods of test.

ALPHONSE A. ADLER.[10] Mr. O'Neill's paper is timely. It shows progress. Perhaps some day we may be able to specify accurately an oil for a particular purpose so as to encourage progressive manufacturers by rewarding them for a meritorious product and at the same time take effective steps in penalizing those whose product borders on the nebulous. It is frequently a difficult matter to deny a product consideration unless there is a simple, cheap, and effective test which is just and which the objectionable product cannot meet by a wide margin.

Discussion of oil qualities in the past were as meaningless as discussions on what pure air is. The science of air conditioning has recognized certain essential qualities like temperature, humidity, air motion, cleanliness, freedom from odors, freedom of dangerous constituents, etc. By an arbitrary rating, for ventilation purposes, agreement was reached by giving each element a certain weight so that several aggregations of the elementary properties might reach the same percentage. The final result is a measure of fitness of the air for the particular purpose. For example, should the air be required for drying, it is possible that the elements pertaining to odor, cleanliness, etc., may not be as important as they are for breathing purposes. Accordingly a new scale of weights would be agreed upon.

[6] Philadelphia Electric Company, Philadelphia, Pa. Mem. A.S.M.E.

[7] Associate Professor, Columbia University, New York, N. Y., and with Westinghouse Elec. & Mfg. Co. Mem. A.S.M.E.

[8] Engineer in Charge of Development, De Laval Separator Company, Poughkeepsie, N. Y. Mem. A.S.M.E.

[9] Process Laboratory, De Laval Separator Company, Poughkeepsie, N. Y.

[10] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

It seems to the writer that some such scheme may be of possible application to oils for specific purposes. One example may suffice. An aircraft oil may be subjected to extremely low temperatures, and it must remain fluid. Another oil may be used in a factory where under no circumstances would the operating temperature fall below about 50 deg. Desirable viscosity at zero would be of no especial value in the latter application.

We should go on with fundamental research to tell the whole story ultimately. In the meantime a partial and somewhat arbitrary analysis is far better than none at all.

Author's Closure

Does the work-factor method of evaluating lubricating oils apply to lubricants for motor cars? The arguments in favor of the affirmative are as follows:

(*a*) The object of lubrication is to substitute liquid friction for solid friction in operating machinery. To do this satisfactorily it is necessary that the oil used for lubrication have low internal friction and be capable of long use with minimum change in its properties. An oil which is unstable and changes rapidly in its properties, when used as a lubricant where the moving parts float on a film of lubricant, is unquestionably not as satisfactory as one which, under the same conditions, shows great resistance to change. The moving parts of a motor-car engine float on a film of oil, and it is reasonable to suppose that an oil which shows great stability in film lubrication will give better service than one of lesser stability. In other words, a high-work-factor oil should be a guarantee of satisfactory operation in any machinery employing film lubrication.

(*b*) Lubricating oil used in the modern Diesel engines is subjected to high heat from the combustion of the fuel and to a fairly long period of service. The conditions under which it is used are between those of a turbine oil and a gasoline-engine oil. During use the main changes in the oil are increases in viscosity, neutralization number, precipitation number, and per cent of carbon residue, or in other words, properties which determine the work factor of the oil; oils having suitable work factors being satisfactory for use in Diesel engines.

(*c*) It is known that certain oils prove excellent lubricants for aircraft engines; they are high-work-factor oils. It is known that certain oils are unsatisfactory for such use; they are low-work-factor oils. Aircraft engines are gasoline engines, and motor-car engines are gasoline engines; however, lubricating oil in an aircraft engine is subjected to more severe service than lubricating oil in a motor car. If it were possible to standardize motor-car-engine operation as thoroughly as is aircraft-engine operation, the quality of the oil, as expressed by the work factor, would be as valuable as it is for aircraft engines. Most of the troubles experienced in motor-car-engine lubrication are due to the use of oil of the wrong viscosity; this trouble is so widespread that it tends to overshadow difficulties which are experienced from the use of poor-quality oils. Consequently, in view of such conditions, it is difficult to obtain widely affirmative data on the value of high-work-factor lubricating oils. The installation on motor cars of reclaimers will tend to standardize their lubrication and will also tend toward the use of more stable or higher-work-factor oils.

It is safe to assume that, if an oil of high work factor is more suitable for use under very severe conditions, as in an aircraft engine, this oil will still be more suitable for use under conditions less severe, but of the same nature, as in motor-car engines.

PME-53-7

Stresses in Dished Heads of Pressure Vessels

By CYRIL O. RHYS,[1] ELIZABETH, N. J.

The paper develops the problem of the proper profile for the dished heads of pressure vessels. The author's company uses large drums for high internal pressures. The results are given of tests made over a period of several years on these drums in which strain-gage readings were taken. The bending stresses therein are more critical than the meridional-loop stresses.



FOR some years past, the problem of the proper profile for the dished heads of boilers and other vessels subjected to internal pressure has been investigated and discussed. As a result a number of papers in technical literature and at least one book, "Ueber die Festigkeit der gewölbten Böden und der Zylinderschale," have appeared. Almost all of these are written in German, and they are not presented in a form convenient for use and reference by the designer. A discussion by Mr. W. M. Coates of a paper presented by Mr. T. W. Greene at the December, 1929, meeting of the A.S.M.E.[2] gives the full theory and shows how it can be applied to heads of elliptical profile.

The main features included in this paper may be described as follows:

The Standard Oil Company (New Jersey) uses large drums for high internal pressures. For several years tests have been made on these and strain-gage readings taken. The drums have sometimes been tested to destruction, and it has been observed that failure has taken place at pressures considerably below what would be predicted by an application of the older formulas, in which the effect of bending stresses is not included.

Most of the heads referred to have profiles made up of two arcs of circles, whose radii are described as the knuckle radius and the crown radius. It is therefore impossible to apply the modern bending-stress theory directly from such a paper as that referred to for elliptical heads. It consequently becomes necessary to develop the theory as applied to these double-radius heads, which have surface discontinuities at the junction of the crown and knuckle as well as at that of the knuckle and shell. This work appears in Appendix No. 1, the final formulas appearing in Table 1, in the body of the paper.

In the following part of the paper, calculations based on these formulas are compared with the results of experiment and test. The effect of such a comparison is best seen by a reference to Fig. 2, in which curves are plotted for the measured and computed strains. As will be observed, the general shapes of the curves show good agreement. The magnitudes of the maximum strains check closely, the differences being on the side of safety.

This appears to show that the theory as developed for the double-radius head corresponds well with the conditions existing in the head of a pressure vessel under load. As the computed strains are obtained directly from the computed stresses by the usual stress-strain relations, it seems reasonable to suppose that the agreement between the computed and actual stresses is equally good. The existing experimental data provide no means for directly checking this, but at all events the theory shows that the pressure causing failure would be accompanied by dangerously high stresses. In other words, the present theory accounts for failure of heads, as the older formulas do not. All the important stresses for the head corresponding with Fig. 2 are plotted on Fig. 1. As will be shown later, the bending stress due to the surface discontinuity between the knuckle and crown (denoted by S'' max) is the greatest unbalanced stress, and will be considered as governing design.

For design purposes a regular procedure is worked out in the body of the paper. It is shown that (for a given diameter and depth of head) particular knuckle and crown radii exist which will make the governing stress a minimum. The plate thickness is calculated from these by the simple relationship

$$S'' \max = \frac{0.147\ PA^2K}{at}$$

which appears in Table 1, the proper radii, as explained, being taken from Table 3. The thickness being determined, other formulas given in Table 1 are used to obtain the remaining important stresses. It can then be ascertained whether, in any particular case, S'' max is or is not the greatest stress existing in the head. The design procedure, in general, leads to proportions for which all the important stresses are of the same order of magnitude, S'' max being somewhat greater than any other net stress.

An interesting comparison can be made of the different thicknesses obtained by the A.S.M.E. rule and the theory including bending stresses. This occupies the last section of the main body of the paper. Examples, illustrated by Figs. 7, 8, 9, 10, and 11, are worked out, which show considerable discrepancies for heads of small knuckle radius. As the best shape of head is approached, these differences tend to disappear.

As already stated, the derivations of all the formulas used are given in Appendix No. 1. It has not been found necessary, as in most of the literature referred to, to go into the somewhat complicated and difficult theory of thin shells. As will be seen, the derivation requires only the ordinary theory of bending, the usual stress-strain relations, and the solution of a simple differential equation.

Appendix 1 also contains the derivation of the equations used in selecting the least value of the governing stress S max, from which Table 3 has been prepared. The recognition of this

[1] General Engineering Department, Standard Oil Development Company. Mr. Rhys was born in London, England, on Jan. 20, 1882. From 1906–1910 he was in the employ of the Globe Pneumatic Engineering Company in London, handling nearly all their designing work. In 1910 he came to the United States and for three years was in the maintenance department of the William Cramp & Sons Ship and Engine Building Company. During 1913 he did engine- and boiler-testing work for the Bowers Southern Dredging Company at Galveston, Tex. He then became a designer of industrial machinery for the Diamond Match Company at Barberton, Ohio, and remained with them until 1918. Shortly after this he became associated with the United Shoe Machinery Company at Beverly, Mass., working in the experimental department. In 1922 he joined the faculty of the Georgia School of Technology, at Atlanta, Ga. In 1924 he entered the employment of the Standard Oil Company.

[2] A.S.M.E. Trans., APM-51-13, p. 143.

Contributed by the Petroleum Division and presented at the Hartford, Conn., Meeting, June 1–3, 1931, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

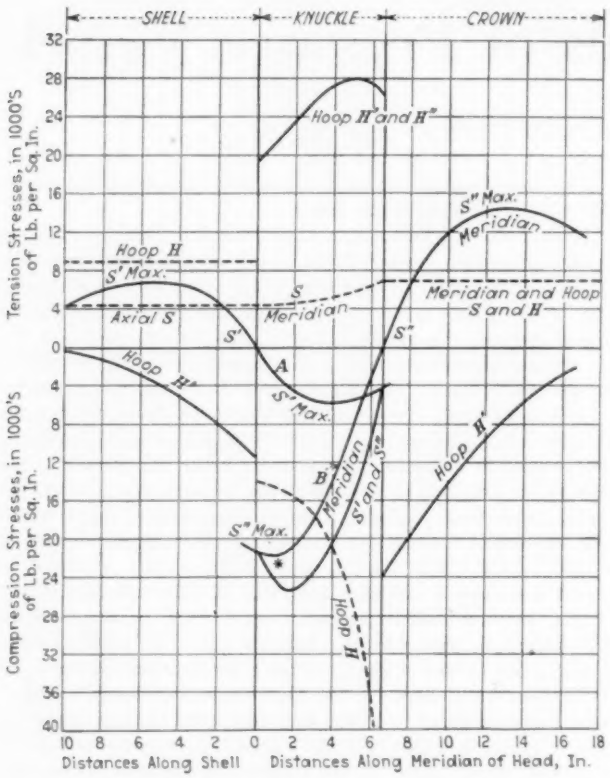


FIG. 1 OUTSIDE BENDING STRESSES IN 6-FT. BY 40-FT. SOAKING DRUM, PRESSURE 500 LB. PER SQ. IN.
(*A*, effect of shell and knuckle; *B*, effect of crown and knuckle.)

minimum and the method for finding it are believed to be new.

To sum up, the present paper can be divided up into five main sections:

1 Development of the bending-stress theory of heads for double-radius head profiles.

2 Comparison of theoretical and experimental results.

3 Development of a method for obtaining the profile of least stress for given diameter and depth of head.

4 Formulation of a procedure for design calculation.

5 Comparison of head thicknesses obtained by the A.S.M.E. rule and the bending-stress theory.

BENDING-STRESS THEORY OF HEADS FOR DOUBLE-RADIUS HEAD PROFILES

All the mathematical work for this is shown in Appendix No. 1. Table 1 contains the formulas for the different stresses existing in the head. Table 2 consists of a list of the symbols used in Table 1, together with their physical meaning.

On Fig. 1 are plotted curves showing all the important bending and membrane stresses for a typical drum and double-radius head. Its dimensions, etc., are given, and it is discussed from a different standpoint, under the heading "Comparison of Theoretical and Experimental Results."

The following discussion of this chart may be of assistance in giving an insight into the bending-stress theory of dished heads. The curves shown in full lines are bending stresses and those in dotted lines are membrane stresses. A membrane stress may be defined as a stress accompanied by a purely radial displacement of a point in the material. Instances of this would be given by the stresses in a soap bubble or a bladder subjected to internal pressure. The material consists of a flexible continuous envelope or surface, which can be stressed by stretching. By a continuous surface is meant a surface whose radius of curvature undergoes no abrupt changes in either the unstrained or the strained state. It can easily be seen that if points on such a surface move out in directions normal to the surface, their distance apart (measured on the surface) will increase, and the material will be stretched. The radius of curvature of the surface will increase, so that the material can be considered as "bent," but a flexible surface offers no resistance to bending, and no bending stresses occur.

The foregoing applies to a thin steel shell, with a continuous surface. The membrane stresses exist as described, and since the material is not flexible, there will be bending stresses. As, however, the normal or radial displacements are very small, the change in radius of curvature is relatively insignificant and the bending stresses almost negligible. When, however, one comes to the kind of surface involved by the combination of a cylindrical shell, attached to a double-radius head, there are discontinuities or sudden changes in the radius of curvature. One of these occurs at the junction of the knuckle and crown, and another at the junction of the knuckle and shell. The shell, knuckle, or crown, considered alone, is a continuous surface, and would have only radial displacements, but those in the sharply curved, stiff knuckle would be exceeded by those in the less curved, more flexible shell or crown. Supposing no separation at the junctions to occur, it is easy to see that the knuckle would be bent outward near the junctions, like a trumpet, and the shell and crown would be bent in. Bending stresses of con-

TABLE 1 STRESSES IN HEADS AND SHELLS OF PRESSURE VESSELS

	Shell	Knuckle	Crown
H	$\frac{Pa}{t}$	$S(2 - R/r)$	$\frac{PA}{2t}$
H max		$S(2 - A/r)$	
S	$\frac{Pa}{2t}$	$\frac{PR}{2t}$	$\frac{PA}{2t}$
S max		$\frac{PA}{2t}$	
H'	$\frac{-Pa^2k}{4t}\,\epsilon^{-n^\circ s}\cos n^\circ s$	$\frac{Pa^2k}{4Rt}\,\epsilon^{-X}\cos X$	
H' max	$\frac{-Pak}{4t}$	$\frac{Pak}{4t}$	
S'	$\frac{3Pk}{4n^{\circ 2}t^2}\,\epsilon^{-n^\circ s}\sin n^\circ s$	$\frac{3Pa^2k}{4Rt\sqrt{3(1-m^2)}}\,\epsilon^{-X}\sin X$	
S' max	$\frac{0.147\,Pak}{t}$	$\frac{0.147\,Pak}{t}$	
H''		$\frac{3PA^2K}{4at}\,\epsilon^{-X}\cos X$	$\frac{-PaK}{4t}\,\epsilon^{-n's}\cos n's$
H'' max		$\frac{PA^2K}{4at}$	$\frac{-PAK}{4t}$
S''		$\frac{3PA^2K}{4Rt\sqrt{3(1-m^2)}}\,\epsilon^{-X}\sin X$	$\frac{3PK}{4n'^2t^2}\,\epsilon^{-n's}\sin n's$
S'' max		$\frac{0.147PA^2K}{at}$	$\frac{0.147PAK}{t}$
X		$\int n.ds$	
n°, n, n'	$n^\circ = \sqrt[4]{\frac{3(1-m^2)}{a^2t^2}}$ = a constant	$n = \sqrt[4]{\frac{3(1-m^2)}{R^2t^2}} = f(R)$	$n' = \sqrt[4]{\frac{3(1-m^2)}{A^2t^2}}$ = a constant

NOTE: For elliptical heads substitute K^2 for k. Expressions in K do not apply.

TABLE 2 LIST OF SYMBOLS

A	= radius of crown
a	= radius of cylindrical shell
b	= depth of dished head
e	= base of natural logarithms
H	= membrane hoop stress
H'	= bending hoop stress due to shell-knuckle edge
H''	= bending hoop stress due to crown-knuckle edge
K	= ratio $(A/r - 1)$
k	= ratio a/r
K	= ratio a/b
m	= Poisson's ratio
P	= internal pressure
R	= hoop radius of curvature of knuckle
r	= meridional radius of curvature of knuckle
s	= distance measured from either edge of knuckle, along knuckle, shell or crown surface
S	= membrane meridional stress
S'	= bending meridionol stress due to shell-knuckle edge
S''	= bending meridionol stress due to crown-knuckle edge
t	= thickness of metal—the same in head and shell
u	= ratio (depth of head)/(shell radius) = b/a
w	= radial displacement accompanying bending stresses
w'	= radial displacement accompanying membrane stresses.

TABLE 3 SELECTION OF OPTIMUM HEAD

1 u	2 r	3 A	4 S'' max	5 S' max	6 S' max
0.1	0.283	66.5	15,200	52	147
0.2	0.65	33.7	842	22.6	36.8
0.3	1.13	23.1	152	13.0	16.3
0.4	1.78	18.1	44	8.25	9.2
0.5	2.59	15.2	16.50	5.66	5.88
0.6	3.76	13.6	7.12	3.9	4.08
0.7	4.96	12.2	3.20	2.96	3.0
0.8	6.2	11.1	1.20	2.37	2.3
0.9	7.63	10.4	0.58	1.92	1.81
1.0	10.00	10.0	0	1.47	1.47

NOTE: See Table 2 for meaning of symbols used.

siderable magnitude are set up in this way. Furthermore, such bending must be accompanied by changes in diameter and consequent large hoop stresses. These are tensile in the knuckle and compressive in the shell and crown.

Consider the curve A on Fig. 1. This represents the outside bending stresses set up by the mutual constraints of the shell and knuckle. It is zero at the junction, since along this boundary the shell and knuckle load each other somewhat after the manner of cantilevers. Going away from the boundary, along the shell or along the knuckle, the stress reaches a maximum and then dies away, as would be expected. The trumpet shape assumed by the knuckle would imply compression, and the inward bending of the shell tension.

Curve B, for the knuckle and crown, is of a character precisely similar to curve A, and the same features apply to it. Curves A and B are combined in the knuckle under S' and S''.

Curve H', in the shell, is the compressive hoop stress set up by the bending inward of the shell end. It is greatest at the boundary, where the decrease in diameter is greatest, and dies away as the decrease becomes less. Curve H'' for the crown is precisely similar. These curves are compressive at any point in the thickness of the plating, and do not change sign for the inside and outside, as the bending stresses do.

Corresponding tensile hoop stresses exist in the knuckle. They are combined under H' and H''.

As these bending hoop stresses are not zero at the boundaries, they illustrate well the discontinuities due to abrupt changes of curvature.

The dotted horizontal lines in the shell are the ordinary hoop and axial stresses in a thin cylinder under internal pressure. They are computed by well-known simple formulas.

The dotted curve in the knuckle marked "S Meridian" is the stress acting tangent to the surface in a plane containing the drum axis. The dotted curve in the knuckle marked "Hoop H" is the membrane hoop stress. It is not evident at first sight that this should be compressive. Consider, however, the shell and crown to be displaced radially outward. Their centers of curvature will still remain on the drum axis, as in their unstrained state, but points on their junctions with the knuckle will be further apart. The knuckle must adjust itself so as still to connect the shell and crown. From a consideration of Fig. 12, it seems almost certain that it does this by shifting the inner end of the knuckle radius and also assuming a flatter curvature. This flattening of the curvature appears to imply a reduction in diameter, with consequent hoop compression. With this and with the trumpet shape due to bending stresses, the strain displacements in the knuckle are of a complicated character, and it is not surprising that, with a sharply curved, stiff form, great stresses result. An inspection of the broken head, Fig. 13, will show that the complete line of fracture lies wholly within the knuckle.

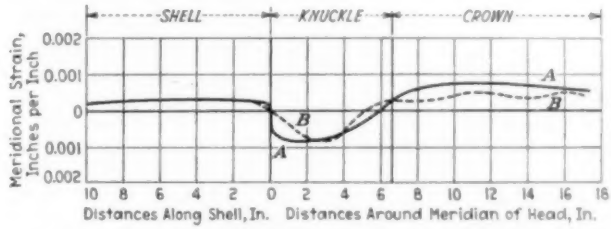


FIG. 2 OUTSIDE MERIDIONAL STRAINS IN HEAD AND SHELL OF 6-FT. BY 40-FT. SOAKING DRUM, PRESSURE 500 LB. PER SQ. IN.
(A, calculated—modulus of elasticity 30,000,000 lb. per sq. in., Poisson's ratio 0.3; B, measured.)

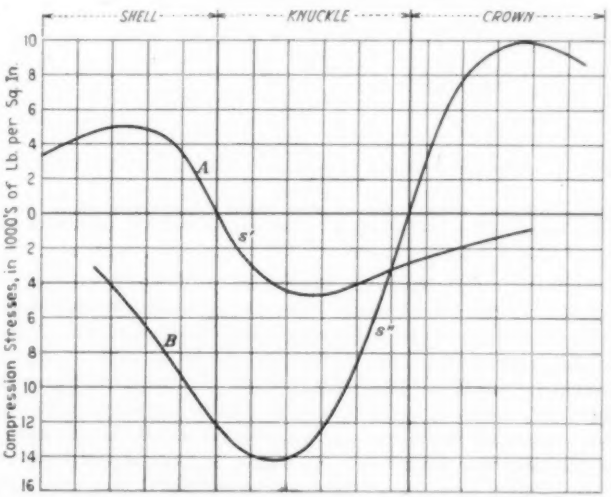


FIG. 3 BENDING STRESSES IN 6-FT. BY 40-FT. SOAKING DRUM
(Knuckle radius, 9.4 in.; crown radius, 55 in.; pressure, 500 lb. per sq. in.; metal, 2 in. thick. A, effect of shell and knuckle; B, effect of crown and knuckle.)

The dotted horizontal line in the crown marked "Meridian and Hoop S and H" (see Fig. 1) represents membrane stresses in a spherical shell under internal pressure. The radius of curvature is constant all over this part of the surface, and the stresses are the same in all directions. All the curves on Fig. 1 are calculated from the corresponding formulas in Table 1. The symbols S, S', etc., H, H', etc., occurring in both Table 1 and Fig. 1, will assist identification.

COMPARISON OF THEORETICAL AND EXPERIMENTAL RESULTS

The following data were taken from a drum subjected to a hydraulic test:

Diameter of cylindrical shell, ft	6
Length, ft	40
Crown radius, in	56
Knuckle radius, in	7

Thickness of head or shell, in.	2
Ratio, head radius to head depth	2
Internal pressure, lb. per sq. in.	500

The stresses plotted in Fig. 1 were calculated from the foregoing. From these, by the usual stress-strain relations, were obtained the strains which are shown in Fig. 2 in comparison with strain-gage readings taken from the drum while under load. The good agreement of theory and experiment in Fig. 2 has already been pointed out.

It will be seen from Fig. 1 that the largest unbalanced stresses are the meridian bending stresses, which reach a value in the

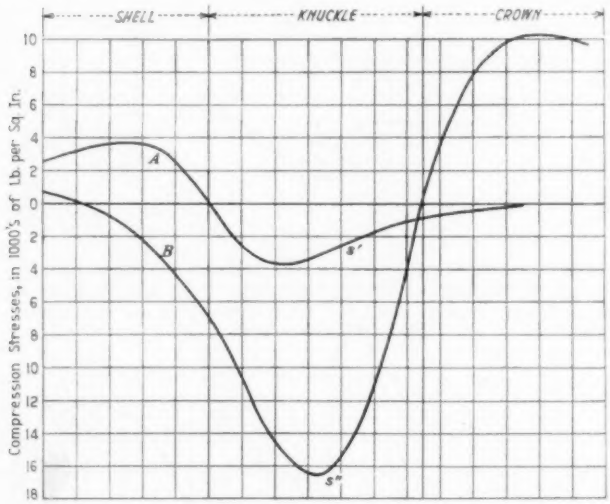


FIG. 4 OUTSIDE BENDING STRESSES IN 6-FT. BY 40-FT. SOAKING DRUM

(Knuckle radius, 12.7 in.; crown radius, 66.5 in.; shell radius, 36 in.; pressure, 500 lb. per sq. in. *A*, effect of shell and knuckle; *B*, effect of crown and knuckle.)

knuckle of about 25,000 lb. per sq. in. This is for an internal pressure of 500 lb. per sq. in. The pressure was increased up to 1600 lb. per sq. in., when failure of the head took place. If the stress-strain ratio remained constant, the stress at this pressure would be 75,000 to 80,000 lb. per sq. in. As is well known, however, the theory only applies up to the elastic limit (in this case about 32,000 lb. per sq. in.), beyond which the calculated stress in general comes out too high. In the present instance the ultimate strength of the material is 50,000 lb. per sq. in. Although the test data do not permit of any direct check as to stress magnitude, the theory at least indicates that rupture would be expected for the pressure which actually accompanied it. The photograph of the broken head (see Fig. 13) has every indication that failure was caused by a high meridional bending stress in the knuckle.

Following are given some examples of applying the formula

$$S'' \text{ max} = \frac{0.147\, PA^2K}{at}$$

to the results of tests. Although the stress equations do not apply beyond the elastic limit of the material, the following results show that failure would be expected.

A head of the dimensions

Crown radius	= 56	in.
Knuckle radius	= 6	in.
Shell radius	= 36	in.
Thickness	= 1.62	in.
Depth of head	= 18	in.

failed at a pressure of 1200 lb. per sq. in. Applying the formula

$$S'' \text{ max} = \frac{0.147\, PA^2K}{at}$$

there is obtained

$$S'' \text{ max} = \frac{0.147 \times 1200 \times 3136 \times 8.33}{36 \times 1.62} = 79{,}000 \text{ lb. per sq. in.}$$

This would certainly indicate rupture, and the nature of the fracture corresponds with failure by meridional bending stress.

After breaking, this head was replaced by one with a 10-in. knuckle radius. From Table 3 it is seen that this is very close to the "best head," which would have a knuckle radius of $\frac{2.59 \times 36}{10} = 9.4$. At 1800 lb. per sq. in., leakage at the shell joints began to occur, and the test was interrupted to make provision for preventing this. Applying the formula, there is obtained, for the stress at this juncture

$$S'' \text{ max} = \frac{0.147 \times 1800 \times 3136 \times 4.6}{36 \times 2} = 53{,}000 \text{ lb. per sq. in.}$$

On continuation of the test, this head finally broke at 2350 lb. per sq. in. From the formula a stress would be indicated of

$$S'' \text{ max} = \frac{0.147 \times 2350 \times 3136 \times 4.6}{36 \times 2} = 69{,}000 \text{ lb. per sq. in.}$$

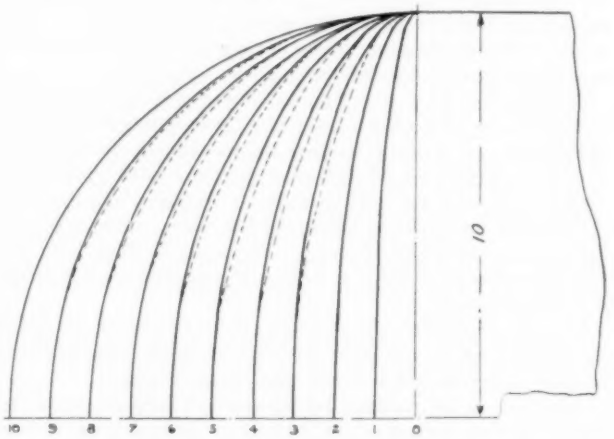


FIG. 5 LEAST-STRESS TWO-RADIUS HEADS COMPARED WITH ELLIPTICAL HEADS

(Broken lines, optimum two-radius heads; solid lines, elliptical heads.)

The head corresponding with the chart of Fig. 1 has a knuckle radius of 7 in. and a crown radius of 56 in. Fig. 3 shows what the bending stresses become when these radii are made 9.4 and 55, the meridian being now approximately an ellipse. The ratio (radius of head divided by its depth), or 2.1, remains the same. The stresses are reduced by about 33 per cent. Looking at the equation for S'' max in the knuckle (Table 1), it is seen that the stress varies directly as A^2K. As shown in Appendix 1, A^2K has a minimum value, with which the latter pair of radii above correspond. For the original head $A^2K = 56 \times 56(56/7 - 1) = 21{,}952$, while for the approximately elliptical head $A^2K = 55 \times 55(55/9.4 - 1) = 14{,}670$. This illustrates the change of stress with the change in knuckle radius. Fig. 4 shows bending stresses with 12.7 in. and 65 in. radii. They are somewhat more than for Fig. 3, as would be expected.

The greatest value of the combined bending-stress curves in Fig. 3 is 18,500 lb. per sq. in. At the inside surface there must

be added to this the meridian membrane stress (see S, Table 1) of

$$\frac{500 \times 36}{2 \times 2} = 4500 \text{ lb. per sq. in.}$$

giving 23,000 lb. per sq. in. in all.

The calculations and curves discussed in the foregoing appear to show good agreement with the results obtained from the 6-ft. by 40-ft. head by actual measurement. In practical design, however, the time involved in going through all this would be prohibitive, and a simpler method of analyzing head stresses must be sought.

If the expression for S'' max is taken from Table 1 and is applied to the foregoing case, there is obtained

$$\text{max bending stress} = \frac{0.147 \times 500 \times 55^2 \times 4.85}{36 \times 2} = 15{,}000 \text{ lb. per sq. in.}$$

and if there is added to this the 4500 lb. per sq. in. membrane stress, there is obtained at once a total of 19,500 lb. per sq. in. as against 23,000 obtained by more elaborate methods. This result is probably very much on the safe side. The crown-knuckle bending stress (curve S'', Figs. 1, 3, or 4) is derived using the hypothesis that there exists a sudden, sharp change of radius at the crown-knuckle edge. A very slight consideration of the manufacturing processes of drum heads will show

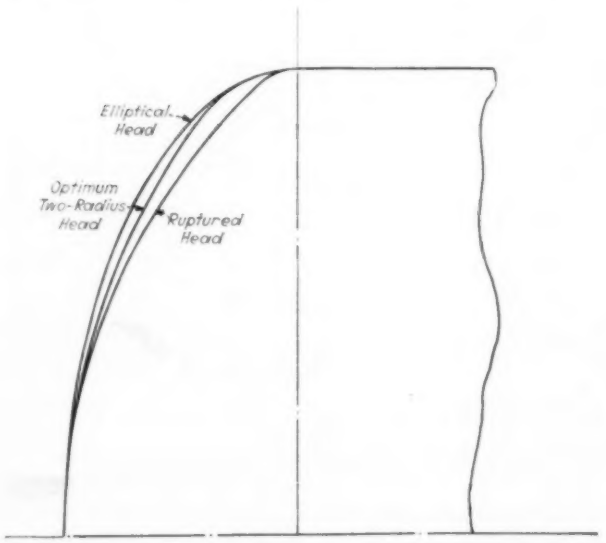


FIG. 6 FAILED HEAD OF A 6-FT. BY 40-FT. DRUM COMPARED WITH THE ELLIPSE AND THE OPTIMUM DOUBLE-RADIUS CURVE

that any such discontinuity as this is almost inconceivable. The knuckle and head almost certainly merge into each other much more gradually, with a consequent reduction in the bending stress. It also has to be remembered that the distortion of the head under pressure has the effect of still further reducing the stress.

From the foregoing considerations it is proposed that the equation

$$S'' \text{ max} = \frac{0.147\, PA^2K}{at}$$

be taken as giving the stress which governs the design of the head.

As an example, this will be applied to the 6-ft. by 40-ft. by 2-in. thick drum with the 400 lb. per sq. in. pressure, for which it was originally designed. (See Tables 1 and 2.)

For the original head, 7-in. knuckle radius, 56-in. crown radius, K would be $56/7 - 1 = 7$. Then the maximum bending stress in the head would be

$$S'' \text{ max} = \frac{0.147 \times 400 \times 3136 \times 7}{36 \times 2} = 18{,}000 \text{ lb. per sq. in.}$$

which is somewhat high. For 500 lb. this would be 22,500 lb. per sq. in. See (*) near S'' curve in Fig. 1 for comparison with

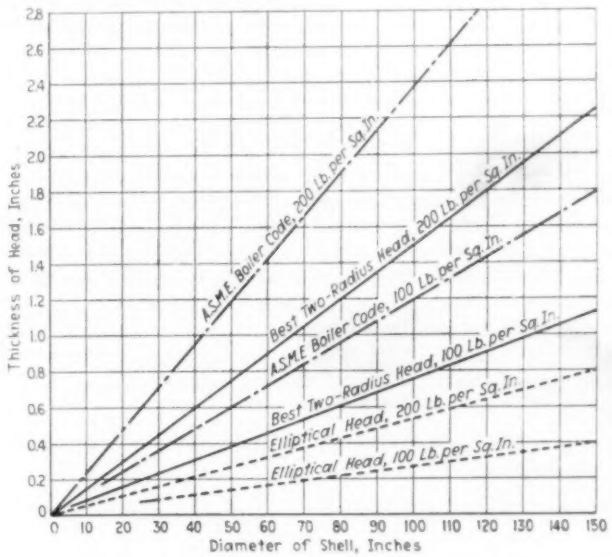


FIG. 7 THICKNESS OF HEAD VERSUS DIAMETER OF SHELL FOR ELLIPTICAL HEADS

(Allowable stress, 11,000 lb. per sq. in. All cases calculated by bending-stress formula.)

more elaborate method. For the 1600 lb. per sq. in. at which the head gave way, the stress would be 72,000 lb. per sq. in. As before pointed out, the stress-strain equations cease to apply accurately beyond the proportional limit, but such a result at least indicates a stress far exceeding this limit, with accompanying grave danger.

The other important stresses in the head are arrived at as follows:

There will be a bending hoop stress in the knuckle of

$$H'' \text{ max} = \frac{400 \times 3136 \times 7}{4 \times 36 \times 2} = 30{,}400 \text{ lb. per sq. in.}$$

and a membrane-hoop stress of

$$H = \frac{400 \times 56}{2 \times 2}(2 - 8) = -33{,}600 \text{ lb. per sq. in.}$$

These two stresses both occur at the knuckle-crown edge, and practically balance each other. Other stresses in the knuckle are relatively unimportant. (See Fig. 1.)

In the crown the maximum bending stress is given by

$$S'' \text{ max} = \frac{0.147 \times 400 \times 56 \times 7}{2} = 11{,}500 \text{ lb. per sq. in.}$$

There would be a bending hoop stress of

$$H'' \text{ max} = \frac{-400 \times 56 \times 7}{4 \times 2} = -19{,}600 \text{ lb. per sq. in.}$$

and a membrane hoop stress of

$$H = \frac{400 \times 56}{4} = 5600 \text{ lb. per sq. in.}$$

leaving a net hoop stress of —14,000 lb. per sq. in. which rapidly decreases as the boundary is left. This result is probably somewhat too high due to the non-existence of really sudden change of curvature.

In the shell, $k = 5.14$ (see Table 2) and the maximum bending stress is

$$S' \text{ max} = \frac{0.147 \times 400 \times 36 \times 5.14}{2} = 5430 \text{ lb. per sq. in.}$$

to which must be added the membrane axial stress of

$$S = \frac{400 \times 36}{2 \times 2} = 3600$$

giving a total meridian stress of 9030 lb. per sq. in. In this case the sudden change of curvature actually exists, as the shell and head are two separate pieces. The efficiency of the riveted joint between shell and head has also to be considered, which will make this stress somewhat more.

There is a bending hoop stress of

$$H' \text{ max} = \frac{-400 \times 36 \times 5.14}{4 \times 2} = -9250 \text{ lb. per sq. in.}$$

which rapidly dies away, as the boundary is left and a membrane hoop stress of

$$H = \frac{400 \times 36}{2} = 7200 \text{ lb. per sq. in.}$$

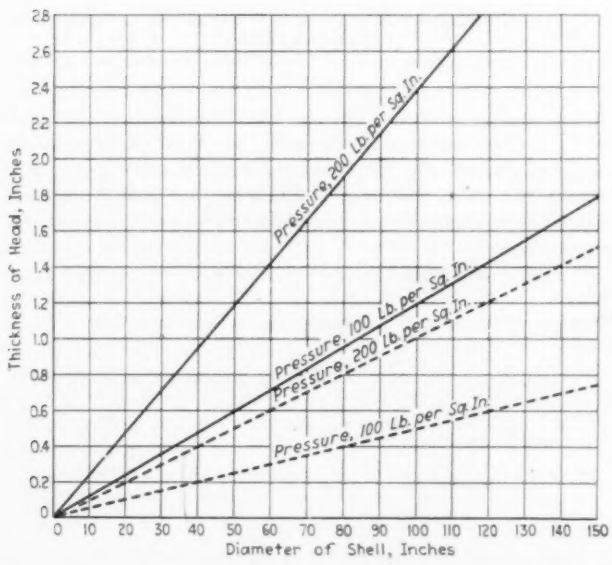


FIG. 8 THICKNESS OF A.S.M.E. HEADS VERSUS DIAMETER OF SHELL
(Knuckle radius = 6% of shell diameter. Solid line, thickness by bending-stress formula; broken line, thickness by A.S.M.E. rule.)

which is the same everywhere. The riveted joint efficiency also increases this.

From the foregoing, the 18,000 lb. per sq. in. bending stress in the knuckle is the stress which should govern design. For the head considered, it is considerably too high. If a head is taken of approximately elliptical form, the knuckle and crown radii would be 9.4 in. and 55 in., respectively (see Table 3), or $K = 55/9.4 - 1 = 4.85$. The maximum bending stress in the knuckle would be

$$S'' \text{ max} = \frac{0.147 \times 400 \times 3025 \times 4.85}{36 \times 2} = 11{,}900 \text{ lb. per sq. in.}$$

Other stresses would change correspondingly. The different stresses in the two cases are given, for purposes of comparison, as follows:

k	5.14	3.82
K	7	4.85
Knuckle radius, in.	7	9.4
Crown radius, in.	56	55
Shell radius, in.	36	36
Ratio radius to depth, in.	2	2
Maximum bending stress in knuckle, lb. per sq. in., S'' max.	18,000	11,900
Maximum bending stress in crown, lb. per sq. in., S'' max.	11,500	7,850
Maximum hoop stress in crown (net), lb. per sq. in., $H - H''$ max.	—14,000	—7,800
Maximum meridian stress in shell, lb. per sq. in., $S + S'$ max.	9,030	7,650
Maximum hoop stress in shell, lb. per sq. in., H	7,200	7,200

For the large knuckle radius, the bending stress in the knuckle still governs the design. If 70 per cent is taken for the efficiency

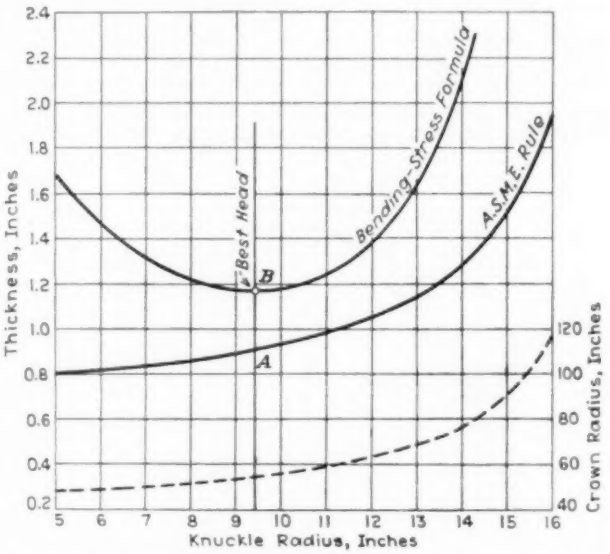


FIG. 9 KNUCKLE RADIUS VERSUS HEAD THICKNESS AND CROWN RADIUS
(Shell diameter, 72 in.; depth of head, 18 in.; pressure, 200 lb. per sq. in. Solid line, head thickness; broken line, crown radius.)

of the riveted joint, there is a hoop stress H in the shell of 10,300 lb. per sq. in. This may be further increased by the effect of manheads, showing that, taking

$$S'' \text{ max} = \frac{0.147\ PA^2K}{at}$$

as the governing stress, gives maximum stresses in the head and shell, which are in fair agreement. In the above, the stresses have been calculated from the given dimensions. If the maximum stress be decided upon first, the thickness can of course be obtained by simple transformation, or

$$t = \frac{0.147\ PA^2K}{a.S'' \text{ max}}$$

If the meridian were formed by one continuous curve, such as an ellipse, so that the radius of curvature varied gradually, the bending stresses due to the crown-knuckle joint would disappear altogether. It might be pointed out, however, that a perfect

ellipse is just about as unlikely as a sudden change of radius. The treatment adopted in this report is based on the assumption of conditions somewhere between these two improbabilities. The results show good agreement with the facts brought out in testing the 6-ft. by 40-ft. soaking drum to destruction, and the discrepancies are on the side of safety.

Profile of Least Stress

From the foregoing it will be seen that, for a given diameter and depth of head, more than one "two-radius" meridian curve can be used. There can be any number of such curves, and, as shown in Appendix No. 2, one of them will give the lowest value for the maximum bending stress. Table 3 has been prepared, from which the crown and knuckle radii for the best "two-radius" head can be selected. The formulas for the greatest bending stresses are also given. For the sake of comparison and completeness, the formula for an elliptical head is included. It

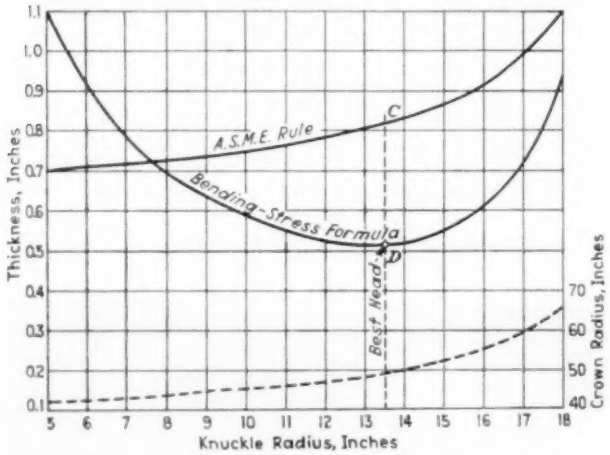


Fig. 10 Knuckle Radius Versus Head Thickness and Crown Radius

(Shell diameter, 72 in.; depth of head, 21 1/2 in.; pressure, 200 lb. per sq. in. Solid line, head thickness; broken line, crown radius.)

might be noted that for the two-radius head, the bending stress is set up by the crown-knuckle junction. For the elliptical head it is set up by the head-shell junction.

In Table 3 the shell radius is taken as 10 in., the internal pressure as 1 lb. per sq. in., and the plate thickness as 1 in. For other values the numbers in the table will change in proportion. Thus for a head where $u = 0.5$, $a = 36$ in., $P = 400$ lb. per sq. in., and $t = 2$ in., there would result

$$S'' \text{ max} = \frac{16.5 \times 3.6 \times 400}{2} = 11{,}900 \text{ lb. per sq. in.}$$

as in the previous example.

Columns 4 and 5 in Table 3 refer to stresses in double-radius heads. Note that in very deep heads, column 5 may give the governing stress. Column 6 is for elliptical heads, and the use of the table for these is self-evident from the foregoing.

Fig. 5 shows the optimum (or least-stress) two-radius heads compared with elliptical heads. As will be seen, the correspondence is close, being almost exact in the knuckle, where the important stresses come. As will be seen from the chart, when the crown is displaced radially under load, the head profile is brought still closer to the elliptical shape. It is of interest to note this coincidence, as it agrees with results of experiments made by Continental investigators.

Fig. 6 shows the head of a 6-ft. by 40-ft. drum, which failed under test, compared with the ellipse and the optimum double-radius curve. The very marked divergence of the ruptured head from the other two, especially at the knuckle, can be readily noted.

Procedure for Practical Design

For practical design the following procedure is tentatively proposed.

Assuming the depth of the head and the shell diameter to be known, the knuckle and crown radii for the optimum head can be selected from Table 3 from columns 2 and 3. For most heads the governing stress is found by means of the formula for S'' max in column 4. For very deep heads as stated in the explanation of Table 3, the governing stress may be that given for S' max in column 5. Column 6, in the table, brings out the great advantage of elliptical heads, except in cases where the hemispherical form is approached.

The internal working pressure being known, a thickness can now be assumed and the governing stress can be calculated, or the stress can be assumed and the thickness can be determined. The other stresses can now be computed, and it can be seen whether the principal stresses in the head and shell (usually the head bending stress and the shell membrane hoop stress) show no great discrepancy. In making this comparison, the increase in the shell hoop stress due to the riveted-joint efficiency should be allowed for and the effect of manheads considered. The

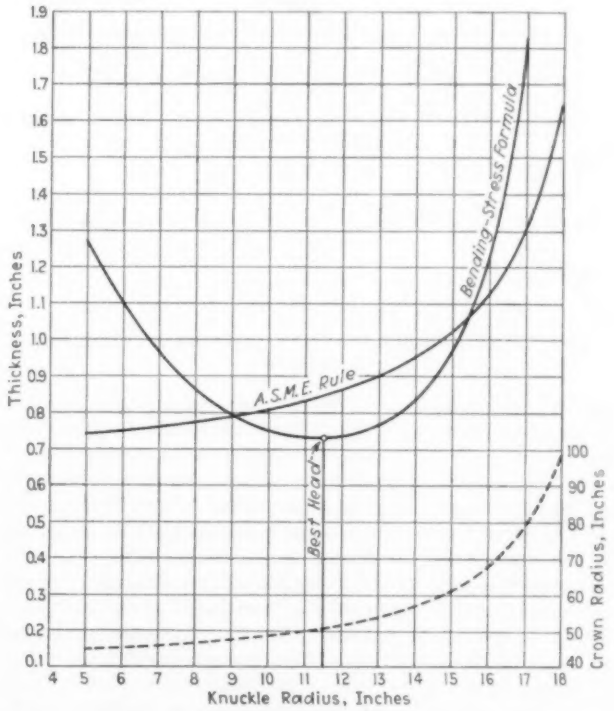


Fig. 11 Knuckle Radius Versus Head Thickness and Crown Radius

(Shell diameter, 72 in.; depth of head, 20 in.; pressure, 200 lb. per sq. in. Solid line, head thickness; broken line, crown radius.)

+ and — membrane and bending hoop stresses also can be checked against each other, the largest difference appearing to be in the crown at its junction with the knuckle. As shown in the foregoing, a radius-to-depth ratio of 2 to 1 gives good results, but special cases may arise in practice which are better suited by different proportions.

Comparison of Head Thicknesses by A.S.M.E. Rule and Bending-Stress Theory

In Fig. 7 are plotted thickness of head against diameter of shell for elliptical heads, "best" two-radius heads (see Table 3), and heads with a knuckle radius of 6 per cent of the shell diameter as permitted by the A.S.M.E. Boiler Code. The last named have an exceedingly small knuckle radius, which gives a meridian curve considerably different from an ellipse. It will be noticed

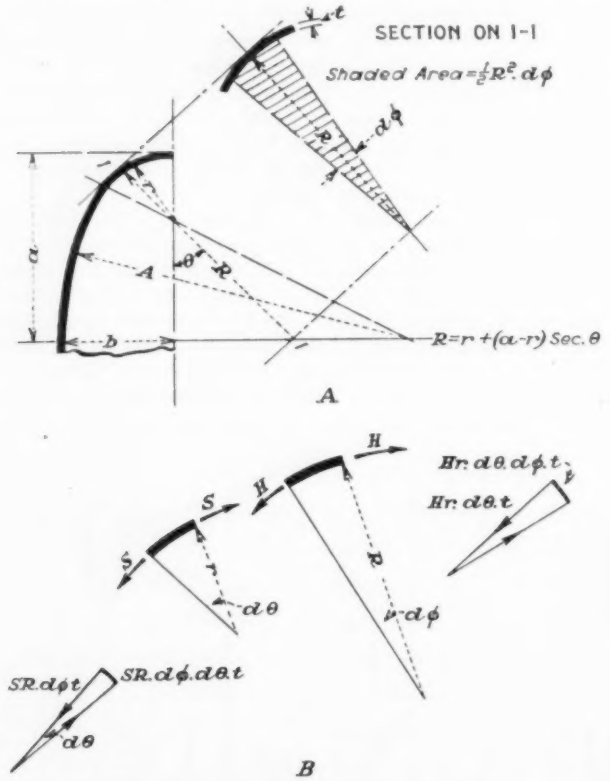


Fig. 12 Head Sections and Force Diagrams for Stress Calculation

that the thickness for the "best" two-radius heads (which is obtained by means of Table 3) lies between the A.S.M.E. head on the one hand and the exactly elliptical head on the other.

The thickness formulas for the three heads plotted reduce to A.S.M.E., 0.000119 *DP;* best two radius, 0.000075 *DP;* elliptical, 0.0000268 *DP*—where *D* is the shell diameter in inches and *P* the internal pressure in pounds per square inch. For other pressures than those given the thicknesses are obtained by simple proportion. For instance, take a "best head" of diameter 72 in. for 400 lb. per sq. in. pressure. For the 200 lb. per sq. in. line on the chart (see Fig. 7) 72 in. diameter corresponds with 1.08 in. thickness, so that 400 lb. per sq. in. would require 2.16 in. thickness. The chart is based on a fiber stress of 11,000 lb. per sq. in. Compare the 2.16 with 2 in. for 11,900 lb. per sq. in. given previously. Again, calculating from Table 3 there should be obtained

$$S'' \text{ max} = \frac{16.5 \times 3.6 \times 400}{2.16} = 11{,}000 \text{ lb. per sq. in.}$$

For any other stress than 11,000 lb. per sq. in., the thicknesses will be obtained by simple proportion.

Fig. 8 shows the thickness required for A.S.M.E. heads with a knuckle radius of 6 per cent of the shell diameter, as calculated by the meridian bending-stress formula and the A.S.M.E. rule. The great increase of thickness required to take care of the crown-knuckle bending stress will be noticed. The knuckle radius is so small that the bending stresses become very high, and these are not taken into account in the A.S.M.E. rule.

With a larger knuckle radius the bending stresses are reduced, and the discrepancy between the A.S.M.E. rule and the bending-stress formula tends to disappear. As an instance, take a head 72 in. in diameter and 18 in. deep. From Table 3 it is seen that $u = 0.5$, and the best shape would be given by a knuckle radius of $2.59 \times 3.6 = 9.35$ in. and a crown radius of $15.2 \times 3.6 = 55$ in. Let the internal pressure be 200 lb. per sq. in., and the tensile strength of the steel be 50,000 lb. per sq. in. Applying the A.S.M.E. rule, for the thickness there is

$$t = \frac{8.33 \times 200 \times 55}{2 \times 50{,}000} = 0.92 \text{ in.} \quad \text{[A]}$$

Calculating from Table 3 and using a factor of safety of 5 (or S'' max = 10,000) there would be

$$t = \frac{16.5 \times 200 \times 3.6}{10{,}000} = 1.19 \text{ in.} \quad \text{[B]}$$

If the depth of the head is increased to $21\frac{1}{2}$ in., or $u = 0.6$, the crown radius for the best shape becomes $13.6 \times 3.6 = 49$ in. The A.S.M.E. rule then gives

$$t = \frac{8.33 \times 200 \times 49}{2 \times 50{,}000} = 0.82 \text{ in.} \quad \text{[C]}$$

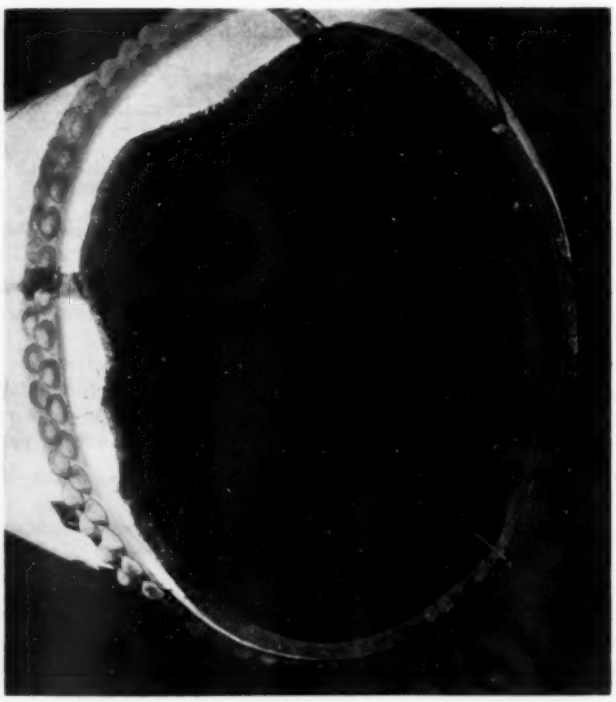


Fig. 13 Broken Head, Showing Line of Fracture Wholly Within the Knuckle

and, calculating from Table 3,

$$t = \frac{7.12 \times 200 \times 3.6}{10{,}0000} = 0.52 \text{ in.} \quad \text{[D]}$$

It thus is seen that, for a value of u (the depth of head/shell radius ratio) somewhere between 0.5 and 0.6, the A.S.M.E.

rule agrees with the bending-stress formula for the best-shaped two-radius head. A head of such proportions would be in accordance with good modern practice. It is interesting to note that the A.S.M.E. rule, the result of long and widespread practical experience, conforms with a bending-stress formula derived from purely mathematical considerations. It is further noteworthy that this conformity occurs only when rule and formula are applied to that head whose meridian contour is theoretically the best.

The foregoing calculations correspond with points *A*, *B*, *C*, and *D* on Figs. 9 and 10. These charts, together with the chart of Fig. 11, illustrate the effect of the small 6 per cent knuckle radius, as shown by the bending-stress formula. They also bring out clearly the existence of an optimum head, or one which requires the least thickness, either a small knuckle radius or a large crown radius accompanying high bending stresses. This can be seen by inspection of the formula for S'' max, reference being made to Table 2 for the full significance of the symbols. These charts also demonstrate how the A.S.M.E. rule and the bending-stress formula approach each other in the neighborhood of the "best head" for heads of about the proportions shown.

If it is desired to compare the A.S.M.E. rule for elliptical heads with the theory, one can proceed as follows: Take a head 72 in. in diameter, 18 in. deep, and 200 lb. per sq. in. pressure, as before. Let the safe working stress be taken as 10,000 lb. per sq. in.

The A.S.M.E. rule requires that, for such a head, the thickness shall be the same as that of the accompanying (seamless) shell. This is given by the formula

$$t = \frac{36 \times 200}{10{,}000} = 0.72 \text{ in.}$$

Applying Table 3, the required thickness would be

$$6 = \frac{5.88 \times 3.6 \times 200}{10{,}000} = 0.425$$

or, for the 0.72 in. thickness, an application of Table 3 would give a fiber stress of

$$S = \frac{5.88 \times 3.6 \times 200}{0.72} = 5860 \text{ lb. per sq. in.}$$

From the first two equations preceding, it is easy to see that the thickness would be 0.72 in each case, if the value 5.88 (see column 6 of Table 3) were changed to 10. This would correspond with a value of u (column 1 of Table 3) of slightly less than 0.4. It is thus seen that the A.S.M.E. rule would require a dish depth of about 25 per cent more than that called for by the theory, for the same thickness of metal.

From these considerations it would appear that the A.S.M.E. specifications for elliptical heads are well on the safe side.

It might be noticed that for any head diameter to dish-depth ratio there is one and only one ellipse, instead of an indefinite number, as in the case of the two-radius head. Such charts as those of Figs. 9, 10, and 11 are therefore impossible in this comparison.

Appendix No. 1

Derivation of Stresses in Heads and Shells of Pressure Vessels

CONSIDER a cylindrical drum with dished heads. Let a be the radius of the shell or cylindrical portion, r the knuckle radius, and A the crown radius; let t be the thickness and P the internal pressure; R is the radius of the curvature in a direction at right-angles to the knuckle curvature $1/r$.

For the shell, considered alone, the stresses are given by the well-known simple relations:

$$\text{Hoop stress} = H = \frac{aP}{t} \quad \ldots\ldots [1]$$

$$\text{Axial stress} = S = \frac{aP}{2t} \quad \ldots\ldots [2]$$

The strain displacements are radial only. If a point in the shell is moved from radius a to radius $a + w'$, the circumference will increase from $2\pi a$ to $2\pi(a + w')$, and the hoop strain will thus be

$$\frac{2\pi\ (a + w') - 2\pi a}{2\pi a} = \frac{w'}{a}$$

There can then be written, by the usual stress-strain equation:

$$\frac{w'}{a} = \frac{1}{E}(H - mS)$$

$$= \frac{aP}{2Et}(2 - m)$$

from [1] and [2], or

$$w' = \frac{a^2P}{2Et}(2 - m) \quad \ldots\ldots [3]$$

In the head, it is seen that the total pressure acting normally to a small sector of area $\frac{1}{2} R^2 \cdot d\phi$ (see Fig. 12-A, section 1-1) would be

$$\frac{1}{2}PR^2 \cdot d\phi \quad \ldots\ldots [4]$$

and this would be accompanied by a normal tension over an area of metal equal to

$$tR \cdot d\phi \quad \ldots\ldots [5]$$

so that the stress in the direction of a tangent to the head, in a plane passing through the axis, would be [4] divided by [5], or

$$S = \frac{PR}{2t} \quad \ldots\ldots [6]$$

Fig. 12-B represents an element of the knuckle. It has radii of curvature r and R for directions at right-angles. The edges are subjected to forces of $SRd\phi t$ and $Hr \cdot d\theta \cdot t$. Their radial components are $SRd\phi d\theta t$ and $Hr \cdot d\theta d\phi t$. They are balanced by the total pressure over the area or $PR \cdot d\phi \cdot r \cdot d\theta$. Putting the pressure equal to the sum of the two radial components, and transposing, gives

$$Hrt = PRr - S \cdot Rt$$

$$H = PR/t - SR/r$$

Substituting from [6], gives

$$H = S(2 - R/r) \quad \ldots\ldots [7]$$

By the same procedure as for the shell, the radial displacement is

$$w' = \frac{PR^2}{2Et}(2 - R/r - m) \quad \ldots\ldots [8]$$

In the crown, R and r are each equal to A, and the radial displacement is

$$w' = \frac{PA^2}{2Et}(1 - m) \quad [9]$$

These displacements are for the shell, knuckle, and crown acting alone. From [3] and [9] it is seen that the displacements are the same, everywhere in the shell and everywhere in the crown. Equation [8] shows that displacement varies in the knuckle, since R changes gradually from A at the junction of the crown and knuckle to a at the junction of the shell and knuckle. There then are, for these edges, proportional radial displacements as follows:

Crown	$(1 - m)$
Knuckle	$(2 - A/r - m)$
Shell	$(2 - m)$
Knuckle	$(2 - a/r - m)$

The radial displacement is positive outward. It is thus seen that the crown and shell expand. Since, for all reasonable proportions, a/r and A/r are numerically more than 2, the knuckle, if free to do so, would contract. This agrees with [7], which shows that the hoop stress would be compressive if R/r exceeded 2. As the crown, knuckle, and shell are not free of each other, it is easy to see from the foregoing that the knuckle will tend to be bent outward at the edges, like a trumpet, while the shell and crown will tend to be bent in. The bending moments will be zero at the edges, since they change sign there. This can also be seen from the fact that the meridian curve will have points of inflection at the junctions of the crown, knuckle, and shell, and the bending moment will there be zero, as in a beam fixed at the ends.

Accompanying bending will be a hoop stress, quite apart from that already derived for simple pressure. The trumpet shape assumed by the knuckle sets up hoop tension. The corresponding inward bending of the shell and crown bring about hoop compression.

There will also be the usual shear stresses, which occur with bending moment.

Let s be the distance measured along the meridian of the knuckle from the shell-knuckle edge. There would then result by the fundamental equation of bending moment (see "Calculus for Engineers," John Perry, p. 108, or other textbook)

$$\frac{d^2M}{ds^2} = F \quad [10]$$

where M = bending moment and F = the force, acting normally to the knuckle surface, per unit of length s. This corresponds with the distributed load on an ordinary beam; i.e., so many pounds per inch.

In a thin cylinder or shell, the bending hoop stress can be taken as constant over the thickness t. Let the bending hoop force per unit of length s be called h, so that h/t = bending hoop stress. The radial component of h is obtained by the method described in the paragraph following Equation [6], and is h/R per unit of length s for a strip of unit width. It is easy to see that this is numerically equal, though of opposite sign, to the force (or distributed load) per unit of length accompanying bending. There is thus obtained from [10]

$$\frac{d^2M}{ds^2} = \frac{-h}{R} \quad [11]$$

Since the hoop strain is w/R, the usual stress-strain relation gives

$$\frac{w}{R} = \frac{1}{E}\left(\frac{h}{t} - S'm\right)$$

where S' is the stress in the s direction and m is Poisson's ratio. Since t is relatively small, and the + bending stresses are equal to the — bending stresses, there can be used the average value of S' or O. Then

$$\frac{w}{R} = \frac{h}{Et} \quad [12]$$

or

$$\frac{h}{R} = \frac{Etw}{R^2}$$

and there is obtained from [11]

$$\frac{d^2M}{ds^2} = \frac{-Etw}{R^2} \quad [13]$$

To get M, the bending moment, take the well-known equation

$$\frac{d^2w}{ds^2} = \frac{M}{EI} \quad [14]$$

A little consideration will show that the narrow meridional strips, considered as beams, cannot deform laterally, as does an ordinary beam. The stress-strain equations would then be of the form

$$\text{Strain in the } s \text{ direction} = \frac{1}{E}\left(S' - \frac{hm}{t}\right)$$

$$\text{Strain in the } h \text{ direction} = \frac{1}{E}\left(h - \frac{mS'}{t}\right) = 0$$

or it can be stated that the net lateral strain for a thin strip is taken as 0, the + strains for one half thickness canceling the — strains for the other half thickness. In any case, from the foregoing equations

$$\text{Strain in the } s \text{ direction} = \frac{S'}{E}(1 - m^2)$$

or E is replaced by $\dfrac{E}{1 - m^2}$ in calculating the bending moment.

I = the moment of inertia of a strip section of unit width = $t^3/12$ (unit width is taken for convenience, since the width cancels out in the final result), and from [14]

$$M = \frac{Et^3}{12(1 - m^2)}\frac{d^2w}{ds^2} \quad [15]$$

Differentiating twice

$$\frac{d^2M}{ds^2} = \frac{Et^3}{12(1 - m^2)} \cdot \frac{d^4w}{ds^4}$$

Substituting this in [13] and transposing, gives the differential equation

$$\frac{d^4w}{ds^4} + \frac{12(1 - m^2)}{R^2t^2}w = 0$$

If there is taken

$$4n^4 = \frac{12(1 - m^2)}{R^2t^2} \quad [16]$$

or

$$n = \sqrt[4]{\frac{3(1 - m^2)}{R^2t^2}}$$

the differential equation becomes

$$\frac{d^4w}{ds^4} + 4n^4w = 0$$

Symbolically $(D^4 + 4n^4)w = 0$

The four roots of this are $n(1 + i)$, $-n(1 + i)$, $n(1 - i)$ and $-n(1 - i)$, which give the solution

$w = b_1e^{-n(1+i)s} + b_2e^{-n(1+i)s} + b_3e^{n(1+i)s} + b_4e^{n(1-i)s} = e^{-ns}(a_1 \cos ns + a_2 \sin ns) + e^{ns}(a_3 \cos ns + a_4 \sin ns)$

Experiment shows that the bending moment dies out as s gets large. This can be possible only if a_3 and a_4 vanish.

By differentiation

$$\frac{dw}{ds} = ne^{-ns}\left[(a_2 - a_1)\cos ns - (a_2 + a_1)\sin ns\right]$$

$$\frac{d^2w}{ds^2} = 2n^2e^{-ns}\left[a_1 \sin ns - a_2 \cos ns\right]$$

At the edge, where $s = 0$, there is a point of inflection (see second paragraph following Equation [9]) or $\frac{d^2w}{ds^2} = 0$ there. The last equation then shows that $a_2 = 0$, or

$$\frac{d^2w}{ds^2} = 2a_1n^2e^{-ns}\sin ns \quad \ldots\ldots\ldots\ldots [17]$$

and

$$w = a_1e^{-ns}\cos ns$$
$$= a_1 \text{ when } s = 0 \quad \ldots\ldots\ldots\ldots [18]$$

Note that n is a function of R, so that this solution applies for a small arc, over which R may be regarded as constant. At the edge $R = a$. Anywhere in the shell $R = a$, corresponding with $n = n^\circ$, and the equation

$$w = A'e^{-n^\circ s}\cos n^\circ s \quad \ldots\ldots\ldots\ldots [19]$$

applies anywhere in the shell, since a and n° do not change with s. For the shell w is negative. The shearing force is given by the well-known relation

$$\text{Shear} = \frac{dM}{ds}$$
$$= \frac{Et^3}{12(1 - m^2)} \cdot \frac{d^3w}{ds^3}$$
$$= \frac{Et^3a_1n^{\circ 3}}{6(1 - m^2)} \cdot e^{-ns}(\cos n^\circ s - \sin n^\circ s) \text{ from [17]}$$
$$= \frac{Et^3a_1n^{\circ 3}}{6(1 - m^2)} \text{ when } s = 0 \quad \ldots\ldots\ldots\ldots [20]$$

This is for the knuckle at the edge. Using [19] there is found for the shell at the edge

$$\text{Shear} = -\frac{Et^3A'n^{\circ 2}}{6(1 - m^2)} \quad \ldots\ldots\ldots\ldots [21]$$

and since [17] and [18] both represent the shear at the edge, $-A' = a_1$.

Now collect from [3], [8], [18], and [19], and obtain

$$\left.\begin{array}{l}\text{Total radial deflection of shell at the edge } \dfrac{a^2P}{2Et}(2 - m) - a_1 \\ \text{Total radial deflection of knuckle at the edge } (R = a),\ \dfrac{a^2P}{2Et}(2 - a/r - m) + a_1\end{array}\right\} \quad \ldots\ldots [22]$$

These deflections must be equal, since the shell and knuckle do not separate. There thus is given $a_1 = a^2Pk/4Et$ if $a/r = k$.

For the knuckle, the bending deflection due to the shell-knuckle edge is, anywhere near the edge,

$$w = \frac{a^2Pk}{4Et}e^{-n^\circ s} \cdot \cos n^\circ s \quad \ldots\ldots\ldots\ldots [23]$$

and

$$\frac{d^2w}{ds^2} = \frac{a^2Pkn^{\circ 2}}{2Et}e^{-n^\circ s} \cdot \sin n^\circ s$$

The section modulus is $t^2/6$, so that the bending stress becomes, using [15]

$$S' = \frac{3a^2Pk\ n^{\circ 2}}{12(1 - m^2)}e^{-n^\circ s}\sin n^\circ s \quad \ldots\ldots\ldots\ldots [24]$$

or since

$$\frac{3(1 - m^2)}{a^2t^2} = n^\circ 4$$

$$S' = \frac{3Pk}{4n^{\circ 2}t^2}e^{-n^\circ s}\sin n^\circ s \quad \ldots\ldots\ldots\ldots [25]$$

for the head.

As already stated, [21] only applies near the edge for the head. To obtain an equation which applies for any distance s from the edge, the procedure is as follows: n is now a function of R. One can then write [24], for points close to the edge, as

$$S' = \frac{3a^2Pkn^2}{12(1 - m^2)} \cdot e^{-ns}\sin ns \quad \ldots\ldots\ldots\ldots [26]$$

which is correct when s is small and n differs very little from n°. When s is not small, one must replace ns by $\int n.ds$. There is also $\sqrt{3(1 - m^2)} = n^2Rt$, so that [26] can be written

$$S' = \frac{3a^2Pk}{4Rt\sqrt{3(1 - m^2)}}e^{-\int n.ds}\sin\left(\int n.ds\right) \quad \ldots\ldots [27]$$

Bending Hoop Stress. This is h/t, and it is seen from [12] that this is Ew/R, or Ew/a at the edge.

Substituting from [23] the hoop stress can be given as

$$H' = \frac{aPk}{4t}e^{-n^\circ s}\cos ns \quad \ldots\ldots\ldots\ldots [28]$$

close to the edge. Remote from the edge, Ew/R must be substituted in [23] and ns be replaced by $\int n.ds$. This gives

$$H' = \frac{a^2Pk}{4t\ R}\left(e^{-\int n.ds}\right)\cos\left(\int n.ds\right) \quad \ldots\ldots\ldots [29]$$

For the bending stresses set up by the knuckle-crown edge, Equation [22] becomes, using [8], [9], and [18] with $R = A$ instead of $R = a$:

Total radial deflection of crown at the edge,

$$\frac{PA^2}{2Et}(1 - m) - a_1$$

Total deflection of knuckle at the edge,

$$\frac{PA^2}{2Et}(2 - A/r - m) + a_1$$

Since these are equal, there is obtained

$$a_1 = \frac{PA^2}{4Et}(A/r - 1)$$

$$= \frac{PA^2K}{4Et} \qquad \text{if } A/r - 1 = K$$

There then are, for the stresses, remembering that s is now measured from the crown-shell edge, equations of the same form as [25], [27], [28], and [29], A being substituted for a and K for k.

The stress formulas derived in the foregoing are collected and arranged in Table 1.

Maximum Stresses

As will be seen from Figs. 1, 3, and 4, the meridional bending stress (due to the crown-knuckle edge) in the knuckle, is the most serious stress. This is S'' for the knuckle in Table 1. Considering the part of the formula $e^{-X} \sin X$, it is seen that this has a maximum value when

$$\frac{d}{dx}(e^{-X} \sin X) = 0$$

$$\cos X = \sin X$$

$$X = \frac{\pi}{4}$$

and the maximum value is

$$e^{-\frac{\pi}{4}} \sin \frac{\pi}{4} = 0.324$$

As this occurs near the shell-knuckle edge, one would be a little on the safe side in taking $R = a$, or

$$S'' \max = \frac{0.147\ PA^2K}{at} \qquad [30]$$

where $m = 0.3$.

By the same method the maximum values of S' for the shell and knuckle and S'' for the crown can be obtained.

The greatest value of the function $e^{-X} \cos X$ occurs when $X = O$, when the function equals unity. Consequently

$$H'' \max = \frac{PA^2K}{4at} \qquad [31]$$

with similar expressions for H'.

Membrane Hoop Stress. The greatest value of this occurs at the crown knuckle edge, and is

$$H = S(2 - A/r) \qquad [32]$$

Membrane Meridional Stress. The greatest value occurs when

$$S = \frac{PA}{2t} \qquad [33]$$

The foregoing maximum stresses are included in Table 1.

By taking the shell radius a as the value of R for the stress S' max in the head, the answer is a little too high. However, a stress of this amount must occur in the shell, since a is constant all over it.

The use of S'' max as the governing stress has already been discussed in the body of the paper.

Appendix No. 2

Best Shape of Meridian Curve Made Up of Two Arcs of Circles

THE maximum stress in any head whose meridian curve is composed of two arcs is

$$S'' \max = \frac{0.147\ PA^2K}{at}$$

and for any given (head radius) : (depth) ratio, or a/b, there can be an infinite number of such meridian curves.

If $r = 0$, there is obtained $A^2K = A^2(A/0 - 1) = \infty$

If $r = b$, A = infinity, there is obtained $A^2\left(\frac{\infty}{b} - 1\right) = \infty$

from which it would appear that somewhere between these two extremes there are values of r and A which make A^2K a minimum. This would give a head for which S'' max would have the least value.

Between A, r, q, and b, there is the relation (see Fig. 12-A)

$$(A - b)^2 + (a - r)^2 = (A - r)^2$$

from which

$$A = \frac{a^2 + b^2 - 2ar}{2(b - r)}$$

and

$$A^2K = \left[\frac{(a^2 + b^2 - 2ar)}{2(b - r)}\right]^2 \frac{(a^2 + b^2 - 2ar - 1)}{2(b - r)r}$$

Differentiating with regard to r, simplifying, and equating to zero, gives

$$2r(b - r)^2 + (a - r)^2(a - b)^2 - (a^2 + b^2 - 2ar)2ar^2 - (a^2 + b^2)(2r - b) = 0$$

Putting $b = ua$, this becomes

$$r^3 - \frac{a(5 - 2u + u^2 + 2u^3)}{2[1 + (1 - u)^2]} r^2 + \frac{a^2(2 - u + 2u^2)(1 + u^2)r}{2[1 + (1 - u)^2]} - \frac{a^2(1 + u^2)^2u}{4[1 + (1 - u)^2]} = 0$$

which is of the form

$$r^3 + C^\circ r^2 + C'r + C'' = 0$$

Put $r = X - C^\circ/3$. Then by the usual procedure for solving cubic equations, there are obtained

$$X = \sqrt[3]{g + \sqrt{g^2 - P^3}} + \sqrt[3]{g - \sqrt{g^2 - P^3}}$$

$$P = \frac{(C^\circ)^2}{3} - \frac{C'}{3}$$

$$g = \frac{-(C^\circ)^3}{3} + \frac{1}{2}C'\frac{(C^\circ)}{3} - \frac{C''}{2}$$

$$C^\circ = \frac{-a(5 - 2u + u^2 + 2u^3)}{2[1 + (1 - u)^2]}$$

$$C' = \frac{a^2(2 - u + 2u^2)(1 + u^2)}{2[1 + (1 - u)^2]}$$

$$C'' = \frac{-a^2(1 + u^2)^2u}{4[1 + (1 - u)^2]}$$

From these equations the knuckle and crown radii in Table 3 have been calculated.

Elliptical Heads. The meridian curve is continuous, so that only the shell-knuckle junction (stress S') has to be considered. S'' will not exist.

The radius of curvature of the ellipse at the junction is well

known to be $a/\underline{K}^2$ if $\underline{K} = \frac{a}{b}$. This is the r of Equation [22] which will read:

$$\text{Total deflection of shell at edge} = \frac{a^2P}{2Et}(2 - m) - a_1$$

$$\text{Total deflection of head at edge} = \frac{a^2P}{2Et}(2 - \underline{K}^2 - m) + a_1$$

From which

$$a_1 = \frac{a^2P\underline{K}^2}{4Et}$$

so that the equations in Table 1 hold for elliptical heads, $\underline{K}^2$ being substituted for k. As before pointed out, the expressions in K do not apply to elliptical heads.

Discussion

W. M. Coates.[3] The results which the author has obtained are rather surprising. The exception is that the method which he has used would show the discontinuity effects introduced by joining knuckle and crown to be of less consequence than those by joining cylinder and knuckle, especially if the head is of favorable form. The results of experiments also point in this direction, but inasmuch as the demonstration here is a purely theoretical one, it will be criticized on that basis. The explanation of the discrepancy between results and expectations lies in an incorrect application of the method.

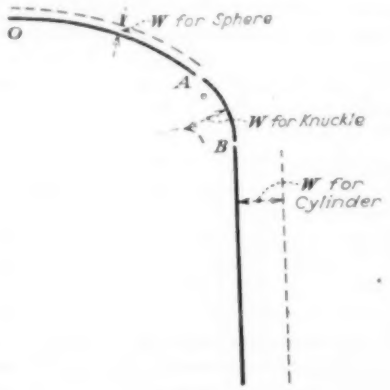


Fig. 14

To show this, consider the accompanying Fig. 14. The heavy lines show the meridional sections of the vessel, and are broken where geometrical changes occur in the nature of the shells. The dotted lines indicate to an exaggerated scale the normal displacements w in each part,[4] the magnitudes being laid off along perpendiculars to the contour. In the figure, the parts are considered as membranes under uniform inner pressure, stretched at their edges by the tensile forces necessary to maintain equilibrium. Then the first state of stress to be considered is the state in the membrane, given by the usual formulas for the case of cylinder and spherical cap. It is clear from the figure that in addition to this, a consideration must be made of the state of stress caused by the mutual restraints at points A and B which prevent the occurrence of the indicated discontinuities. This second group of effects is the one referred to as "discontinuity effects." There are very significant errors in the computation of both sets of effects in this paper.

Turning to Appendix No. 1, Equations [1] to [3] and [6] and [7] represent the well-known membrane formulas for the stresses and displacements, [1], [2], and [3] for the cylinder, [6] and [7] for the general curved shell. The last two hold for both crown and knuckle, the latter being the zone of a torus, starting from the equator.

Equation [8] represents the membrane normal displacement of the knuckle surface, but is true only for points on the equatorial circle. The correct value for points off this edge circle may be derived by the general method[5] and is very different from that given. To illustrate, the values of w at the top of the knuckle, point A in Fig. 14, are:

From data at bottom of third page	by Equation [8]	$-\frac{p}{Et}$ (9880)	error is approx. 70 per cent
	correct value	$-\frac{p}{Et}$ (5790)	
From data at bottom of fourth page	by Equation [8]	$-\frac{p}{Et}$ (11,970)	error is approx. 60 per cent
	correct value	$-\frac{p}{Et}$ (7450)	

The error in this expression is of fundamental importance, since w is used to determine the constants which enter the final expressions for the discontinuity stresses; see the work at the bottom of the eleventh page and the top of the twelfth page. It is much too high, as are also these stresses. Equation [30] is consequently incorrect, and the discussion which is based upon it in Appendix No. 2 and in the body of the paper loses thereby its significance.

It is to be regretted that the author has done so much work without proper study of the literature. This general method of attack will work if correctly applied; however, it should not be assumed that either its derivation or its application is new. Geckeler, for example, has given both in "Über die Festigkeit achsensymmetrischer Schalen," Forschungsarbeiten a. d. Gebiete des Ingw., V.D.I., No. 276. He uses a slightly different method of applying the boundary conditions at the junction of crown and knuckle.

In applying the method used in this paper, it should be kept in mind that the fundamental differential equation which is to give the discontinuity effects,

$$\frac{d^4w}{ds^4} + 4n^4w = 0$$

is strictly true only for the thin cylinder. However, it is approximately true also for the equatorial zone of the knuckle, if the thickness is not too great and the radius not too small. In going to the top of the knuckle, n is variable and the equations take the form of [27] and [29]. Although the author states these equations, they are not used, and additional errors are the consequence.

In comparing heads in the group of "favorable shapes," it must be shown that the small quantities neglected in the foregoing simplified form of the differential equation are of higher order than the differences to be expected between the results for

[3] Professor of Engineering Mechanics, University of Michigan, Ann Arbor, Mich.

[4] For the sake of the figure the values shown for w in the knuckle are made smaller than the actual ones, in comparison with those shown for crown and cylinder.

[5] See, for example, the reference quoted at the beginning of the paper, Höhn-Huggenberger, "Über die Festigkeit der gewölbten Böden," etc., part II, Equations [69a] and [69b], and note the misprint in the latter. This method applies also to the case of the cylinder, but is unnecessarily complicated there, and is generally replaced by that given in the paper, between Equations [2] and [3].

heads which are themselves only slightly different; otherwise the conclusions based upon the computation may give false testimony as to the actual state of affairs. Some discussion on this question has been given elsewhere.[6] The tangential displacement at the top of the knuckle, point A, should be investigated in connection with the determination of the constants of integration; also, the discontinuity hoop bending stress should be included among the stresses.

There are other points in the paper with which it is not possible to agree, but in view of the seriousness of those mentioned, it is not thought worth while to list more.

E. O. Waters.[7] Engineering practice with regard to pressure-vessel heads has passed through an interesting process of development. Starting with the flat head, it was realized at an early stage that a head of this form has no membrane strength whatever, and it has been abandoned for pressure-vessel use, excepting those cases, such as tanks for liquid storage, where the pressure is low and external reinforcement can be provided. Next came the spherical sector head with sharp knuckle; this had a membrane strength which was easily calculable, but it was obvious that it set up very severe stresses in the shell even if the junction between head and shell were assumed to be flexible; and, if the joint were not flexible, severe bending stresses were encountered in both shell and head in the vicinity of the junction. The first of these disadvantages led to the adoption of hemispherical heads, as in water tanks and steam accumulators, but that did not remove the second source of weakness, i.e., the bending stresses.

This condition was mitigated by the next step in development, the use of a knuckle having a small radius of arbitrary amount, together with a spherical crown, and more recently, the evolution of this type into (*a*) the spherical crown with large radius knuckle and (*b*) the full semi-elliptical head. Having reached this point, there may be some who have been misled into a belief that now we have only to design for the membrane stresses and the bending stresses will take care of themselves. To disprove this, one has only to study this very admirable paper, which shows that, as far as double-curvature heads are concerned, the critical stress exists near the shell-knuckle junction rather than in the crown where the membrane stress is the highest. This paper and a recent one by W. M. Coates, "The State of Stress in Full Heads of Pressure Vessels," (Trans. A.S.M.E., APM-52-12) are most timely in that they bring emphatically to the attention of American engineers a condition which, until recently, has received mention only in foreign literature.

Mr. Coates confined his analysis to elliptical heads, and in consequence his method of computing the boundary stresses is quite laborious; the present author has gotten around this by assuming the head to have constant radii of curvature, between limits. This raises the question as to whether the bending stresses at the edge of an elliptical head could not be computed by the method which the author gives for double-radius heads. Assuming that the radii of the ellipse do not vary greatly over the region in which the bending stresses exist, and assigning to them an average value in Mr. Rhys' formulas, results would be obtained with, perhaps, a sufficiently high degree of accuracy, and certainly with much less labor.

Both Mr. Coates and Mr. Rhys have emphasized the bending stresses near the junction of the head and shell. It is the writer's belief, however, that where the surface has a continuously varying curvature, these stresses may continue right up to the center of the crown and have a magnitude that is far from negligible. A series of strain-gage measurements on elliptical heads which he has observed during the past year has indicated that the apparent stress at the center of the crown is 5 to 10 per cent higher than the values obtained by the membrane formulas, and he is at a loss to explain them in any other way. The following analysis will throw further light on this question.

The meridional bending moment on an element of the head is (see W. M. Coates, loc. cit.):

$$M_1 = -\frac{Et^3}{12(1-m^2)}\left[\frac{d^2w}{ds^2} + \frac{m\tan\beta}{R_2}\frac{dw}{ds}\right]$$

where

E = Young's modulus
t = thickness of head
m = Poisson's ratio
w = radial displacement of the element
s = distance from the extremity of the major axis to the element, measured along the meridian
β = angle between the axis of the pressure vessel and a tangent to the meridian at the element
R_2 = radius of curvature of a section perpendicular to the meridian, at the element

The foregoing equation assumes that the displacement of the element under stress is purely radial. Precise calculation shows that this is sensibly true near the center of the crown; it is of course exactly true at the center. E.g., compare the values of u and w in columns 9 and 10, Table 1, of Coates' paper, for $\beta > 70$ deg. Then, from the relations (see W. M. Coates, loc. cit.):

$$w = \frac{R_2}{E}(S_2 - mS_1)$$
$$S_1 = pkav/2t$$
$$S_2 = S_1\left(2 - \frac{1}{v^2}\right)$$
$$R_2 = kav$$
$$\frac{ds}{d\beta} = R_1 = kav^3$$

where

S_2 = hoop stress on the element
S_1 = meridional stress on the element
p = fluid pressure per unit area
k = major axis/minor axis
a = semi-major axis
$v = 1/\sqrt{1 + (k^2 - 1)\cos^2\beta}$
R_1 = meridional radius of curvature

it may be shown that

$$M_1 = -\frac{(2-m)(k^2-1)}{24(1-m^2)}pt^2\left[\frac{k^2-1}{2}\cos^2 2\beta + (k^2+1-m)\cos 2\beta + \frac{k^2-1+2m}{2}\right]$$

and the bending stress at the outside

$$S_b = \frac{6M_1}{t^2} = -\frac{(2-m)(k^2-1)}{4(1-m^2)}p\left[\frac{k^2-1}{2}\cos^2 2\beta + (k^2+1-m)\cos 2\beta + \frac{k^2-1+2m}{2}\right]$$

When $m = 0.3$, $k = 2$, and $\beta = 90$ deg. (center of crown), $S_b = 1.96\ p$. Moreover, $v = 1$, and hence the membrane

[6] See the closure to the discussion of the paper, Coates, "The State of Stress in Full Heads of Pressure Vessels." Trans. A.S.M.E., APM-52-12, p. 130.

[7] Associate Professor of Mechanical Engineering, Yale University, New Haven, Conn. Assoc-Mem. A.S.M.E.

stress $S_1 = S_2 = pa/t$; accordingly, the bending stress is 196 t/a per cent of the membrane stress. For low pressures, where t/a is small, this may be an insignificant percentage; but for high pressure drums, where t/a is of the order of 1/20 to 1/10 or more, the bending stress appears to be far from negligible.

T. McLean Jasper.[8] The tests presented in this paper will add considerably to the knowledge of vessel phenomena dealing with riveted construction. The results indicate something of the real factors of safety which have prevailed in the past for such construction and will help to clear up the misconceptions which seem to prevail in the minds of a great number of engineers. It was the writer's privilege to obtain some of the original test results and to see some of the test work in progress. It is believed that the test specimens were prepared with great care and that the results were obtained with painstaking skill. It therefore should represent a very good picture of a type of construction which has given fine service over a great number of years.

May it be added that the stationary steam service, which has used this construction for a great number of years, has consistently designed on the basis of a factor of safety of 5. These tests therefore indicate very clearly the real factors of safety which have prevailed so far as heads are concerned, and are therefore exceedingly valuable contributions to the knowledge of vessels.

The thing which is very clearly brought out by this paper is that a dished head 72 in. in diameter, 2 in. thick, and with a knuckle radius of 7 in. failed at a hydraulic pressure of 1600 lb. per sq. in. With a uniform shell section of this thickness and diameter, the calculated stress at the failure of the head would be around (1600 × 72)/4 or 28,800 lb. per sq. in. The steel used is given a minimum specification strength value of 50,000 lb. per sq. in. The head undoubtedly failed when the stress reached the ultimate strength. The best that the writer can see to this head is that its value was not above 28,800/50,000, or 0.576 of what it should be for the cylinder providing a joint of 100 per cent could be made. It is presumed that the metal used had a strength somewhat above the minimum specification.

The writer ventures to state that today vessels can be constructed with joints as strong as the plate without increasing the thickness of the vessel section at the joint.

A second head with a knuckle radius of 6 in. and a thickness of 1.62 in. failed at a pressure of 1200 lb. per sq. in., or a calculated stress of 26,600 in the cylinder wall of the same thickness. This gives an efficiency for this head of about 53.2 per cent.

The third head, with a knuckle radius of 10 in., showed a value of 42,300/50,000, or 84.5 per cent. Undoubtedly this last head is as good as a riveted vessel with an 85 per cent joint efficiency needs. However, with no more metal, or at least very little more, an elliptical head can be constructed which will fail a vessel in the cylinder away from a properly constructed joint if properly attached.

May it be said that an elliptical head of 2 to 1 ratio when used with a riveted joint is so restrained at the joint by a double thickness of metal that it cannot develop the strength that it would develop with a vessel of uniform thickness at the joint.

The writer is sure that the author is in error when he states that the A.S.M.E. is well on the safe side with the elliptical heads recommended. They are just right in their recommendations for elliptical heads.

He wishes also to correct the impression that there is only one elliptical head design. Elliptical heads of 2 to 1 ratio will fail a cylinder wall of the same thickness if properly made and attached. An elliptical head of a ratio less than 2 to 1 will be even stronger, and such heads have been made on occasion for special reasons such as to avoid excessive reinforcement when openings are introduced into them, etc.

[8] Director of Research, A. O. Smith Corporation, Milwaukee, Wis. Mem. A.S.M.E.

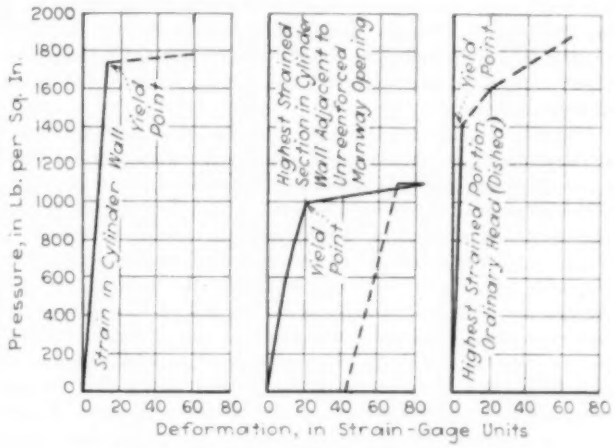


Fig. 15 Graph Showing Strain Values

(Given, under the same pressure conditions, in the main cylinder, at the most stressed situation around manway openings, and in the most stressed situation in a dished head. The yield point indicates when the metal has reached the same stress at each position where the strain measurements were made.)

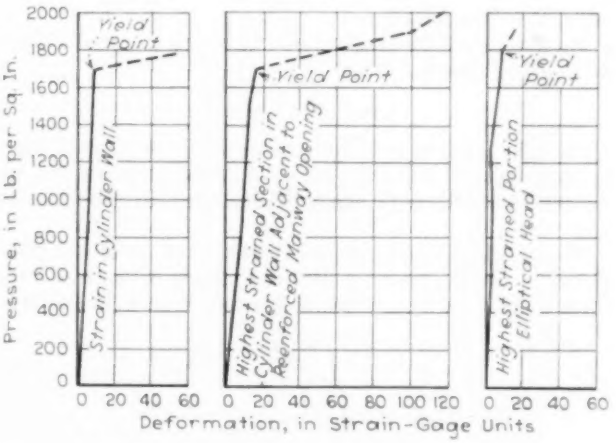


Fig. 16 Graph Showing Strain Conditions

(The appropriate conditions are shown when manhole openings are reenforced and when an elliptical head is used in place of the ordinary dished head.)

The writer desires to present Figs. 15 and 16, which indicate how a vessel design within the elastic limit of the steel behaves. These tests were made on a 72-in. welded vessel with 2-in. wall thickness and without a double thickness at the joint. The results on a large-radius dished head check very closely the author's results. The failure results on these tests checked the elastic results very closely.

D. S. Jacobus.[9] The paper is a valuable contribution, as it compares theory with experiment and shows the forms to be preferred in the construction of dished heads. The advantage secured by employing a longer knuckle radius than 6 per cent of the shell diameter, as specified in the A.S.M.E. Code, is brought out, and a statement is made that the A.S.M.E. specifications for elliptical heads are well on the safe side.

When the rules for dished heads were brought up for revision by the Boiler Code Committee, there was a desire on the part of a number of manufacturers to keep the knuckle radius as small

[9] Advisory Engineer, Babcock & Wilcox Company, New York, N. Y. Mem. A.S.M.E.

as practicable in order to use a flatter head than would be possible with a longer knuckle radius. The rules finally adopted called for heads with a knuckle radius of 6 per cent of the diameter of the shell and a thickness considerably greater than for elliptical heads. In the revised rules the knuckle radius was increased to about twice that formerly used and the thickness of the heads was also increased.

Professor Jasper has brought out an important feature—namely, that where the elastic limit of the material in the head is exceeded, the elastic formulas no longer hold. There are many designs based on the results secured in practice rather than on any particular theory; for example, in the case of a staybolted boiler, the structure should be strong enough to allow one of the staybolts to rupture without rupturing the others. If this feature were taken into account in the design of a staybolted structure and an endeavor were made to use an elastic formula and to keep within the working stresses, it would be found to lead to impractical results. Long use has shown that certain formulas give safe results for structures where the form is such that on exceeding the elastic limit at certain points there is an advantageous redistribution of the stresses.

In structures operated at the higher temperatures, where the effect of creep is the governing factor, the elastic formulas do not hold and the redistribution of the stresses due to creep is advantageous in all structures which tend to distort to a stronger rather than to a weaker form.

The author is to be congratulated on having prepared a useful and highly interesting paper.

Author's Closure

The principal criticism in the discussion by Mr. Coates relates to the validity of Equation [8]. The author presumes that the "correct values" given by him have been obtained in the following way:

The equations for determining displacement, [69a] and [69b], quoted by Mr. Coates in his discussion, appear in the paper "The State of Stress in Full Heads of Pressure Vessels" in the form,

$$\left.\begin{aligned} u &= \cos\beta \int^{\beta} \frac{\epsilon_1 R_1 - \epsilon_2 R_2}{\cos\beta}\, d\beta \\ w &= r\tan\beta + \epsilon_2 R_2 \end{aligned}\right\} \quad \ldots\ldots\ldots [20]$$

For a two-radius head R_1 would be the knuckle radius r, and R_2 would be given by the equation

$$R_2 = r + (a - r)\sec\beta$$

as shown in the Coates paper, at the top of page 10. ϵ_1 and ϵ_2 are the strains. By means of the usual stress-strain relations [21], the equations for membrane stress and the foregoing expression for R_2, the strains can be written as functions of β. Inserting all this in [20] and performing the necessary integrations, there results for the radial displacement,

$$w = \frac{Pr^2}{2Et} K \sin\beta \left\{ \tan\beta \left[2m + \frac{K^2}{3}(\sec^2 B + 2) \right] + K\left(m + \frac{1}{2}\right)\left[\tan\beta\sec\beta + \log(\tan\beta + \sec\beta) \right] \right\} + R_2\epsilon_2$$

where $K = a/r - 1$ and $m = 0.3$, Poisson's ratio.

From the data at the foot of the third page of this paper, there would be, at the knuckle-crown junction,

$$\sec B = 1.69,\ \sin B = 0.085,\ \tan B = 1.36,\ K = 4.15$$

and the quantity in the bracket above would become

$$1.36\ [0.6 + 5.72\ (2.84 + 2)] + 4.15 \times 0.8\ [1.36 \times 1.69 + \log 3.05] = 38.4 + 11.3 = 49.7$$

From the relations above, $R_2E_2 = \dfrac{PR_2}{2Et}\left(2 - \dfrac{R_2}{r} - m\right)$

$$= \frac{P}{2Et}(3136)(-6.3)$$

$$= \frac{P}{2Et}(-19{,}750)$$

and

$$w = \frac{P}{2Et}\left[49 \times 4.15 \times 0.805 \times 49.7 - 19750 \right] = \frac{P}{Et}(-5800)$$

which agrees closely with the result obtained by Mr. Coates.

When the membrane displacement of the crown is calculated, and the quantity a_1 is found from the bending and membrane displacements of the knuckle and crown (lack of space forbids giving this work in full), there results

$$a_1 = \frac{P}{Et}(3450)$$

and the bending stress becomes, for 400 lb. per sq. in. pressure, $\dfrac{P}{at}(2020) = 11250$ lb. per sq. in. There would also be a membrane stress of about 4600 lb. per sq. in. (using average of hoop radius R in knuckle), which brings the tensile stress up to 15,850 lb. per sq. in. It would appear that something should be added to this for the bending set up by the knuckle-shell discontinuity, though this may be dying away at the point considered.

In the list of stresses, see the second column on the sixth page, the stress governing design, calculated from the formula $S' \max = \dfrac{0.147\ PA^2K}{at}$ is given as 18,000 lb. per sq. in. Comparing this with the foregoing, it will be seen that results from the simplfied formula for practical calculation do not differ much from those obtained by the complete theory. As indicated elsewhere, Mr. McLean Jasper has found a correspondence with experiment.

The results given in the paper can be obtained without using membrane displacements.

At the crown-knuckle joint, the membrane hoop stresses, calculated from the dimensions of the knuckle and crown, respectively, are, by Equation [7],

$$\frac{PA}{2t}\left(2 - \frac{A}{r}\right) \quad \text{for } R = A$$

$$\frac{PA}{2t} \quad \text{for } r = R = A$$

If the head be considered to act as a membrane only, this would mean two different hoop stresses acting simultaneously round the crown-knuckle edge. Such an impossibility can be averted by bringing in the bending hoop stresses.

From [12] and [18] the bending hoop stress is, at the edge:

$$\frac{h}{t} = \pm\frac{Ea_1}{A} \text{ for the knuckle or crown, respectively.}$$

Then the total hoop stresses become, at the edge:

$$\text{For the knuckle } \frac{PA}{2t}\left(2 - \frac{A}{r}\right) + \frac{Ea_1}{A}$$

$$\text{For the crown } \frac{PA}{2t} - \frac{Ea_1}{A}$$

Equating these, and solving for a_1,

$$a_1 = \frac{PA^2K}{4Et} \quad \text{if } K = \left(\frac{A}{r} - 1\right)$$

This is the result given at the top of the first column of the twelfth page, from which the crown-knuckle bending stresses are obtained. The expression at the top of the second column of the eleventh page, for the shell-knuckle edge, can be obtained in the same way. In the foregoing no use has been made of expressions for membrane displacement.

The point brought out by Mr. Waters, regarding bending stresses near the center of the crown, is of great interest. As, however, they are relatively small as compared with those at the surface discontinuities, the latter would always govern design.

Mr. McLean Jasper mentions in his discussion the impression that there is only one elliptical-head design. The last paragraph before the Appendix, on the ninth page, to which this appears to refer, means that there is one ellipse for a given major and minor axis. It is gratifying to note the close check which he finds with the results given in the paper.

The discussion by Dr. Jacobus emphasizes the very important principle that a theory should be considered much more as a guide than a dogma. The author has tried to keep this in mind during the preparation of the paper, letting at least the general results of experiment be the determining factor.

Practical Application of Alloy Steels to Locomotives

By C. E. BARBA,[1] NORTH BILLERICA, MASS.

In the critical railroad situation, methods for reducing costs are important. The reduction of locomotive-maintenance costs by the judicious use of new ferrous-metal products is advised—those very products developed by steel-mill research for the industries aided by the railroads. The best of engineering practice should be applied to the design of built-up assemblies that will realize maximum strength and will reduce wear and tear of joinings, such as the integral bed frame for the locomotive foundation. The high initial cost of using nitrided steel and other special materials, as advocated in the paper, is expected to be justified by the reduced costs of maintenance and loss of service of locomotives.

ECONOMICAL and dependable transportation has been responsible in a great degree for our social, economic, and industrial development. For 75 years the railroads have provided such facilities, with only the competition of waterways and horse-drawn vehicles. For the last 25 years they have remained the main channel over which goods and people have been moved, but during this period the development of the automotive and the aviation industries has contributed in no small measure to the critical transportation situation now termed "the railroad problem."

The following discussion aims to focus attention upon the reduction of locomotive-maintenance costs by the judicious use of new ferrous-metal products developed by extensive research in the mills of the steel producers for those very industries built up by the railroads.

Carbon Steel

A special grade of plain carbon forging steel for heavy forgings, such as axles, driving rods, and equalizers, with rather close limits on carbon, manganese, and silicon, has been extensively used for a number of years. The degree to which this becomes a special steel rests with the customer, as to how narrow are the limits in chemical analysis he imposes upon the steel maker, and with the steel maker as to how much care and time he takes to produce a clean, sound material. More time in the furnace for refining and deoxidation adds to the net cost of the material, and the customer who buys the cheapest product will undoubtedly get the poorest product.

The chemical analysis of the ordinary carbon steel is generally within the following limits: Carbon, 0.40 to 0.55; manganese 0.50 to 0.80; sulphur, 0.04 (maximum); phosphorus, 0.04; silicon, 0.15 to 0.25. This steel is generally normalized and drawn after forging.

The following physical properties are representative of this material as heat treated: Yield points, 47,300; ultimate, 81,900; elongation, 25 per cent; reduction, 50 per cent; Brinell, 170; Izod, 10 to 20 ft-lb.

Carbon Vanadium

Modern locomotives brought about a demand for higher-grade steels. A carbon-vanadium steel was the next step in the forging-steel development. The chemistry of this steel is practically the same as the ordinary carbon steel, with the exception that manganese is increased and vanadium is added to show 0.15 to 0.18 in the steel. Vanadium in this amount helps as a scavenger to clean out the manganese oxides and iron oxides, and leaves very minute particles of finely divided vanadium oxide in the metal. These fine particles act as mechanical obstructions to prevent the growing of large grains in heating operations, producing a much finer grained steel than the plain carbon grade. It is generally stated by railroad men that their experience has shown that increasing the vanadium content above 0.25 does not improve the steel, and many go so far as to say that higher vanadium is even detrimental.

An average analysis of carbon-vanadium forging grade is as follows: Carbon, 0.50; manganese, 0.90; phosphorus, 0.030; sulphur, 0.040; silicon, 0.27; vanadium, 0.23.

The best practice on this steel has been found to be as follows: After forging, heat slowly to 1600 deg. fahr. to break up forging grain size; cool in still air; reheat to 1475 deg. fahr. to refine grain; cool in air; draw at 1100 deg. fahr.; cool in furnace.

The following are representative physicals after this treatment: Yield point, 58,000; ultimate, 105,500; elongation, 25 per cent; reduction, 54 per cent; Brinell, 185; Izod, 10 to 25 ft-lb.

These physical properties offered a distinct improvement over the plain carbon steel, and a large tonnage of carbon-vanadium steel has been applied to locomotives in the last 10 years.

Nickel and Nickel-Manganese Steels

About five years ago the need was felt for still better physical properties in the way of toughness and shock resistance. The nickel-manganese steel was the result of this search for better materials. Its production has steadily increased over the last four years as more railroads have specified its use in new engines and in repairs to existing locomotives. Nickel atoms displace iron atoms in the iron-space lattice and increase the toughness of the ferrite. Manganese is specified from 0.75 to 0.95, which raises the physical properties, while carbon is reduced to a range of 0.20 to 0.27, thus producing a steel with only half as much of the carbon—the hardening and embrittling element. The higher manganese neutralizes the effect of the lowered carbon and the nickel toughens the whole mass. The result is a steel that when normalized and drawn shows physical properties well up with carbon vanadium, but the ductility and toughness are very appreciably increased.

The following values were observed on a normalized and drawn piston rod, 5¼ in. in diameter: Yield point, 61,900; ultimate,

---

[1] Mechanical Engineer, Boston & Maine Railroad. Mem. A.S.M.E. Mr. Barba was graduated from Lehigh University in 1901, with the degree of Mechanical Engineer. His engineering activities have been equally divided between manufacturing and railroad engineering—in the former as superintendent of manufacturing enterprises and in the latter as draftsman to mechanical engineer. He studied manufacturing methods in England and Scotland and railroad practices in Germany.

Contributed by the Railroad Division and presented at the Birmingham Meeting, Birmingham, Ala., April 20 to 23, 1931, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

91,300; elongation, 28 per cent; reduction, 68 per cent; Brinell, 183; Izod, 90 to 93 ft-lb.

The Izod value or shock resistance of this steel is from five to ten times greater than carbon or carbon-vanadium steel.

The possibilities of this nickel-manganese steel, from the standpoint of heat treating to obtain higher physical properties and reduce the weight of moving parts, are just beginning to be realized. A $5^1/_4$-in.-diameter piston rod was water-quenched and drawn at 800 deg. fahr., and the following physicals were found in a test taken midway from surface to center: Yield point, 90,000; ultimate, 113,000; elongation, 23 per cent; reduction, 66 per cent; Brinell, 241; Izod, 92 to 95 ft-lb.

This high elastic limit with the high impact figure opens up the possibilities of weight reduction of reciprocating parts. A set of rods based on a 90,000 elastic limit, and calculated on the same factory of safety as a 45,000 elastic limit steel, would show a net reduction of 40 per cent in the weight of the reciprocating parts. When adopting this specification, it is necessary to redesign the rods so as to produce a full streamlined effect and get away from all short corner fillets and to use the thin-sectioned, wide and high I-beam section for the center parts of rods. A weight reduction of 40 per cent in reciprocating parts would be of incalculable benefit in the reduction of vibration, pound on rails, and in the bending moments on axles, which matters are at present receiving some attention.

Boiler Plate

Boiler plate for shells has been increased in strength by addition of nickel, and engines can now carry higher pressures without increase in shell thickness and boiler weight. Nickel-steel fireboxes are still in the experimental stage, some showing up well and others very poorly.

Chromium iron, with 18 per cent chromium, is under test for heat-resistant side sheets in fireboxes, and so far appears to be the answer to the fire and corrosion cracking experienced where engines are operated at a high rate of evaporation.

Wrought Iron and Steel

Low-carbon steels and irons are rapidly replacing the old wrought-iron staybolt. Canadian railroad operators have for nearly 10 years been using a low-carbon steel or ingot iron—on one road with a nickel content of 5 per cent and on the other of 2 per cent. The ingot iron with 2 per cent nickel is rapidly gaining on United States railroads.

Copper-Bearing Steels

Copper-bearing steel with copper up to 0.25 has many adherents as a staybolt material. Another type that is standing up well in service is an ingot iron with 0.50 copper and 0.10 to 0.15 molybdenum. The latter analysis is also used in heavy tonnage in the form of boiler tubes in stationary boilers and is proving very satisfactory in districts where corrosion and pitting of locomotive flues is a serious factor. For firebox plates in corrosive waters, it is showing definite superiority over ordinary steel plate.

Carburizing Material

A recent development in carburizing material suitable for case hardening in general use on railroads is the 2 per cent nickel iron. This material can be quenched in water direct from the carburizing pot and will always produce a hard case and a very tough core. Wrought iron is the only other iron that will do this. Steel invariably shows a brittle core when quenched direct from the carburizing pot. The 2 per cent nickel iron, for this reason, is distinctly a railroad material and has been developed as such, both for staybolts and for case-hardened brake rigging, spring rigging, and motion work pins. It has some application in the form of case-hardened liners for driving rods to protect them against wear from floating bushings.

Special Steels

In the matter of bolts used in frames and boiler studs, wrought iron grew up with the railroads, but increased stresses are gradually eliminating its use. The elastic limit of wrought iron is too low to carry the loads imposed upon it in modern locomotive construction, and when overloaded, it stretches, then becomes brittle (crystallized), and fails without stretching.

Wrought iron physicals are about as follows: Yield point, 35,000; ultimate, 48,000; elongation, 30 per cent; reduction, 40 per cent.

To replace this material for bolts subjected to nominal strains, we have turned to the S.A.E. 2115 analysis. This shows about 0.15 per cent carbon and 1.50 per cent nickel. It is generally found to show about the following physicals: Yield point, 43,000; ultimate, 62,000; elongation, 28 per cent; reduction, 65 per cent.

This analysis is doing very well when applied to boiler studs, cylinder studs, and frame bolts in the lighter engines; but there are applications requiring still higher physical properties, and these are met by using various analyses, such as S.A.E. 2320, 3120, 3130, 6130, and 1330 (carbon manganese), and the $2^1/_2$ per cent nickel-manganese steel used by one of the Canadian railroads. Chromium steel 5130 is used by one road in this country.

The general treatment is to quench the forged bolts or parts in oil or water and draw back to the required physical properties. The final properties are generally about as follows: Yield point, 90,000; ultimate, 105,000; elongation, 20 per cent; reduction, 60 per cent; Brinell, 225; Izod, 50 to 70 ft-lb.

The following parts are often made from the foregoing steels: Superheater bolts, cylinder bolts, frame crosstie bolts, binder bolts, coupler bolts, piston keys, driving-rod collars. Many other applications have been found where a heat-treated alloy analysis will stop breakage and renewal of parts.

Nitrided Steels

About the last word in the steel world are the special steels that have been produced to use in applying the super-hard cases that are built up by heating the finished parts in ammonia gas. These are known as nitrided steels.

The ammonia gas dissociates into nascent nitrogen and hydrogen at about 850 deg. fahr. The parts to be hardened on the surface are first formed, then heat-treated, and then finish-machined, ground, and polished before placing in the retort for nitriding. The nascent nitrogen at 925 to 975 deg. fahr. combines with the iron, chromium, and molybdenum in the steel to form nitrides of these elements, and these nitrides form the hardest wearing surface that science has been able to develop, excluding the carbide cutting tools. It has been known as early as 1907 that nitrogen would, under certain conditions, combine with iron to form iron nitrides and that these iron-nitride grains were exceedingly hard. But the hard cases formed were brittle and spalled off from the base metal easily.

Various investigations worked to bring out steels that would form tougher cases with nitrogen. The result has been a steel containing chromium, molybdenum, and aluminum. The chrome and molybdenum toughten the case, and the aluminum acts as a catalyzer and stabilizer to maintain the high surface hardness. The steel is produced in four carbon ranges, running as low as 0.12 to as high as 0.65.

The heat treating and machining are all completed before the hardening operation, and that is conducted at low temperatures, 950 to 1075 deg. fahr., so that distortion and subsequent grinding are eliminated and parts may be machined to very accurate di-

mensions or fit and then hardened with little danger of warping.

This case is the hardest and most uniform structure ever produced on steel, and as such its great wear resistance is opening up undreamed-of possibilities of long-lived parts for locomotives.

The basic idea in the application of nitrided steels to locomotives is to select for its application those parts in which frictional wear causes heavy expense for renewals.

The fundamental law in bearing metals is that if the two parts are of equal hardness they will both wear. If one is hard and the other soft, the softer will take the greater wear.

In applying nitrided steel the endeavor should be to replace those parts that are most expensive to replace and to throw the wear over on the least expensive part.

Where a rotating shaft or pin is to be considered, it is generally cheaper to make the bushing take the wear.

The metallurgists have some restrictions as to microstructure, effects of time and temperature, gas pressure in retorts and their effects on the hardness, depth of case, toughness, etc., but the mechanical engineer is mainly interested in two things:

(1) Can wear be reduced without sacrificing strength?
(2) Will the increased life of wearing surface compensate for the higher cost of the material used?

In preparing a wearing surface of nitrided steel one must remember that he is dealing with a surface hard enough to cut glass. Nitrided wearing surfaces must be lapped in or polished, or glazed, before the nitriding treatment is applied. After the nitriding it is too hard to do much with, and furthermore the best wearing part is on the surface, and grinding after nitriding is considered detrimental and as poor practice. As to the selection of bearing metals to be used against nitrided surfaces, this is of vital importance. New and harder bearing metals are being used with great success. Any application of nitrided steel should also be considered from the standpoint of the bearing metal to work with it, and the men who are building up test data and experience should be consulted.

Theoretically, all moving parts should be on a par with each other as to wear-resisting qualities, so that the weakest link, all other things being equal, in the maintenance and replacement of parts becomes the locomotive tires.

The service life of wearing details by the judicious selection of materials and improvement of design to best utilize these materials then becomes a multiple rather than a fractional part of the tire-service l fe.

The following parts in the running gear of a locomotive are enumerated in the order in which renewals due to wear take place: Piston rings, valve rings, cylinder bushings, valve-chamber bushings, piston heads, piston-valve parts, piston rods, valve rods, valve crossheads, crosshead shoes, guides, valve-motion pins and bushings, valve links and blocks, locomotive tires, trailing-wheel tires, crown brasses (driver box), hub face (driver box), side-rod bushings, wrist and knuckle pins, crankpins, shoes and wedges, driving axles, hub of driving-wheel centers, hub of trailing-wheel centers, hub of engine-truck wheels or centers, swing-motion details of engine truck, spring-rigging details.

Similarly, the boiler may be subdivided into the following renewable details: Grates and appurtenances; firebox sheets, side, crown, back, and flue; tubes and flues; staybolts, rigid and flexible; superheater, header and units; exhaust nozzles; smoke-box netting.

In order to illustrate just what can be accomplished, it might be well to start at the power end of the locomotive, the cylinders, and analyze conditions.

Locomotive cylinders are bushed with cast-iron bushings as a means of preserving the cylinder and saddle throughout the life of the locomotive. Cast-iron cylinder bushings of refined iron are undoubtedly the best practice developed through the years, but due to the unsupported piston, the bushings will wear out of round and in time must be rebored. The same thing holds true of the piston, or of the composite type, the bull ring, which obtains a similar wear and must be built up or replaced with a new one.

This, however, requires dismantling of the cylinder heads, piston, etc., and the tieing up of a locomotive, and it would seem rational to assume that if the present bushings were replaced with nitrided-steel bushings, having a hardness equal to 850 to 950 Brinell, lapped to a very high polish, the hardness would never be disturbed, as the range of temperature is well within that used for nitriding purposes. The same holds true for piston-valve bushings.

Such an installation would undoubtedly mean the elimination of future rebores and the prevention of leaks past the snap rings, thereby conserving steam.

Proposed Manufacture of Cylinder Bushings

The development of nitralloy cylinder bushings, other than forgings, presents a most interesting problem in that plate can be used by rolling into shape and welding by the atomic-hydrogen-arc process, after which it may be machined to size, bore-lapped, and nitrided prior to application. The following methods are used: (1) Welding of $1^1/_4$-in. thick plate; (2) heat treatment of welded assembly; (3) machining and nitriding of welded assembly.

The work was undertaken with an objective of forming large cylinder bushings for locomotives. The bushings were to be made

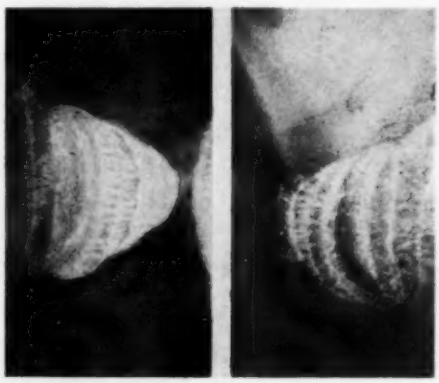


Fig. 1 Microphoto Through the Case
(At left, through the weld after heat treatment; at right, across the weld.)

of $1^1/_4$-in. plate rolled into cylinders, welded, heat-treated, machined, and then nitrided.

After two or three trials the proper technique was developed, and several welds free from blowholes were made. Chemical analyses of the wire used and the plates and welds were as follows:

	C	Cr	Al
Welding wire.....	0.256	01.42	1.13
Plates welded.....	0.26	01.21	0.95
Weld No. 1.....	0.122	01.32	0.86
Weld No. 2.....	0.178	01.30	0.86
Weld No. 3.....	0.139	01.35	0.82

The three welds shown averaged about 10 points less carbon and 27 points less aluminum than the original analysis of the rod. The plates had about the same carbon content as the rod. The aluminum remaining in the welds is well above the 0.60 per cent minimum required to produce a proper case. The drop in carbon is really beneficial, as the weld is more ductile than if the carbon was all retained.

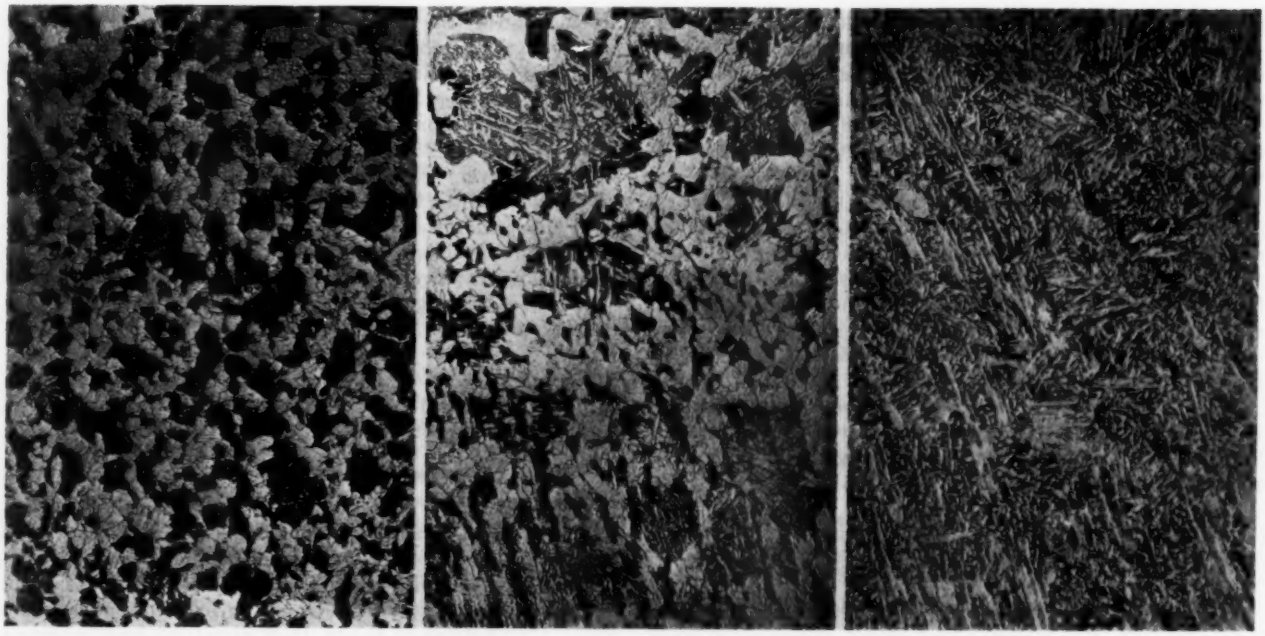


FIG. 2 MICRO-STRUCTURES AS WELDED × 100
(At left, parent plate; in center, weld junction; at right, welded metal.)

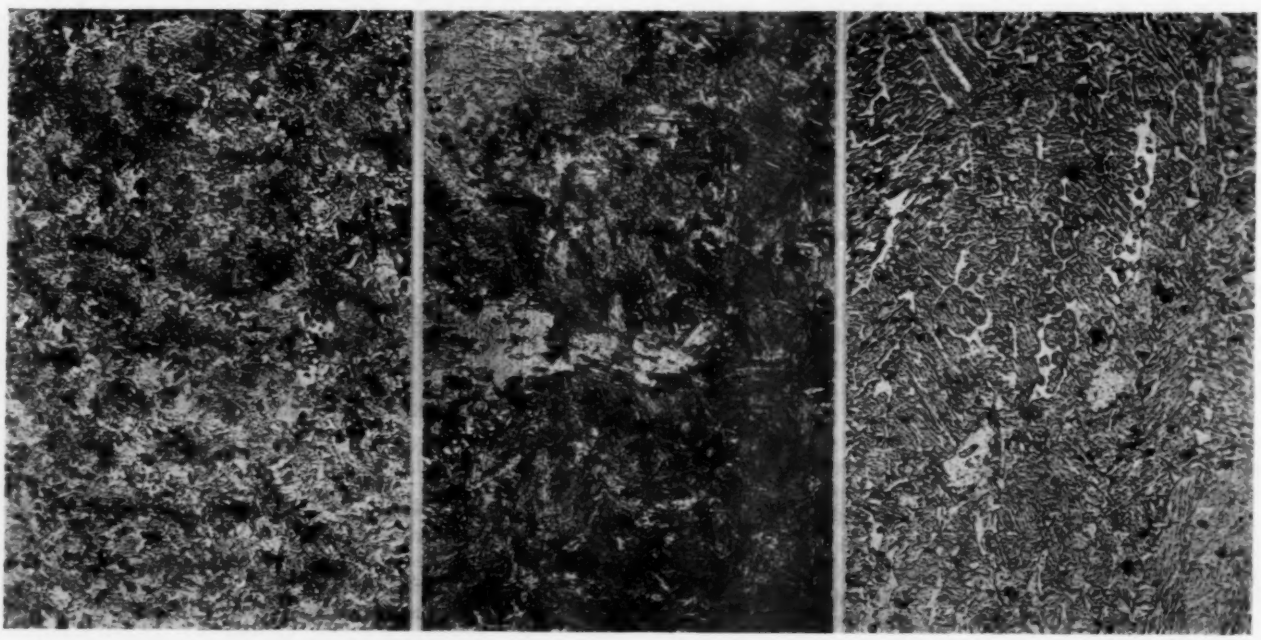


FIG. 3 MICRO-STRUCTURES AS TREATED × 100
(At left, parent plate; in center, weld junction; at right, welded metal. Water quenched at 1700 deg. fahr.; drawn at 1325 deg. fahr.)

A section about 6 in. long with weld running full length of it was heat-treated, 1700 deg. fahr. in water and 1325 deg. fahr. in air, machined from $1^1/_4$ in. thick down to $^3/_4$ in. thick, ground on both flat surfaces, and then nitrided for 42 hours at 925 deg. fahr.

Hardness readings showed around 850 to 900 Vickers Brinell. No difference was apparent between the readings on the welds and those made on the parent stock.

Micro-section through the case showed the same penetration on weld as on parent stock. Microphotos, Figs. 1 to 5, show the various structures.

The atomic hydrogen process can be used successfully to obtain welds of special steel analysis that will produce a nitrided case identical in hardness and micro-structure with the parent metal.

There are dendritic structures retained in the welds that might be broken up by forging, but are not thought to be detrimental where the bushings are under no particular strain and are only applied to obtain a surface for resistance to wear of piston and rings.

The success of the operation is entirely dependent on the skill and art of the welder, and a policy of rapid fusion and the application of small drops of molten metal at a time must be observed. Puddling the metal will invariably cause excessive loss of aluminum and will develop blowholes in welds.

The utilization of special steels and the engineering problems in connection therewith are outlined for various locomotive details as a means of stimulating the art.

Piston Heads and Rods

The piston head, irrespective of design, can be made of a special steel forging or casting and the rim be nitrided after grooves have been machined and lapped to dimensions, so as to eliminate groove wear from snap-ring action; likewise piston-valve parts.

With respect to snap rings, these can be made of a bronze alloy, L-shaped, capable of sustaining temperatures from 900–950 deg. fahr., without removing the snap, which, in contact with a highly polished surface of extra hardness, is bound to give maximum life, as well as that of the piston head.

The possibilities of the use of an extended piston rod should be given further consideration, as the conventional design now in use cannot be sustained as one in which the best of engineering principles have been incorporated.

The weight of the piston should be supported at the rear by the crosshead and at the front by a pivoted bearing for alignment, which undoubtedly would minimize or practically eliminate the wear of the piston-rod packing and the circumferential wear of the piston head, and simplify the work of the snap rings, since we are starting out with a true bore lapped for accuracy and hardened. The only compression variable of the rings would be that created by the vertical and lateral wear of the crosshead.

The extended piston rod is not a new idea, but its abandonment from general use was largely due to the inability of controlling the wear between crossheads and guides, both vertical and horizontal, within limitations which would not affect the bearing surface of the extended piston-rod crosshead, or support, at the front end of the cylinder.

The adoption of nitrided steel for piston rods the wearing surface of which can be hardened so as to eliminate wear should prolong the life of the piston-rod packing, as it is certainly less expensive to replace rings than it would be to grind down piston rods.

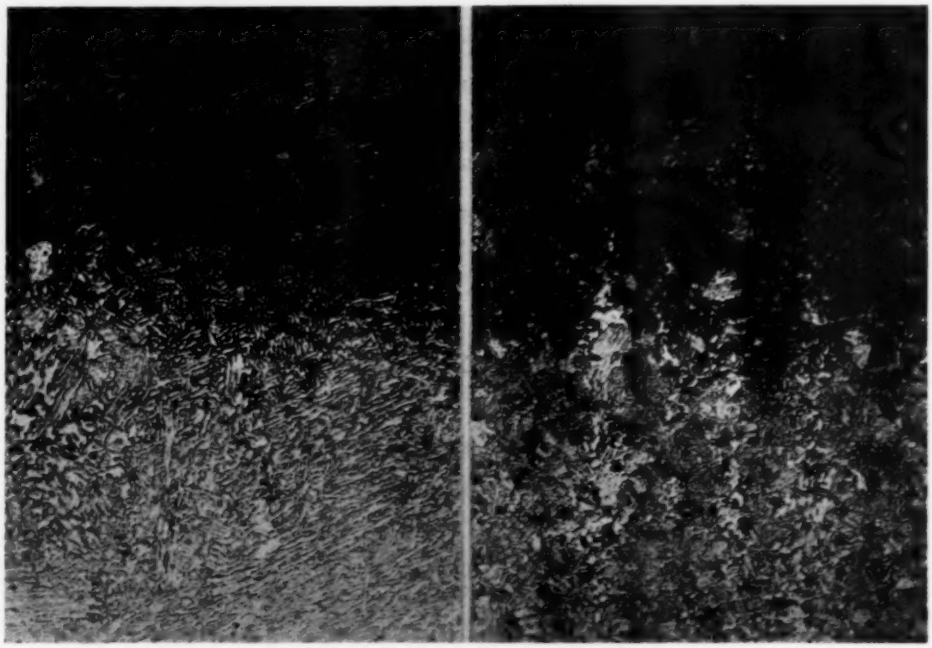


Fig. 4 Nitrided Cases
(At left, welded metal; at right, parent plate.)

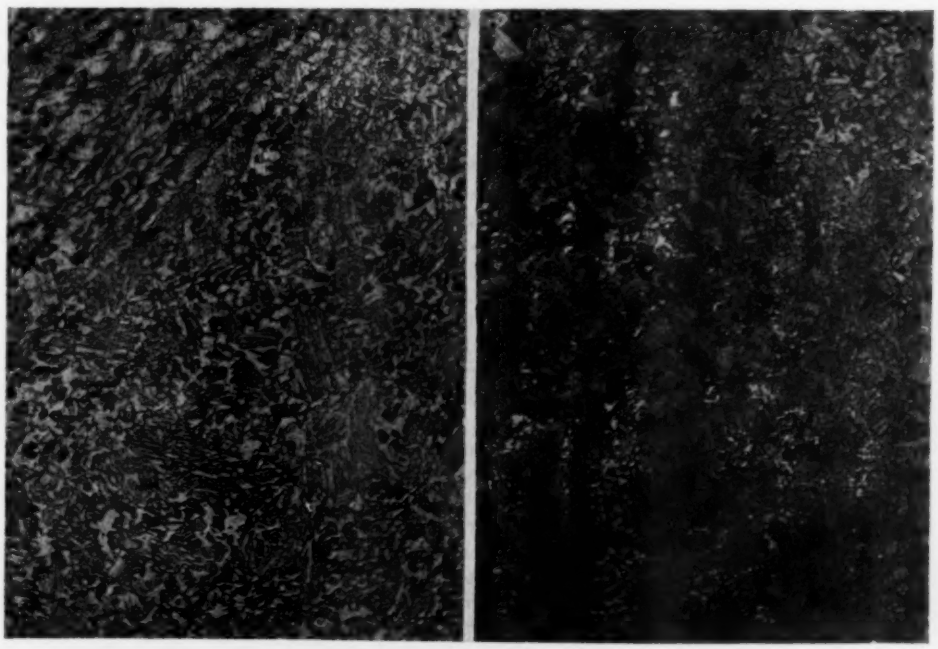


Fig. 5 Core Structures
(At left, welded metal; at right, parent plate.)

Crosshead Shoes and Guides

In the case of crosshead shoes and guides it is considered best to apply nitrided plates to the guides and to apply bearing metal to the crossheads or crosshead shoes in the form of channels,

which can be quickly removed and replaced at very low cost. The present practice calls for removal of worn guides and building them up with welding bronze; then they go to the blacksmith shop to be annealed and straightened, then to the planer and the surface grinders and finally back to the engine, where they must be reset. Generally the boltholes all have to be reamed again and new bolts fitted, so that rehabilitating a worn crosshead guide becomes an expensive job.

The proper selection of steel for controlling the wear of crosshead shoes and guides, or the designing of either on the basis of increasing the bearing or pressure area, thus reducing the pressure in pounds per square inch to a minimum and thereby increasing its life, or the utilization of a special steel and maintaining the present bearing areas, should assist in reestablishing a design of paramount importance in the reduction of maintenance.

As a matter of fact there is no reason why guides and crossheads should not be capable of operating 150,000 miles without the necessity of a take-up during that period. The wear of guides is partly due to having a piston hanging at the end of a rod one end of which is fastened to the crosshead, very much out of balance and ready to drop in its movement equal to the difference in diameter between the piston head and the bore of the cylinder, the tendency of which is to cock the crosshead at the end of the stroke.

Side-Rod Bushings

Side-rod bushings have ever been a source of trouble due to rapid wear. The selection of bearing metals is a matter of utilizing a bronze that can be subjected to a higher working temperature than is now possible, and when used in connection with hardened crankpins the wearing qualities can be increased, while seizure from operating temperatures and lack of proper lubrication will be practically eliminated.

Driver Boxes

Driver boxes as designed at present have not sufficient sustaining qualities for the application of a pressed-in crown brass, due largely to the limitations of present pedestal spacings, all of which can be modified by the elimination of shoes and wedges.

The present use of shoes and wedges is to take care of wear of the box, liners, and shoes. To minimize this wear, so that they will operate between shop repairs within the tolerances prescribed, will necessitate the use of nitrided-steel wearing surfaces. Furthermore, the use of shoes and wedges ofttimes creates improper transverse alignment or parallelism of driving axles, which, if elminated, simplifies shop operation.

The design of the box could therefore be modified by utilizing a portion of the thickness of the present shoes, wedges, and floating liners by increasing the over-all width. The wearing surface of the driver boxes and pedestal fit should be a flanged liner, on both pedestal and box, made of manganese steel not less than $^3/_{16}$ in. thick, the life of which should not be less than 200,000 miles.

Such a design would permit the use of a crown brass, more rugged in design, and which can be applied without pressure, so that replacement is simplified and present costs minimized.

The ever-increasing lateral between driver box and wheel hub can practically be eliminated by the application of nitrided-steel liners inserted into a recess in the hub of the driving wheel and welded on the periphery, in conjunction with a similar liner inserted in the face of the driver box, the hardness of which should be approximately 850 to 950 Brinell. Such liners, when coming in contact, either lubricated or non-lubricated, should be good for 200,000 miles.

Driving Axles

Locomotive-axle failures are an increasing difficulty and expense. There are two types of failures. The one type is a fatigue crack starting in the fillet next to the wheel center. This type of failure generally starts in a rough finished fillet or where the brass cuts a shoulder in the fillet. Anything that will produce a notch effect on the bearing surface of the axle will start a crack that will progress until the heavy starting torque and the bending moment on the axle, when the crankpin passes over the center, will produce enough strain to rupture the remaining portion. The second type of failure is that of hot boxes, which produce thermal checks on the journal surface, and these are starting points for progressive fractures.

Anything that would relieve the axle of the duty of supplying a wearing surface as well as resisting the high fiber stress due to flange pressures would prolong the life of the axle. Axle bushings of nitrided steel, shrunk on the axle, are now being tried out in testing laboratories.

Furthermore, there is no reason that a pair of wheels, when mounted, should have journals re-turned so as to true up the wear or abrasion; as with a pair of driving-wheel centers, properly mounted on a nitrided-steel axle and having journals hardened or bushed together with nitrided-steel crankpins, properly quartered, the unit as a whole should give a maximum life and should never be disturbed exclusive of the replacement of tires.

Locomotive Links and Motion Parts

Locomotive links, the design of which has not been altered for many years, require continual regrinding, due to the movement of the block, but if made of nitrided steel, regrinding can be eliminated if used in conjunction with a hardened block having a Brinell of 500 to 600.

When one considers the importance of proper steam distribution, the ideal valve motion, irrespective of type, would be one in which all parts would never require any adjustment due to wear of pins and bushings. Here again the utilization of nitrided steels will go a long way toward establishing a greater degree of accuracy and more perfect steam distribution than is now possible.

Spring Rigging

The function of a spring rigging is one to take care of track irregularities, tire wear, etc., and the proper distribution of weights on the running gear. Consequently, spring design and the materials selected play a most important part. The gradual elimination of the present elliptic spring and the adoption of a different form of spring suspension than used at present, together with a redesign of hanger that will produce minimum wear and preserve the alignment of locomotive, irrespective of tire or crown-brass wear, is in order.

There is no good reason why the designing and manufacturing of locomotive springs should not follow automotive practices, due to their widespread application and their varied services.

From the foregoing it will be observed that there is opportunity for radical departures from present-day practice; but unless those directly interested in engineering principles covering locomotive design change their viewpoint in the utilization of ideas and materials used so successfully in other lines of industry, we will be unable to profit from the progress which has been made in those industries for our own use, as a possible means of decreasing maintenance.

Firebox Steels

The major problem in the selection of steel is one of temperature, and naturally a steel is required that will resist whatever the maximum temperatures of a firebox may be (under various operating conditions) so as not to break down the structure of the material. This in itself is quite a large contract, and various

grades of steel have been tried with varying results, but not with sufficient assurance that their adoption would eliminate all of the troubles now experienced. It would therefore appear that one of the major problems is one of design, in which the flow of gases over the arch will be similar to the flow of water over a dam—that is, straight-line movement. The influence of the draft-producing medium at the center of the smokebox undoubtedly causes a stream flow of the gases converging toward that center, thus materially lessening gas flow and heat transfer away from the center of the tube sheet, all of which has a tendency toward creating hot spots in the firebox rather than maintaining a uniform temperature throughout.

Furthermore, the percentage of carbon in firebox sheets is believed to require careful consideration and should be a minimum, and excluded if possible, as any unnatural carbon content must have a hardening influence when heated and cooled, sometimes abruptly, under the present operating conditions. Experiments are now being conducted on a brick side-wall lining to determine the effect of controlling the rapidity of temperature change by a more uniform heat distribution.

Grates

The design of grates and the selection of a material that will resist heat at higher temperatures than any material now in use would be a step in the right direction, although a grate may be so designed that its section may be sufficient to reduce the temperature at the top of the grate, well within the confines of the critical range of the material, by the net area and distribution of the air inlets. This, however, requires some experimentation to determine.

Staybolts

Design likewise should more fully enter into the final analysis of the staybolt rather than the mere selection of better material, or type of material, as steel versus wrought iron. The entire firebox (were it not for the attachment of the firebox sheets to the mud ring) is a floating chamber supported by staybolts. The water legs, increasing in width from the mud ring to the crown sheet, with the space between the crown sheet and the wrapper sheet being in excess of that of the widest width of the leg, calls for staybolts of varying lengths, but practically all having the same diameter.

Considering the staybolt as a cantilever beam—that is, by assuming that there is no movement of the wrapper sheet relative to the firebox sheet—the number of pounds required to deflect a long staybolt is naturally less than that for a short staybolt having the same diameter. Consequently there are zones in each firebox side sheet that are restrained due to the constant force of expansion at uniform temperatures, by the shorter bolts not deflecting the same amount as the longer ones, which produce strains within the sheets that in time will produce hairline cracks around staybolts, and eventually failures will result.

As a suggestion why not divide the application of staybolts in a vertical plane into horizontal zones and design the staybolts in certain zones according to their length by assigning diameters so that all staybolts may deflect in unison, assuming that the expansion force is constant. Undoubtedly an application of this character might show some very interesting results.

Wheel Centers

The design of wheel centers and the character of steel used is a subject that, in the author's estimation, requires more research. Taking a concrete example, specifications require that all steel castings must be annealed so as to relieve strains introduced during cooling. This seems to be general practice.

On the other hand, the practice is to take a wheel center, press it on an axle, introducing compressive strains in the hub which may or may not be transferred to the spokes, and then press in a crankpin, the stresses of which go into the hub and spokes, and finally shrink on a tire, which not alone compresses the rim but likewise the spokes, and the result is a wheel center having all sorts of internal strains. As a matter of fact, it is strained far in excess of any strains that may have been introduced originally during the casting and cooling process, and still it is insisted that those strains must be removed and nothing is said of the strains of the completed article.

It would therefore seem that tests are in order calling for the application of extensometers on spokes, hubs, and rims, covering all of these operations for the determination of stresses or for the possible redistribution of metal if it is found necessary, to take care of these strains, which should assist in furthering the life of the present wheel centers.

Locomotive Tires

Present practice in locomotive tires, or for that matter in any tired wheel, irrespective of size, calls for three grades of material—that used in passenger, in freight, and in switcher service—representing graduated increases in carbon content and consequently hardness to resist wear. These materials have undoubtedly stood the acid test for many years. There is a question, however, with metallurgical engineers as to whether the railroad engineering profession has not overstepped its limits with respect to wheel loads, taking into consideration the small contact area that a tire makes with the rail.

This subject undoubtedly covers a great many years of intensive research, primarily metallurgical, but limited in engineering considerations, as mass has not been considered an important item where fluidity of metal (due to great pressures) is concerned. It is believed that better results could be obtained by increasing the thickness of a tire over the conventional thickness so as to have more backbone in assisting the transmission of weight to the rail, and naturally giving more resistance to wear on account of this additional thickness.

Toughness in tire materials is a determinant in wear resistance, but unfortunately in ordinary tire steels toughness is obtainable only by increasing the carbon content, which produces a material of improved wearing characteristic but of a hardness with increased tendency to fracture under shock stress. However, with the change in speeds of freight locomotives narrowing the gap differentiating from passenger-operating characteristics, it is felt that the harder B-grade tires can be used for all A-grade purposes.

Further, to meet the demands of maximum hardness and for high tensile and elastic limit to resist flowing tendencies under heavy wheel loads, and to advance the service life of tires to meet the possibilities of increased life in other wearing parts, considerable attention should be given to alloy-steel development for tires. Tires are now being used with nickel and manganese content, giving harder wearing surfaces without the brittle tool-steel characteristics of high-carbon-content tires. The surface as yet, however, has only been scratched.

Tank Steel

The selection of materials likewise can be extended to the use of tank sheets instead of the present open-hearth steel. Even though copper-bearing steels are being used to a very large extent, other materials can be procured in which corrosion can be eliminated, and therefore the initial cost can be justified, considering the elimination of tank repairs which are now prevalent.

It also would be helpful in designing the slope sheets to have the joints made on the vertical sides instead of where the slope sheet coincides with the side sheets, and providing at least a 6-in. or 9-in. radius in the corners, not alone to assist the coal in

passing to the conveyor more freely, but likewise preventing the corrosive elements from entering the joints made in present-day construction.

The utilization of alloy steels is not the only attack on maintenance. The best of engineering practices and the industrial arts must be drawn upon in the development of integral designs to replace built-up assemblies for the purpose of realizing maximum strength and reducing wear and tear of joinings. This is strikingly exemplified in the design of the integral bed frame for a locomotive foundation.

This represents a radical and bold departure from past practices; yet experience has now been of sufficient length amply to justify the design.

It is believed that the money expended for the initial cost of using nitrided steel and the other special materials referred to, as a means of decreasing maintenance, will be more than justified in the subsequent reduction in maintenance cost and lesser loss of service of locomotives due to required maintenance.

An accounting system should, if necessary, be developed by which the cost of repairs or replacements of individual details of each class of locomotive can be recorded for easy reference and to serve as a source of reliable reference data.

In conclusion it might be well to emphasize the wonderful possibilities in design in conjunction with the utilization of high-grade material which the railroad engineering profession should consider; as the selection of steels to cure all evils that may predominate is only a portion of the problem. Unless the engineer and the metallurgist exchange ideas more freely as to requirements due to present operating conditions, in order that maintenance may be reduced lower than it ever has been in the history of railroading, the advancement of locomotive design will not progress any more rapidly than it has in the past.

The author is greatly indebted to the efforts of the Republic Steel Corporation, through its able assistants Messrs. Morris and Miller, of the Metallurgical Department, for the development of various applications of nitrided steels for locomotive use.

Appendix

INFORMATION TRANSMITTED BY AUBERT & DUVAL, OF PARIS, FRANCE, ON THE REQUEST OF DR. ADOLPH FRY, OF THE KRUPP WORKS, ESSEN, GERMANY

Nitralloy has been used successfully in locomotives for crankpins for driving and for coupling. Some of these pins have been in service for seven years without showing any wear.

A noteworthy fact is that the wear of the bearings has practically stopped on locomotives so equipped; heretofore, when case-hardened pins were used, bearings had to be replaced at least four times a year. During a much longer period of time in using nitralloy, no replacements were necessitated.

On a locomotive of the Deutschen Reichbahn pins have been in service for 350,000 kilometers, or approximately 262,500 miles. These pins showed some wear in places, which necessitated grinding; after grinding the pins were re-nitrided and put back into service, and they gave complete satisfaction.

Nitralloy is also used with success for different parts of the poppet valves of the Caprotti type now being used to replace slide valves. In a poppet the rods, cams, guides, etc., undergo considerable wear, resulting in an ordinary steam loss of about 3 per cent. Tests made to date using nitralloy for these parts seem to show that it eliminates all wear, with the result that there is little or no loss of steam.

Several tests are now being conducted using nitralloy crosshead guides. These tests have not been completed, so that no report can be made at the present time.

Discussion

HOWARD L. MILLER.[2] The intense heat developed by stoker fires of locomotives and the oil-burning locomotive within the past three or four years has proved to be beyond the capabilities of ordinary steel. Firebox renewals have been increasing at a very rapid rate. The main difficulty seems to be in the side sheets, which are exposed to intense heat under the brick arch. This type of failure is distinguished by cracks running from the staybolt holes, mainly in a vertical direction. The rapid rise and fall of temperature in the firebox due to stoker fire is responsible for this type of fatigue failure.

The antidote to this trouble has been proved up in service, and the 18 per cent chrome iron with carbon under 0.10 per cent and silicon around 1 per cent, known as Enduro type A, such as is used for nitric-acid tank cars, has proved to be able to withstand the terrific heat to which the side sheets of the firebox are exposed. This material has a very low coefficient of expansion throughout the range of temperatures involving service and shows about two to three times the creep limit of common steel in the service temperatures of firebox plate. It is also practically non-corrodible as regards boiler water. This material can be welded to common firebox plate using a coated common steel wire.

J. H. HIGGINS.[3] Much success has been achieved during the past 15 years, and the manufacturer feels no little pride in the part he has taken in these improvements, for there has been close cooperation with the mechanical engineer and an earnest endeavor to work out and improve locomotive equipment parts.

The author has covered the progressive steps from carbon steels into the alloy steels, and while these steps have materially reduced failures, some still occur both by wear and fracture, so that we are compelled to continue our experiments along the lines of recent metallurgical discoveries.

Nitralloy steel, with a core hardness up to 350 Brinell if desired and a case hardness of 950 to 1100 Brinell, offers the mechanical engineer opportunities never before at his command.

Many railroads have used nitrided parts on locomotives, such as valve-motion parts, guides, piston rods, and main as well as intermediate crankpins, and outstanding results have been achieved. Many more parts remain to be tried, including driving axles, which undoubtedly could be designed to permit shrinking a nitrided-steel band on the journal. The high surface hardness of this band should aid in lubrication and retard friction. The heat-resisting qualities of Nitralloy steel should be an important factor in reducing, if not eliminating, heat checks. Both these possibilities can be developed without interfering with the present physical requirements of the various specifications. Nitrided steels present opportunities of reducing the present-day cost of maintenance.

AUTHOR'S CLOSURE

With respect to Mr. Miller's comments on firebox steel, the author agrees that the ordinary steels for certain grades of coal and rates of firing do not stand up and that cracks do develop on the water side at staybolts for a distance of approximately 18 in. above the mud ring.

On the other hand, it is believed that the present design of water leg assists in aggravating the conditions, due to the cross-section being too narrow, and that the design must go hand in hand with better grades of steel to minimize failures. The use of high-grade steels alone will not cure all failures, although it may reduce them.

[2] Republic Steel Corporation, Massillon, Ohio.

[3] Materials Engineer, Engineering Materials Department, Camden Forge Company, Camden, N. J.

Coolness of firebox sheets is somewhat of a variable, since the volume of water coming in contact throughout the firebox area varies in density, and it therefore cannot be expected that the heat transfer to the water from a uniform thickness of firebox sheet can maintain the same temperatures, resulting in spotty expansion.

This expansion is controlled by staybolts, all of the same diameter, but with varying lengths in the water leg, resulting in varying resistance offered by each staybolt at its attachment to the firebox sheet, thus causing unequal stresses, as the deflection of a staybolt 12 in. long is greater for a given load than one 9 in.

All firebox sheets should expand uniformly, and staybolts should permit of this, irrespective of the grade of steel used in firebox sheets. This is a contributing factor along with the kind of material selected.

Mr. Higgins' review of the application of nitrided steels is an indication of sound engineering principles brought about by intensive metallurgical developments.

Maintenance is a very large factor in operating expenses, and therefore materials and workmanship are the only known means to bring about a reduction.

Increased wearing qualities cannot be obtained from the present grades of bearing metals and steels coming in contact with them, irrespective of the various systems of lubrication now in vogue, without paying some attention to hardened surfaces, either sliding or rotative.

RR-53-2

Metallurgy in the Railroad Field

By A. E. WHITE,[1] ANN ARBOR, MICH.

The alleged conservatism of the railroads toward changes in the materials going into equipment is explained by their strict adherence to a safety-first policy. The subject matter of the paper is discussed under three main headings of manufacturing methods, materials, and specifications. It is pointed out that, in order to improve a given part, changes in design should be considered which would minimize the chances for defects due to improper or improperly controlled manufacturing practices. Next, a plea is made for better steel-mill practice, closer control, and better inspection. After all these points have been considered, further increases in strength and ductility can be obtained through the use of alloying elements. Attention is given to the steps taken in improving some parts which have previously been a source of serious concern to the railroads.



THIS paper on metallurgy in the railroad field is quite definitely a record of the reactions of an outsider, for the author has done most of his metallurgical work in connection with central stations, the automobile industry, and on the metallurgical aspects of materials for ordnance. It was perhaps the intention in requesting such a paper to secure the unprejudiced point of view of one who holds no special brief for either the steel manufacturers or the railroads.

The railway systems of the United States represent one of the largest industries in the country. They are among the largest consumers of metal products. They have, in consequence, been one of the most important factors in the development of the great iron and steel industries and have contributed to the growth of the non-ferrous industries as well. In this paper discussion will be confined entirely to materials of a ferrous character, such as cast iron and steel.

Railroad companies have been criticized for their apparent backwardness in the adoption of materials which have been developed or have come into common use, particularly in the automotive field, during the past 20 years. They also have been criticized for not taking advantage of the possibilities of increasing strength and other physical properties by heat treatment, in order materially to reduce weights and sections. These criticisms have not taken into account the fact that the by-word of railroad transportation is "safety." In view of what would be the results of any other policy, such a conservative attitude is most certainly to be commended. When one considers that he is practically as safe on a railroad train, even though that train may be speeding through the night at the rate of 60 miles per hour or more, as at home, it is easy to appreciate what the railroads have done in assuring their passengers the maximum degree of safety. How correct this statement is may be appreciated by the fact that only about four or five reportable train accidents occur on Class 1 railroads in the United States per million locomotive miles and that all of these do not result in injury or death. The railroad systems of the United States have evidently done wisely to adopt this policy of safety first, and it is easy to understand the underlying causes for their conservatism toward changes in materials or in their treatment.

The discussion of the subject matter of this paper has been divided into three major sections: manufacturing practice, materials, and specifications.

Manufacturing Practice

Under manufacturing practice will be discussed foundry, steel mill, forge shop, heat-treatment department, and machine shop. A résumé of the problems encountered in each department is presented.

Foundry. The foundry is the source of all cast materials and cast products. The quality of the materials produced in the foundry has been somewhat improved in the last few years by the use of alloying elements. A larger portion of this improvement, however, must be ascribed to more careful methods of handling, closer control, and to more rigid standards of inspection. In some cases the source of the increased quality must be traced back to the design. Properly designed sections are much less likely to carry blowholes, inclusions, or other unsound characteristics than improperly designed ones. When the designer does not take into due account the difficulties that may arise in casting, it is obvious that the composition and quality of the metal have little bearing on the actual strength of the finished piece.

Gray Iron: Railroads use both gray iron and malleable iron. Most of the gray iron carries no alloy additions, though without doubt the advisability of such alloys is now receiving considerable attention. One of the most desirable of the possible alloy additions is nickel, for when present it gives a stronger iron without increasing the difficulty of machining. One must appreciate that it produces these properties only when the cast iron is sound, for it does not, like Aladdin's lamp, possess qualities of making unsound material good.

Malleable Iron: The quality of malleable iron has been vastly improved in the past 15 years. This has been largely due to the efforts of the Malleable Iron Research Institute, which has been instrumental in raising the ductility of the iron from values of a few per cent when expressed in per cent elongation to values of 15 per cent or better. This improvement also has been coupled with increased strength without in any way affecting machinability characteristics. Probably no better example of what may be done through cooperative endeavor in the metallurgical field can be found than that which has been accomplished by the Malleable Iron Research Institute and by those with whom it has cooperated.

Steel Castings: The standards of quality of steel castings for railroad work have been notably raised during the past few years. One of the most important cast parts of a locomotive

---

[1] Director of the Department of Engineering Research and Professor of Metallurgical Engineering, University of Michigan. Mem. A.S.M.E. Received his B.A. from Brown University in 1907 and his Sc.D. in 1925. After graduation he served a few years with the Jones & Laughlin Steel Company, Pittsburgh, Pa. He has been at Michigan since 1911, except for two years spent with the Ordnance Department during the World War.

Contributed by the Railroad Division and presented at the Semi-Annual Meeting, Birmingham, Ala., April 20 to 23, 1931, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

is its side frame. This is a large and a difficult casting to make. One of the large railroad companies reports that in 1923 castings of this type showed properties as follows:

Tensile strength, lb. per sq. in.	73,000
Yield point, lb. per sq. in.	40,000
Yield point as expressed as per cent of tensile strength	55.0
Elongation, per cent	29.0
Reduction of area, per cent	47.0

This company then raised these requirements as follows:

Minimum tensile strength, lb. per sq. in.	78,000
Minimum yield point, lb. per sq. in.	43,000
Minimum yield point expressed as per cent of tensile strength	58.0
Minimum elongation, per cent	22.0
Minimum reduction of area, per cent	43.0

Considerable objections were advanced by the manufacturers of steel castings to this new specification, but the company remained firm, and through cooperation with one or more foundries obtained material that met the specification by more careful practice.

In 1925 this company raised its standards still further, calling for properties as follows:

Minimum tensile strength, lb. per sq. in.	83,000
Minimum yield point, lb. per sq. in.	45,000
Minimum yield point expressed as per cent tensile strength	54.0
Minimum elongation, per cent	25.0
Minimum reduction of area, per cent	45.0

This requirement was successfully met by more careful foundry practice and by the use of alloying elements. To show concretely how these increased standards have been met, the comparative results of tests on two alloy steel frames are submitted; one of vanadium steel and the other of nickel steel.

COMPARATIVE RESULTS OF TWO ALLOY STEEL FRAMES

	Medium-carbon vanadium steel	Low-carbon nickel steel
Average tensile strength, lb. per sq. in.	87,590	79,472
Average yield point, lb. per sq. in.	49,442	48,495
Average yield point expressed as per cent tensile strength	56.5	61.0
Average elongation, per cent	28.5	30.4
Average reduction of area, per cent	53.6	55.8
Brinell hardness	150–165	140–150
Analyses:		
Carbon	0.36	0.17
Manganese	0.94	0.80
Phosphorus	0.015	0.014
Sulphur	0.028	0.028
Silicon	0.35	0.34
Nickel	..	2.70
Vanadium	0.19	..

The values in the four preceding tables are given to demonstrate that an improvement in the quality of the castings was produced, first through better foundry control, and then through the addition of alloys. It should be emphasized that nothing is to be gained through the addition of alloys with poor manufacturing practice, but that when the manufacturing practice is satisfactory, considerable benefit may be secured through the addition of alloys.

Another important casting is the truck frame. Failures of this part have been the cause of derailments. The arch bar truck has now been outlawed by the American Railway Association. This action, together with the adoption of new A.R.A. specifications which require 50 per cent more strength than former A.R.A. specifications, should go far toward eliminating failures of this part. These frames are, however, so far as the author is aware, for the most part of plain carbon steel. Possibly the addition of alloys might still further increase the strength of this casting.

Steel Mill. Much depends upon the steel mills for the production of quality material. Steel mills, however, should by no means be held entirely responsible for all metal failures, since design, forging, heat treating, and machine-shop practice are also important factors in the production of fabricated materials. Yet if steel is finished with pipes, seams, laps, inclusions, or has not been properly heated before it is subjected to the various rolling operations, it is apt to fail because of these conditions. In fact, subsequent forging, heat-treatment, or machining operations cannot eliminate these difficulties. Nor should one unduly criticize steel mills for furnishing materials of this type in the absence of adequate specifications or when the prices have been forced down through competitive bidding to a point which discourages rather than encourages careful control and minute regard for all details which lead to the production of metal of the highest quality. One detail of steel-mill practice of great importance in the production of sound steel is the assurance of proper discard of the ingot. This discard should in all cases involve both a top and a bottom discard. The top discard should be of sufficient amount to render the metal free from piping and undue segregation. In the case of top-poured ingots, the discard should probably be 25 per cent or more. In some cases it should amount to as much as 40 or 50 per cent. It should be noted that no metal has yet been produced which does not have a certain amount of segregation. Careful inspection of steel really requires a deep-etch test in order to ascertain if the metal is reasonably free from segregation.

Ingots when cast should be cooled slowly in order to free them, to as great an extent as possible, from internal strain. Conditions of cooling are not the same for all metals. Plain carbon steel can be cooled at a more rapid rate than certain alloy steels. In fact, some of the tender alloy steels should be most slowly cooled. Ingots, when placed in soaking pits preparatory to blooming, should remain in the pit long enough to permit a uniform temperature to be secured between the center and the outside of the ingot. Under no conditions should an ingot be rolled when the metal in the center is in a liquid state, since this would result in a bled ingot. The heating of billets previous to rolling should be carefully and intelligently performed. The metal should not be brought up to heat too rapidly, nor should it be brought to too high a heat.

The rolling operation also should receive careful consideration, as it is unwise to reduce sections too drastically—that is, by too few passes through the rolls.

All of these comments relate to both plain carbon and to alloy steels. Alloy steels, however, as a class, are more tender than plain carbon steels and require, in consequence, more careful handling and treatment. In procuring alloy steels, therefore, considerable weight should be given to the experience and reputation of the manufacturer.

Forge Shop. What has been said with respect to the heating and the rolling operations in the steel mill applies with equal force to the forging operations. Care must be taken to see that the metal is not brought up to the forging heat at an undue rate and that it is not heated above the proper forging temperature.

Further, in the forging operation, care must be taken not to work the metal much below its upper critical temperature. Again, alloy steels, since they are more delicate than plain carbon steels, should be handled with greater care.

Heat-Treatment Department. Various types of heat treatment have been recommended for some of the various classes of railroad steels. These suggested treatments have, in the main, consisted of a water quench, followed by either a low- or high-temperature draw; an oil quench, followed by either a low- or

high-temperature draw; a normalizing treatment, followed by a low- or high-temperature draw; or an annealing treatment. In the main, the most desirable heat treatment is to normalize and to subject the piece to a high-temperature draw. Normalizing consists of heating above the critical temperature and allowing the metal to cool in the air; drawing consists of reheating the material to some point below the critical temperature to remove internal strains. Such an operation does not, of course, bring out the maximum strength possible in the metal, though it does bring out maximum toughness. It is also less likely to produce checks and cracks in the metal and fits it to withstand shocks more successfully. This treatment is probably the best all-round one that can be recommended for railroad material of both the plain carbon and alloy steel grades. There is also a tendency in practice to lower the carbon somewhat so as to still further improve the toughness and ductility of the metal.

Machine Shop. Operations in the machine shop must be performed with care and precision. The author's attention has been directed to a number of failures which have occurred because of tool marks which apparently acted as the foci of failure. It seems hardly possible that these conditions could exist if, when the machining operations had been finished, the materials were subjected to adequate inspection.

Materials

The ferrous materials used for railroad equipment may be classified according to the use to which they are put. It is possible to place them in five general classes: structural steels for bridges and buildings, structural steel for cars and coaches, reinforcing steel for concrete, locomotive steel, and rail steel. It would of course be quite impossible to discuss the materials used for all of the various equipment parts which are purchased. This discussion has been limited to a consideration of a few of the important parts in rolling stock and rails.

Rolling Stock. The material of which a few of the principal parts of rolling stock are made are discussed at this time.

Side Frames: Locomotive side frames, which have been previously mentioned, are usually made of cast steel, either alloy or plain carbon. When alloys are used, they are generally of the carbon-vanadium or carbon-nickel type. The addition of these alloys makes it possible to retain strength, to maintain and, in fact, build up, ductility, and at the time same to permit the lowering of the carbon content. In recent years much attention has been given to these parts, and a very decided improvement has been made in their quality. Carbon-vanadium and plain carbon steels are also used in forged driving rods, crankpins, and similar parts.

Wheels: Car wheels originally were made of cast iron. This practice led to many failures accompanied by derailment as the size and weight of the equipment increased. For the most part, cast-iron wheels are not used on equipment when the weight is over 70 tons. Rolled steel wheels are generally used on all freight cars of over 70 tons, on all passenger cars, and on all locomotives and tenders, except for the driving wheels and the large-sized trailing wheels. Because of the gradual displacement of cast-iron wheels by steel wheels there has been a material decrease in derailments. The railroad engineers are therefore recognizing that, in general, failures due to wheels are no longer a source of serious importance.

Springs: Although many springs are still made of plain high-carbon steel, there is a trend toward the adoption of alloy steel. Many springs are now made of either silico-manganese or chrome-vanadium steel. No indications have come to the author's attention of any great difficulty experienced with springs, so that this matter has not been given prominence in this discussion.

Boiler Plate: The advisability of using nickel and silico-manganese steels for boiler plate is attracting considerable attention at the present time. This has been due to the fact that the designers of locomotive boilers are going to higher operating pressures and temperatures. These designers desire to maintain the same factor of safety as formerly without increasing weight. They are therefore giving consideration to alloy steels of the type mentioned. Already, the Canadian Pacific Railroad has used boiler plates of nickel steel with very satisfactory results. This steel has a further advantage in that it is not apparently subject to aging or recrystallization at operating temperatures. It must be recognized that though, when properly made, it is an improvement over the plain carbon steel, yet it is not as simple a steel to make as a plain carbon steel and requires considerable care in its manufacture and in the inspection of the finished material.

Tubes: The boiler and superheater tubes in locomotives are, for the most part, of plain carbon steel. The author sees no occasion at this particular writing for considering changes in the composition, particularly for boiler tubes, although it is claimed by some railway systems that tubes of nickel steel undergo less deterioration from corrosion. With the increase in operating temperatures and pressures, it may be desirable to resort to some type of alloy which has a greater stability at high temperatures than plain carbon steel. Several alloys of this type are now in process of development, though, so far as the author is aware, there has been no general test, or at least no general adoption.

Axles: The one part of the moving equipment which is still giving considerable trouble is the axles. A large portion of the axle failures result from overheating at the journals. Efforts are being concentrated upon improvements in the quality of lubricating materials, upon shop practices, and upon closer inspection. Some of these failures are attributed to tool marks; others are said to result from a too rapid cooling of an overheated journal which is responsible for checks or cracks. These in time develop into fractures and failures. Many of these axles are of plain carbon steel; others are of alloy steel, frequently of the carbon-chrome type. Some engineers have suggested raising the carbon content of the axles in order to increase their strength. This, however, is not the remedy for failures resulting from overheating of the journals, since higher carbon will decrease the ductility and hence increase the chances of cracking. There is a tendency, therefore, in the production of axles to reduce the carbon and build up the desired strength by means of alloys. Likewise, the heat treatment given to these axles is usually of the normalizing type, followed by a high draw. These changes—the lowering of the carbon content and the alloy additions which do not harden, but toughen, the steel—combined with the type of heat treatment mentioned, produce a steel which is as little affected by service conditions as lies within the power of the metallurgist to produce.

Rails. Rails are, in the main, of three types: plain carbon steel, a semi-alloy steel, and a straight alloy steel. By far the largest tonnage of rails are of plain high carbon. A semi-alloy steel containing from 1.25 to 1.75 per cent of manganese is finding considerable favor in practice. Other alloy steels have been tried out experimentally, although too recently to permit any conclusions regarding their comparative value. A straight alloy steel frequently used by the railroads for frogs and crossings is the manganese steel carrying from 11 to 14 per cent of manganese.

Because of disasters which have occurred due to rail failures, the quality of rails has been continuously and consistently investigated for many years. The one thing most feared in rails is transverse fissures. These fissures vary in size from

sections of small area to sections up to 4 square inches. They are of infinitesimal thickness. As the name implies, they are at right-angles to the length of the rail. Many theories have been advanced explaining the cause, but as yet no one of these theories has gained general acceptance. The railroads have spent hundreds of thousands of dollars endeavoring to develop an inspection method that would detect fissures before rail failures occur. They met with little, if any, success until the problem was given to the late Mr. Elmer A. Sperry, who worked out an electrical resistance method for the detection of these flaws. This method is now being used by a number of systems and represents a very great advance over all previous methods. It was almost, if not quite, the last great piece of research which Mr. Sperry brought to a successful culmination.

Special Purposes. There are a number of places in which alloy steels are used for definite and specific purposes. For instance, the bearings of the generator developing electricity for the locomotive headlights, etc., are alloy steels, some of which are of the carbon-chrome type and others of the carbon-chrome-molybdenum type. The roller bearings on locomotives, tenders, and freight and passenger cars when used are mainly of alloy steel.

Of course, in railroad equipment some parts, particularly freight cars operating in an industrial section, are subject to corrosion. The plate for these cars frequently carries a small percentage of copper, as this type of material is somewhat more resistant to corrosive conditions than plain carbon steel. Stainless steel is too expensive a type for this use, although it is finding some favor today in the fittings of passenger and Pullman cars, particularly the latter.

Distribution of Material. Although in this paper the subject of alloy steels has been extensively discussed, plain carbon steels and ordinary cast and wrought irons still represent the bulk of the material that goes into a locomotive. This fact is well brought out in the table herewith, which gives a material distribution on the basis of weight of one type of modern passenger locomotive.

DISTRIBUTION OF MATERIALS IN A LOCOMOTIVE

(Light weight of engine and tender is approximately 455,200 lb., of which 19,355 lb. represents important specialties such as air pumps, booster, feedwater heater, and stoker. This leaves a balance of 435,945 lb. for consideration.)

	Weight, lb.	Per cent light weight, engine and tender (435,945 lb.)
Alloy steels. Normalized carbon vanadium. Main rods, side rods, trailing-truck equalizers, front and rear main engine transverse equalizers. The equalizers form a part of the spring rigging	4,800	1
Iron, wrought. The principal items consist of wrought iron used for staybolts, engine bolts, piping, drawbars between engine and tender, drawbar pins, safety chains, and engine truck equalizers, the latter being a part of the truck spring rigging	26,300	6
Iron, cast, malleable, and gun. About the only item of importance in cast iron consists of the superheater header. Gun iron is used for cylinder and valve chamber bushings, which, although of considerable importance, represent only a small percentage of the total weight of the locomotive. The use of malleable iron is confined to certain miscellaneous details	4,300	1
Miscellaneous. Bronze, copper, wood, glass, brick, asbestos, etc.	9,500	2
Plain carbon steel, wrought and cast irons. This group covers the bulk of the locomotive and represents such items as boiler and tank steel, main engine frames and cylinders, tender frames, engine and tender trucks, brake and spring rigging parts, valve motion, wheels, axles, pilot, etc.	391,100	90
Totals	436,000	100

Specifications

All of the major railroad companies either have their own specifications or else use those of the American Railway Association or those of the American Society for Testing Materials. In fact, the specifications used by the railroads draw very largely, if not entirely, upon these two latter sources. These specifications have done much toward building up and improving the quality of materials used by our railroads. It is interesting to note that in some micrographic tests are required in order to assure the purchaser of suitable grain refinement.

The specification of one of the large systems for cast-steel truck side frames is likewise most interesting. This specification is unusually thorough and complete and has doubtless gone far toward the great improvement in quality which has taken place in the last few years in this particular part.

Acknowledgments

The author wishes to express his appreciation to Miss M. A. Brook and to Mr. R. Schneidewind for their assistance in collecting the material, and to Mr. Schneidewind for his aid in the preparation of this paper; also to Mr. W. H. Sellew for his kind suggestions and review.

Bibliography

Barr, W. M., "Alloy Steels in Locomotive Construction," *Ry. Mech. Engr.*, vol. 103, pp. 739–742 (1929).

Colvin, F. H., "New Alloys in Railway Rolling Stock," *Am. Mach.*, vol. 71, pp. 449–450 (1929).

Greene, O. V., "Locomotive Forging Steels," Trans. A.S.S.T., vol. 13, pp. 573–588 and disc. 588–603 (1928).

Griffiths, F. J., "Alloy Steel in the Railroads," Proc. Ry. Club, Pittsburgh, vol. 26, pp. 94–98 (1927).

Humpton, W. G., Huston, F. P., and McKay, R. J., "Nickel-Clad Steel Plate Work," Trans. A.I.M.E., Metals Divn. (Feb., 1931).

Johnson, W. A., "Use of Alloyed Steels in Construction of Locomotives," Proc. Inst. Mech. Engrs., pp. 1087–1097 (1929).

Langenberg, F. C., and McKnight, C., "Alloy Bearings Applied to Trains," *Iron Age*, vol. 121, pp. 130–131 (1928).

MacQuigg, C. E., "Alloy Steels," *Mining & Met.*, vol. 11, pp. 578–580 (1930).

McKnight, C., "Recent Developments in the Use of Nickel Steels," Trans. A.S.M.E., paper IS-50-6 (1927–1928).

McKnight, C., "Alloy Steels for Boiler Construction," Trans. A.S.S.T., vol. 13, pp. 638–657 (1928).

McKnight, C., "Alloy Steels in the Railroad Field," Trans. A.S.M.E., paper RR-52-7 (1930).

Newmann, W. A., and Pascoe, C. F., "Making Locomotives Stronger," *Iron Age*, vol. 119, pp. 701–704 (1927).

Pohl, A., "Pearlitic Manganese Steel as Material for Rails," *Stahl und Eisen*, vol. 50, pp. 1574–1580 (1930).

Stumper, R., "Research on the Influence of Rolling Temperature on Properties of Rails," *Stahl und Eisen*, vol. 49, pp. 177–187 (1929).

Thum, E. E., "Medium-Manganese Rail a Success," *Iron Age*, vol. 123, pp. 940–942 (1929).

West, A. K., "Locomotive Equipped With Roller Bearings," *Blast Furnace and Steel Plant*, vol. 18, pp. 1147–1148 (1930).

Report of Committee 4, Rail Appendix C, "Transverse Fissure Statistics," Proc. A.R.E.A., vol. 29, pp. 573–579 (1928).

Report of Committee 4, Rail Appendix B, "Rail Failure Statistics for 1926," Proc. A.R.E.A., vol. 29, pp. 562–573 (1928).

Report of Committee 4, Rail Appendix D, "Cause and Prevention of Rail Battering," Proc. A.R.E.A., vol. 29, pp. 580–591 (1928).

Report of Committee on Rail-Mill Practice, "Operating Results of A.R.A. Rail-Fissure Detector Car," "Rail-Fissure Statistics," "Tests of Alloy and Heat-Treated Carbon-Steel Rails," Proc. A.R.E.A., vol. 32, pp. 345–398 (1931).

TEX-53-1

Control of Picker-Room Humidity

By NORMAN E. ELSAS,[1] ATLANTA, GA.

This article attempts to deal with the subject in a non-technical manner. The control of the amount of moisture in cotton laps delivered to the card room is highly desirable. Cotton manufacturers desire uniform weights for any given unit of length from the ounces per yard of picker laps to the grains per hank in the final yarn. If laps delivered to the card room contain widely varying percentages of moisture, though a lap scale may indicate identical gross weights, a variation in weight per yard is immediately begun at the cards which must be corrected by gear changes in subsequent operations. By holding picker laps to very close limits as to the net weight of cotton contained for the same number of yards of lap produced, the beginning of evenness or the elimination of variation in subsequent operations is attained.

THE matter of humidity control and "regain" in cotton has been given much more thought and study during the past few years than in the many centuries since the beginning of the cotton-textile industry. At any rate, the thought has been of a constructive nature, rather than merely attempting to get along with the least amount of equipment necessary to aid nature.

The term "regain" as it will be used hereafter has reference to the percentage of moisture in cotton on a basis of weight to a bone-dry condition. While it is possible to work out a quite definite relation between relative humidity and per cent "regain," from a practical standpoint the variation in the nature of the raw stock itself will have as much or more bearing on the final "regain" in stock leaving a picker room as does the variation from a true parallel between relative humidity and this "regain."

The United States Department of Agriculture has set forth the fact that it considers the cotton fiber in its normal condition when it contains $8^1/_2$ per cent by weight of moisture from the bone-dry condition. Cotton manufacturers have found that the friction in drawing one fiber over another causes sufficient heat to decrease the moisture content to such an extent that humidifying apparatus is necessary to prevent this "evaporation." Cotton must have a certain amount of moisture present in the fiber in order to prevent brittleness.

Throughout the Southern States the meteorological bureaus have indicated a variation of relative humidity that would indicate a "regain" as low as $1^1/_2$ per cent to as high as 13 per cent.

With this variation before us, it is evident that there can be a variation in the weight of any given amount of cotton by as much as 10 per cent on account of the moisture content alone. These extremes of atmospheric condition will often take place within a space of 48 hours.

It seems rather peculiar how few manufacturers have paid a great deal of attention to this feature, in view of the usual care taken to see that picker laps weigh exactly alike. (A finisher picker is set up to operate for an exact number of yards, and then the flow cotton is cut off; assuming the same weight per yard, all laps should weigh the same.) In the event that there is a widely different amount of moisture contained in the cotton itself, does it not seem rather futile that the mill men have attempted to hold lap weights within a variation of less than a quarter of a pound from a standard?

With the thought in mind that this particular condition could be greatly improved upon, study was given to some manner of controlling the atmospheric conditions in the picker room.

One of the first ideas that occurred was to determine upon some means of recirculating the air that was exhausted to the dust pit, after relieving this same air of the dirt and lint. This appeared to be rather simple at first, but the opinions of the many different companies that were approached with the definite proposition for accomplishing this were all rather dubious and indicated an expense far beyond any practical amount.

The next idea was to so condition the air introduced into a picker room that it would be unnecessary to recirculate the air discharged through the picker fans. This second thought was taken up also with various other equipment manufacturers, and this cost was also found to be quite excessive.

For the information of those not familiar with the amount of air normally handled by an individual picker it should be stated that each picker fan handles a minimum of 1200 to 1800 cu. ft. of air per minute. The normal picker room for many years has contained three picking operations, Southern mill operations requiring approximately 12 finisher pickers to take care of 50,000 spindles. In addition to the finisher pickers there were as a rule 12 intermediate pickers and approximately 8 or 10 breaker pickers. The total number of fans operating, therefore, exhausting air from the picker room, would be approximately 32 to 34, moving over 40,000 cu. ft. of air per minute.

Several attempts were made to control the weight of laps indicated by the lap scale, to make due allowance for the relative humidity changes in the atmosphere. Several scales were drawn up, so that an indicating hand could be set on the lap scale to allow for "regain" as indicated from a relative-humidity scale. There also was brought out an accurate type of indicator that will read "regain" directly for the indicator on the lap scale. Devices of this kind must necessarily depend on human supervision.

The importance of accurate moisture-content regulation in laps going to the card room warrants close control. Certainly these devices are better than none at all, and quite satisfactory results can be obtained where consistently followed up by the management.

Before leaving the matter of devices that will aid in the maintenance of even numbers through the spinning, it is well to mention "evener drawing." One of the best mills in this country does not attempt to weigh laps except once a day in order to check on the different machines to see that they are in reasonable adjustment. They then make no attempt to hold the sliver from the cards to a definite number of grains per yard. They do use two processes of drawing, however, the first drawing being

---

[1] Secretary, Fulton Bag and Cotton Mills. Jun. A.S.M.E. The author was born in August, 1896, in Atlanta, Ga., and attended local public schools and later military schools locally and in Massachusetts. He was graduated from Phillips Academy, Andover, Mass., in 1914, and Cornell University with M.E. (E.E. certificate) in 1918. He was in the Naval Aviation Corps during the war, and entered the employ of Fulton Bag and Cotton Mills in 1919, being elected director and secretary on Jan. 9, 1922.

Contributed by the Textile Division and presented at the Semi-Annual Meeting, Birmingham, Ala., April 20 to 23, 1931, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

of the ordinary type, but the second drawing being "evener drawing." Of course by the time the cotton fiber has reached this second process of drawing, the amount of moisture contained is quite uniform and in accordance with the controls operating throughout the card room. Therefore the "evener drawing" can be made to operate quite satisfactorily to eliminate any unevenness in the weight per yard of the various slivers being fed to it.

Evener drawing is quite expensive, however, and certainly too expensive for a mill already in operation to install at the expense of throwing out otherwise satisfactory drawing frames. In addition, "evener drawing" requires considerably more attention on the part of the section men and fixers and a higher cost of upkeep in operation.

The first really practical method of controlling the amount of moisture in picker laps resulted from the development in recirculating the air necessary for picker operation, within the picker itself.

The idea which started the investigation of recirculating the air in a picker began with accomplishing this in a soft-waste machine. In one mill it became necessary to move a soft-waste machine to a new location, and the suggestion was made to put it on one of the upper floors of the mill where it could be in proximity to the point producing the soft waste. It then seemed difficult to connect this machine in its proposed location with the dust pit. Since all the stock to be fed through the machine is free from dust and trash, it seemed possible to recirculate the air within the machine. It was then done by mounting the machine on skids and flooring with sheet metal. In addition, since soft-waste machines are a continual source of fire, which spreads throughout the dust pit, this trouble was eliminated.

This was the basic idea on which further developments were carried out.

For the past 10 years tremendously rapid strides have been made in the opening processes, in an effort to eliminate as much of the undesirable fiber and foreign matter as is possible before the stock is fed to picking equipment for making into a lap. Most remarkable results have been obtained in this field, it being possible now to deliver stock that is practically free from all dust, which would often form a veritable cloud in many card rooms.

With the ability to eliminate dust and the heavier types of dirt in the opening processes, it became evident that the air being discharged by the fans in the final picking operations was relatively clean. In fact it is quite feasible for one to breathe comfortably, the air being discharged from a picker where the opening processes are efficient.

It was necessary to experiment to a great extent to develop the proper type of baffles and air passages to insure performance in a picker set-up to recirculate its own air, equal to that obtained from a picker discharging normally into the dust pit. This has been finally accomplished, and continual watching and testing have shown the laps produced on recirculating pickers to be indistinguishable from those produced on the pickers operating in the usual way in the same normal atmosphere.

After equipping a complete picker room with the recirculating device, it was then only necessary to install sufficient humidity and proper automatic control to maintain an approximately constant "regain" in the cotton. Satisfactory results were noted immediately, and they have been continually obtained over a period of more than two years. The necessity for continually changing the draft gearing on drawing and subsequent operations was immediately cut to about 10 per cent in number and magnitude.

Most cotton mills that are carefully run have followed the practice of sizing drawing at least three times a day, and frequently as many as five or six times a day, and making adjustment in the drafting to compensate for the varying weights in the drawing produced. Naturally this showed quite a wide variation, as the laps from the picker room contained varying amounts of moisture according to atmospheric conditions. With control of humidity in the picker room, it is now unnecessary to change the gearing more than two or three times a week, and in each case the changes are of a very slight nature.

The necessity for continual changes of gearing brought about some additional variation in the sliver from that which would normally be brought about on account of varying "regain" in the stock itself. Cutting out the necessity for these changes has eliminated further variations in the weight of the sliver and subsequent rovings.

Naturally the advantages obtained in evenness throughout the card room are transmitted on through the spinning to the weaving. It would be most interesting if a change of this kind in controlling picker-room humidity could be made in a mill and the subsequent benefits checked directly without other mechanical improvements being made during the same space of time. Unfortunately, the author has no way to allocate the benefits derived solely from humidity control in picker rooms, since new equipment was in process of installation in various departments at the same time the development was being accomplished in the picker room.

There is no question of the value of the improvement itself, since the benefits are immediately indicated in the evenness of the lap weights, the evenness of the card sliver, and rovings, during consistent weighing throughout the plant and at widely varying seasons.

Another most important direct saving from the use of recirculating pickers is the elimination of enormous demands for heating steam during cold seasons. The heating of thousands of cubic feet of air per minute in freezing weather is no little expense. This heating also adds a variation factor to the moisture content of the cotton which must be corrected in the gearing changes later. A feature of controlling the humidity in the picking and opening departments that it might be well to mention is the necessity in most mills to transport cotton by means of a condenser from opening equipment or from bins to the final distributing device over the pickers. This cotton has already been thoroughly cleaned in the opening processes, and the atmosphere in the opening room or the bins may be held to a constant relative humidity. Naturally to draw the air from the bin room or the opening room and to discharge it into the dust pit through a condenser located in the picker room itself would constitute a loss in humidity that would have to be replenished by excessive humidifying capacity in the bin or opener room. To prevent this it was found that exhaust from the condenser fan could be returned to the room from which the condenser system is fed, by enclosing the mouth of this discharge pipe in a long burlap bag. This burlap bag acts exactly in the same manner as the ordinary vacuum cleaner bag familiar to every housewife. The efficiency of the burlap for this particular work is practically 100 per cent, and the air escaping is free from lint and dust. Of course, the bags must be cleaned periodically, but once a week is sufficient.

All of this is simple enough where the cotton contains normal or less than normal moisture.

There is one further step in controlling the percentage of moisture in cotton that is rightfully a part of this discussion, but experiments have not been carried sufficiently far to indicate that a definite recommendation toward their adoption would be in order. As originally stated, raw cotton fiber contain varying amounts of moisture but $8^1/_2$ per cent by weight has been accepted as the standard by the United States Government. Unfortunately, the cotton does not know this, and from

actual tests bales may be delivered to a mill with a percentage of moisture varying from as low as 3 per cent to as high as 18 or 20 per cent, without it being readily detected by "feel."

From the fact that the moisture content of the raw cotton can easily vary 15 per cent from bale to bale, it is clear that some recognition should be given this by the cotton manufacturers so as to bring all stock to the same condition of "regain" prior to the picking operation. It is evident that, if cotton contains more than the proper percentage of moisture, this excess cannot be removed merely in the picking operation, where there is a controlled percentage of humidity approaching $7^1/_2$ per cent "regain" for the cotton.

Cotton fiber absorbs moisture most readily, but the rate at which it gives off this moisture is very much lower under normal atmospheric conditions, mainly because the "terminal differences" are not so great.

Many mills provide bins in which cotton is allowed to age after passing through the opening and mixing processes, before being fed to the pickers. Where there is considerable variation in the moisture content of different bales, excellent results can be obtained by having the humidity in the bin room controlled to give a $7^1/_2$ per cent "regain" and by having fans mounted on the bin walls so as to cause a constant circulation of this controlled air over the stock in the bins. The circulation has little effect on cotton at the bottom of a pile in a bin, but it is exposed a sufficient length of time to the controlled atmosphere while the bin is being filled and emptied.

When cotton contains upward of 10 per cent of moisture, there is a strong possibility that some inexpensive method of drying this excess moisture would be of material assistance before the picking operation. Rather crude experiments have been carried on in direct manner to accomplish this where cotton is conveyed over a reasonable distance to a condenser. The method pursued is to pass the air feeding into this conveyer pipe over hot coils that immediately brought the "regain" of air in the pipe down well below 5 per cent, regardless of outside conditions. Naturally, this air being quite warm and in intimate contact with the loosened fibers during the conveying process, a good deal of moisture is removed from the stock. In addition, this hot blast of air must pass through the cotton deposited on the condenser screen before it can be released to the dust pit, so that some additional elimination of excess moisture takes place at this point.

There are probably several simpler methods by which raw cotton, when once opened, can be so treated that its percentage of moisture can be brought down slightly below normal, and when this development is made, with controlled humidity conditions in the opener room, in the storage bins, and in the picker room, certainly great improvement in the regularity of manufacturing cotton into yarn can be expected.

Discussion

R. S. Curley.[2] The control of humidity in cotton prior to and during the picking operation has not been given the attention which its importance warrants. While some attempts have been made to humidify picker rooms where 20 to 30 fans were each drawing out 1500 or more cubic feet of air per minute, the expense entailed and the somewhat negative results obtained have not encouraged any general adoption of this practice. Some experiments have been made to determine the amount of variation in the moisture content of finished laps. Many attempts have been made to vary the standard weight of the finished lap in accordance with variation in relative humidity in the picker room. None of these experiments have given any practical answer, because they did nothing to overcome the evil at its source.

The system described in this paper gives a definite and effective means of control of the humidity from the very beginning. The opening room can be easily humidified to the same degree as the picker room, because the air drawn by the fans in transporting the cotton from one operation to the next is returned to the room of its origin. In this way cotton is subjected while in a finely divided state to enormous quantities of air passing through it during every operation of picking and opening. Cotton in contact with rapidly moving air takes in or gives up moisture very much more rapidly then when simply exposed to still air. A recognition of the value of condensers for removing free dust and for quickly balancing the moisture content in the cotton is apparently one of the foundations on which this system was formulated.

Taking air at 70 deg. fahr. and 65 per cent relative humidity, each cubic foot contains 5.25 grains of moisture. Each fan in a picker moves about 1500 cu. ft. of air per minute. The average production of the modern machine is about 360 lb. per hour, or 6 lb. per minute. That is, 250 cu. ft. of air passes through each pound of cotton going over the screens of the picker. The amount of moisture necessary for $7^1/_2$ per cent regain in cotton is 525 grains per pound. The 250 cu. ft. of air conditioned as described contains 1312.5 grains of moisture, or $2^1/_2$ times the moisture required to bring the cotton from dry to $7^1/_2$ per cent regain. It is seen from the foregoing that a proper condition of the air drawn through the screens has a great influence in maintaining the desired regain in the picker lap.

Another important feature is the control of cleaning. Where cotton is held at $7^1/_2$ per cent regain, its freedom from static during the beating operation permits the elimination of quantities of light particles which under dry conditions would tend to adhere to the fibers. The fibers are not so brittle in themselves and tend to lie straighter under conditions of proper regain, so that there is less possibility of breakage from beating.

Perhaps one of the most important results obtained is the maintenance of nearly constant density during the passage under the evener roll. It is well known that an evener weighs by measuring the thickness of the cotton as it passes between the plates and the roll. It is also known that the evener can only be accurate as long as the density remains constant. It is one of the unfortunate qualities of cotton that its density varies widely with changes in moisture content. A thorough exposure of cotton to large quantities of air with proper moisture content permits delivery of the cotton to the evener at a constant density which gives the evener a chance to function at its highest possible efficiency.

It is in this particular that this system is vastly superior to that of changing the standard weight in accordance with changes in relative humidity of the surrounding air. This latter arrangement leaves the evener operating under conditions of constantly changing density for which it is unable to compensate.

The author states that the humidifying of the bins is only effective during the filling and emptying of the bins. This statement is quite true and should be further emphasized. Cotton in a loose state is an excellent insulator, and air does not ordinarily penetrate far beneath its surface. Because of this fact, cotton does not change its moisture content to any extent while in the middle of a bin. It will change on the surface and on the floor, particularly on a cement floor. It is therefore evident that the benefits of aging cotton in the bins were largely imaginary. The only real benefit came from blending many bales through the stack of loose cotton. Where mixing feeders are arranged to obtain this blending, much better humidity control is possible through the use of extra condensers and fans, which pass the

[2] Saco-Lowell Shops, Newton Upper Falls, Mass.

moist air through the cotton in such large quantities as to insure constant regain.

In view of the advantages gained through this system of humidity control, it is certain that many mills can make definite advance in quality of product by its adoption.

The pioneer work already done must direct attention to this opportunity for sound progress.

C. L. Emerson.[3] The author has described an improvement in cotton-mill operation for which engineers have been searching for some time. Heretofore variations in moisture content of picker laps have introduced irregularities in sliver weight at the very beginning of the process, and subsequent operations were forced to even out the inequalities as best they could.

The system is another and very important contribution to the general progress being made in obtaining uniformity so valuable in manufacturing conditions. It is also interesting to note that this system of picker-room humidification would hardly have been practicable until the improvement in opener-room equipment had reached its present state of efficiency. If the cotton had been fed to the pickers in a condition which 10 years ago would have been considered satisfactory, it is quite probable that the air passages in the recirculating pickers would have been clogged with the dirt and leaf removed by the machine, but which under present conditions is extracted in the opener room before it ever goes to the pickers.

In an effort to supply some humidification to picker laps during particularly dry periods, other mills have previously installed one or two atomizer heads to spray directly on the finished picker. These heads were usually hand controlled and were at best only a makeshift device to provide a certain rough compensation.

The bins for cotton storage between the opening and picking processes which are proposed by the author to permit a considerable amount of the humidification being accomplished in the opener room itself also have certain subsidiary advantages and are frequently used by other mills which do not attempt to humidify their picker rooms. The argument for their use is to permit the aeration of the fibers and also to let them uncurl from their forced compression in the bale. The system here discussed would add another incentive for the use of bins.

The saving in the steam required to heat the picker room in winter is quite an appreciable one. Some mills, in order to save part of this heating expense, allow the picker and opener rooms to remain at a temperature much lower than other parts of the mill. This is not at all conducive to efficiency on the part of the opener-room and picker-room operatives.

H. S. Busby.[4] The thing of outstanding interest about this paper is the method of attack of the problem, which is a purely analytical and engineering one, rather than that of rule-of-thumb procedure so common in textile manufacturing.

[3] Robert & Co., Inc., Atlanta, Ga.

[4] Director, A. French Textile School, Georgia School of Technology, Atlanta, Ga.

The conclusions reached in the paper would seem to indicate the advisability of supplying cotton for shipment wrapped in a moisture-proof container or one as nearly so as possible and practical.

This may bring about a reconsideration of the present method of baling cotton, and suggests a close-mesh cotton-fabric container, if it can be shown that such a wrapper will retain moisture within the package for a considerable period of time.

It also suggests the advisability of the cotton manufacturer purchasing fiber so processed in the field as to secure dependable uniformity in moisture content. An important development has taken place in this field within the past year, and economy in shipping, in manufacturing, and in appraisal value is closely tied to this problem.

It is to be regretted that a more general check is not available of the method of recirculation described in the paper. It seems that if the mills having different types of opening processes were to report results based upon this or similar procedure, most of the information regarding practical ranges of recirculation, cleaning, and net yield could be reported dependably within a reasonably short period of time. There is no doubt that this would cause basic redesign of textile preparation equipment.

It is significant that most of the recent developments in preparation machinery, with the exception of the Lord card, have contributed little, except revision, refinements, and details, for a number of years. The huge tonnage of machinery necessary for the production of a single thread, with its consequent investment burden, suggests a redesign along such lines as indicated by this and similar experiments to be not only of fruitful outlook, but an emergency measure.

Author's Closure

Mr. Curley has brought out a pertinent point not touched upon previously. There is certainly a wide variation in the operation of the evener on a picker due to the moisture content of the stock passing through it. While Mr. Curley does not give any specific figures, nor has the author any available, it has been the experience of a number of mill men that a picker running properly on dry cotton and turning out the proper weight laps will, when switched to stock having a higher moisture content, immediately begin delivering laps that are considerably overweight. This excess of weight is not due solely to the moisture contained in the stock, but, as pointed out by Mr. Curley, is due to the increased density of the stock permitting more actual pounds of cotton to pass through the eveners per revolution.

The author regrets that there were no discussions disagreeing with the proposal of the original paper, since there are undoubtedly a number of able textile manufacturers who do not believe in this proposed method of controlling picker-room humidity, though it is in actual operation at the present. It is only by disagreement and discussion that new features can be brought to light, and the author wishes to take this opportunity to invite correspondence on this matter.

Development in Gaseous Fuels and a New Singeing Application

By R. S. VAN NOTE,[1] BALTIMORE, MD.

Liquefied petroleum gases are finding an application in various process-heating work. In the textile industry butane is available as fuel for singeing, calendering, gas tenter-frame heating, and yarn-singeing operations. New cloth-singeing equipment has recently been developed and is discussed in the paper.



THE customary usage of manufactured gas, gasoline gas, natural gas, and oil as fuels for cloth singeing, calendering, yarn singeing, and tentering operations is too familiar to call for extended discussion here. Their use throughout the industry is general, and knowledge as to their characteristics and methods of application is well known. A brief résumé of the use of these familiar fuels is desirable, however, and recalls that: (1) Manufactured gas, gasoline gas, and natural gas are suitable fuels for open-flame singeing, rotary- and stationary-plate singeing, calender and tenter-frame heating, and yarn-singeing operations; (2) oil is frequently employed in heating rotary- and stationary-plate singers.

The commercial production of the liquefied petroleum gases made available another source of fuel for the operations mentioned. Handled as a gas at the burner, with flame characteristics comparable to those of natural gas, liquefied petroleum gases are, from the application standpoint, capable of performing satisfactorily. The scope of their usefulness centers about the economics involved; in short, it is a question of cost per 1,000,000 B.t.u. of one fuel versus another.

The more important commercial liquefied petroleum gases are propane and butane. These are obtained by "stripping" so-called "wet" natural gas or from by-product still gas as produced in oil-refinery practice. They exist as gases under normal atmospheric pressures and temperatures, but are capable of being stored and transported under pressure in the liquid state.

Propane, in addition to its industrial uses, has found a wide application as a domestic fuel for cooking and water heating in areas removed from manufactured-gas service. Butane, in addition to its industrial application, serves as a source of town gas supply through the medium of the butane-air gas carburetion process, and as an enriching medium for gases of low thermal value.

A partial list of the physical properties of commercial propane and butane (a mixture of normal and iso-butane), as given by one distributor, follows:

	Propane gas	Butane gas
Normal state at 60 deg. fahr., 30 in. mercury		
Specific gravity of liquid	0.51	0.57
Specific gravity of gas	1.52	2.07
Latent heat of vaporization, B.t.u. per gallon	778.2	775.7
B.t.u. per gallon liquid	91.370	101.800
Pounds per gallon at 60 deg. fahr.	4.25	4.78
Cubic feet per gallon	36.51	31.27
B.t.u. per cu. ft. of gas	2500	3260
Vapor pressure of liquid, lb. per sq. in. (gage):		
10 deg. fahr.	32	1
32 deg. fahr.	58	8
60 deg. fahr.	98	24
105 deg. fahr.	192	62
Cubic feet of air required for complete combustion of 1 cu. ft. of gas	23.91	31.1

The utilization of butane or of propane for textile process-heating operations is essentially the same; and for the sake of brevity, discussion will be confined to butane.

Shipped by rail in tank cars of special construction, butane is transferred to storage tanks on the manufacturers' premises. Customarily, storage capacity equivalent to four to six weeks' supply is furnished. From the storage tank butane liquid is conveyed by storage-tank pressure to the point of vaporization, usually a small building or shed adjacent to the main plant buildings. Here it passes through straining, governing, and vaporizing equipment. The vaporizer, or gasifier, is nothing more than a heat exchanger, the heat of vaporization being supplied by hot water or low-pressure steam. In passing through the vaporizer, the butane liquid, with a low latent heat of vaporization, is readily converted to the gaseous state, the amount of vaporization being automatically controlled to correlate the supply to the demand.

At the outlet of the vapor governor, adjacent to the vaporizer, the fuel is completely gasified, containing 3260 B.t.u. per cu. ft. From this point on utilization practice differs somewhat. The gas may be diluted to 525 B.t.u. per cu. ft., the average heating value of city gas; it may be diluted to a completely combustible mixture of 96 B.t.u. per cu. ft.; or it may be carried as a straight 3200-B.t.u. per cu. ft. butane gas, undiluted. The dilution medium is air.

When carried as a 525-B.t.u. gas or as a completely combustible mixture, an automatic proportioning and compressing machine is placed in the line next to the vapor governor. It is supplied with the rich gas undiluted, and discharges, in the one case, a partially diluted 525-B.t.u. gas, and in the other a completely combustible 96-B.t.u. gas. The maximum discharge pressure is 2 lb. per sq. in.

Since complete combustion requires 31 parts of air to 1 of gas, in the former case the remainder of the air is supplied at the point of gas utilization; in the latter, being totally premixed at the machine, no further air is supplied. When totally premixed in this manner and carried through the pipe lines as a mixture approved by the Underwriters' Laboratories, protective devices should be provided to insure against burning back through the line. These protection devices are available.

If conveyed as the straight 3200-B.t.u. gas, leaving the vapor governor at approximately $^1/_2$ lb. pressure, the fuel receives its air supply from air blowers, the air line being tied into the gas line at a point adjacent to the burners.

---

[1] Sales Engineer, C. M. Kemp Manufacturing Co. Mr. Van Note was previously connected with the Public Service Electric and Gas Company, Newark, N. J., in the capacity of industrial-gas representative and assistant to the general industrial-fuel representative. He is a graduate of Rensselaer Polytechnic Institute.

Presented at the meeting of the Textile Division, Providence, R. I., Oct. 6, 1931, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

The basis for recommending which of the three methods outlined should be used depends on the process to which the gas is to be applied and existing plant conditions. If a plant has been using manufactured gas and changes to butane, it is customary to use a 525-B.t.u. butane gas. This is because manufactured city gas has approximately the same thermal value, and existing pipe lines and burner equipment can be utilized without major change. Where the fuel consumption is large or processing requires burner equipment adapted to a complete butane-air mixture, the 96-B.t.u. gas proves advantageous. This method is highly efficient thermally and requires but one pipe line. The use of the undiluted or 3200-B.t.u. gas at the burner requires equipment capable of proportioning a large quantity of air with a relatively small quantity of gas, maintaining these proportions under various rates of flow.

The prevailing cost of other fuels in the particular plant locality as balanced against butane costs, taking into consideration all charges involved in either case, is the predominating factor in determining its use. It goes without saying that in isolated regions the choice of a fuel is narrowed down by the inaccessibility to public-utility manufactured or natural-gas lines. This condition exists in many textile plants, and where fuel consumption is large enough to justify the installation, a butane plant may prove desirable.



FIG. 1 BUTANE TANK CAR AND CUSTOMER'S 13,500-GALLON STORAGE TANK

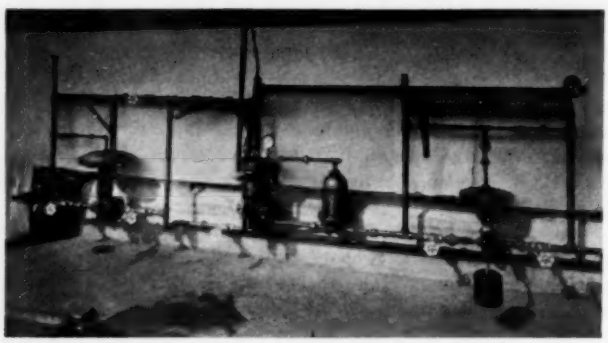


FIG. 2 BUTANE PRESSURE-GOVERNING AND VAPORIZING EQUIPMENT

Butane, as the source of town gas supply, is growing rapidly. The low investment required to build a butane plant makes it economically possible to render gas service in the smaller communities where the higher cost of water or coal-gas plants has hitherto hindered such developments. In certain cases, where obsolete coal or water-gas plants have been in operation, they have been replaced by butane plants.

The single-industry town exemplified by the typical mill village where an industrial load exists can serve both industrial and domestic purposes, the one aiding the other in making the gas-plant installation economically justified.



FIG. 3 FLAME-COMPRESSION BURNER UNIT

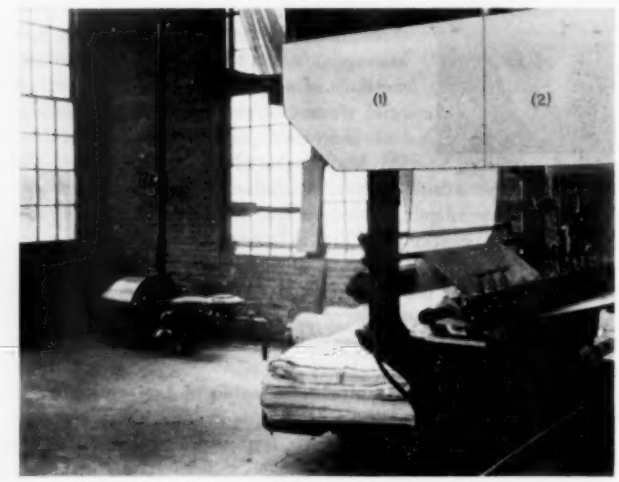


FIG. 4 AUTOMATIC GAS-AIR PROPORTIONING MACHINE AND SINGEING MACHINE, FEED END

In a cotton-finishing plant, the point of heaviest gaseous-fuel consumption is ordinarily in the singe room. Butane is finding applications for this process in open-flame singeing and flame compression singeing.

CLOTH SINGEING

Cloth-singeing operations, in general, have been divided into three classes: (1) plate singeing, (2) open-flame gas singeing, and (3) flame-compression singeing. The first is divided into stationary- and rotary-plate singeing. Plate singeing and open-flame singeing have been in general use for many years. Flame-compression singeing is a relatively new development.

1 Plate singeing depends on cloth contact with heated metal surfaces for its effectiveness. The condition and temperature of the plate surfaces are primary factors in determining the quality of singe secured.

Stationary-plate singeing employs one to seven semi-circular plates, usually of copper, underfired from either side with oil or gas. Thermal efficiency is relatively low due to high radiation losses from the plates and furnace structure. Maintenance

charges are relatively higher than other methods because of the necessity of replacing burned-out plates or of reshaping plates.

Rotary singers are divided into two classes: (*a*) externally fired and (*b*) internally fired. Singeing is effected by cloth contact with a heated alloy rotary plate or roll. In the one case, heat is applied to the external roll surface; in the other, combustion takes place inside the roll itself.

2 Open-flame singeing is done by passing the fabric over a series of gas burners. Singeing is secured by the contact of the flames with the cloth fibers. Gaseous fuel is supplied to the burners in one of the following ways:

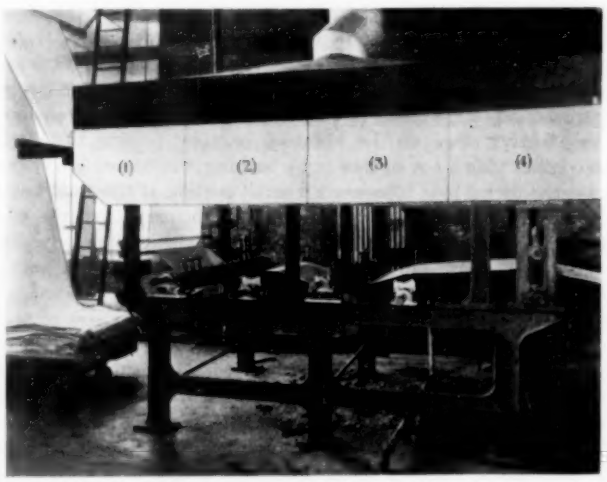


FIG. 5 FOUR FLAME-COMPRESSION SINGEING UNITS, FOR TWO-SIDE SINGEING

a Premixed with air at a central point, a mixture of gas and air under a pressure of $1^1/_2$ lb.

b As straight gas under a pressure of 3 to 10 lb., inspirating air for combustion from the room atmosphere.

c As straight gas, receiving its air supply from a low-pressure air blower, and proportioning near the burner.

3 Flame-compression singeing, developed during the last two years, embodies the principle of liberating a combustible air-gas mixture in a confined space, with the result that combustion takes place in a chamber having a positive pressure. From this it is seen that the flame is not permitted to expand to its normal size, thus the name "flame compression."

In practice, the drilled burner surface forms the bottom of the combustion chamber, water-cooled cast iron the side walls, and the cloth to be singed, the roof. Gaseous fuel, in order to burn in the confined space beneath the fabric, must be fed to the burners as an accurately proportioned and completely combustible air-gas mixture, under an appreciable mixture pressure. This is accomplished by utilizing an automatic proportioning machine, the function of which is accurately to proportion air and gas in the desired ratio, compress and mix to the proper operating pressure, and maintain this pressure and mixture under varying conditions of demand.

Flame-compression singeing is being carried on with city gas, natural gas, gasoline gas, and butane gas as fuels. All of these gases, when proportioned and premixed as heretofore described, are suitable for this work.

To understand the totally different singeing effect secured by flame compression, one must remember that with open-flame singeing good practice calls for a rich gas flame (one containing a deficiency of oxygen for complete combustion), which follows along with the cloth, singeing as combustion completes itself. The operation is carried on in the room atmosphere in the presence of cooling air currents. In addition, when high speeds are run, a current or film of air travels with the cloth. The virtually pressureless soft open flame is unable to penetrate this, and consequently a large amount of the potential singeing effect is lost.

With flame compression the trapped flame exerts a positive pressure on the surface of the cloth, and little heat is dissipated to the surrounding atmosphere. Four flame-compression singers for each singe frame are sufficient to assure high production and a close singe, two units being used for each side of the goods. Each unit consists of a cast-iron, water-cooled combination burner and combustion chamber on which are provided machined cloth-contacting surfaces. Water- and gas-supply valves working in unison enable the operator to control water and gas flow as desired. Flame cut-offs available in every width permit the turning off of any sections of the flame when narrow cloth is being run, and in some installations, permit the operation of two narrow widths of cloth simultaneously over one singer. If independent supply controls are provided for each half of the burner, these widths need not be of the same weight and quality; thus two different singeing effects can be secured on two widths of goods from one singe frame at one time. In an installation such as this, a combined production speed of 600 yd. per min., while singeing face and back, can be secured.

The installation of flame-compression units can be made on existing singe frames with but minor changes in the frame itself. In an installation of four units, alternately facing up and down, cloth is singed both face and back in a single horizontal passage through the singer. By placing the burners off-center, their contact surfaces take the place of carrier rolls and provide the desired contact angle.

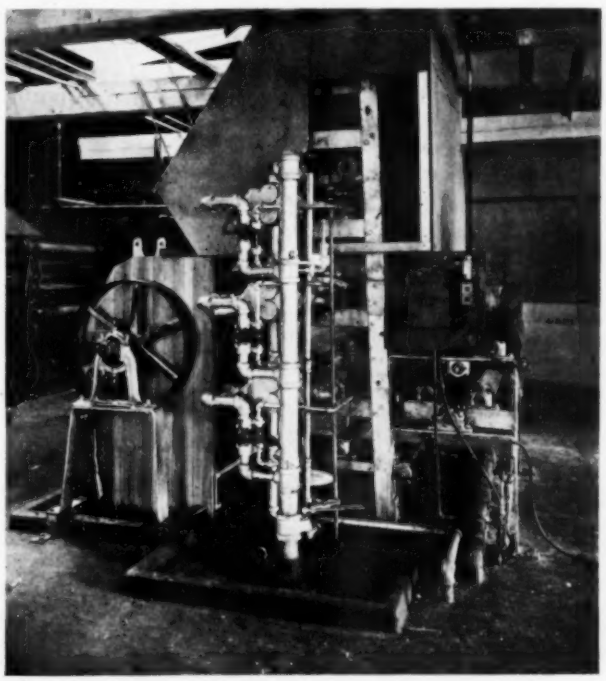


FIG. 6 VERTICAL-SINGEING THREE-FLAME COMPRESSION SINGEING UNITS FOR SINGLE-SIDE SINGEING

An excellent installation of flame-compression units can be secured with the cloth traveling in a vertical plane, the burners being mounted one above the other. This type of singe frame, styled "vertical," occupies less floor space than the conventional horizontal type and facilitates smoke removal.

Discussion

L. E. WAGNER.[2] The author has made some comments on the fuels available for textile operations. The writer is led to make a few observations on the selection of a fuel.

The economy of one fuel over another is probably the most important single factor. It is self-evident that this point must be decided only after a complete study of all the conditions that exist in a given locality. All increments of cost entering into the purchase, delivery, storage, preparation, compression of air or gas, pumping, or cost of auxiliary steam, etc. must be considered. Any tangible savings in labor, production, better quality, and such intangible items which are frequently difficult to isolate must be scrutinized so that the final comparison is made on the basis of fuel burned, balanced against final production.

The physical properties of each fuel must be compared in relation to their effect on the organization and product. Such items as control of combustion, flexibility, cleanliness, etc. all have a definite bearing on the problem, but are sometimes not easy to define.

Present relative values are so disrupted that the problem of selection of fuel is further complicated. For this reason it would seem wise to scrutinize the respective unit costs over a long-time period and to observe past fluctuations in price and to determine if possible the reasons for such fluctuations, so that future trends may be forecast with reasonable accuracy.

In the adoption of a fuel one must be assured of a supply of such fuel, in quantities and at such times as needed. Conditions over which no control is possible frequently affect deliveries. In this respect the public utility is fortunate in that, in case of fuel shortage or other emergency, priority orders are authorized.

Fuel occupying storage space may be at a disadvantage, as space so utilized may earn greater return if put to productive use. Storage of inflammable material always presents a hazard which may affect insurance of the plant as a whole. For instance, liquified petroleum gases are of high gravity, running from 1.5 to 2.0 in the gaseous state as compared to air as 1.0. It is therefore wise to consider the possibility of leaks accumulating as an explosive mixture in basements, etc. Great care should therefore be taken in the design and installation of a handling plant for this material.

Frequently the service rendered by a public utility in assisting its clients in the selection of proper equipment and in following up such equipment has been a deciding factor in the selection of a fuel. A number of large industrial organizations have placed a value of from 11 to 15 cents per 1000 cu. ft. of manufactured gas purchased on such service and on the assurance of a continuous supply of fuel from the gas company.

In the second half of the paper attention is called to a burner which appears to represent a definite improvement in singe burners. The efficiency of this operation always has been low. An effective singe can be obtained only by passing the cloth in contact with a hot surface or by bathing it in flame. In either case the useful heat represents a small portion of the total heat. The rotating plate singer has a greater efficiency than the stationary plate, due to confinement of the heat within insulated walls.

With the new burner, the flame is confined within walls, the cloth to be singed forming the cover. The heat is directed as closely as possible on the work it has to do. Users of this equipment advise that efficiency approaching twice that of open-flame singeing is being secured. Speed of singeing is also greater. In this connection it is only fair to state that greater speed can be secured with gas than with gasoline. The writer would expect this to be true also with manufactured gas over natural gas or liquefied petroleum gases, giving as a reason for this the increased amount of air required for complete combustion of these gases, with the consequent addition of more inert nitrogen to the burning mixture, which slows down the rate of flame propagation and reduces the theoretical flame temperature. Natural gas of 1000 B.t.u. requires nearly twice as much air for complete combustion as does manufactured gas, and butane requires about seven times as much.

This increased amount of inert diluent also may give rise to troublesome carbon deposits within the burner due to the slow-burning mixture coming in contact with the water-cooled burner enclosure. The writer does not think that these deposits would occur in using manufactured gas.

[2] Industrial Sales, Providence Gas Company, Providence, R. I. Mem. A.S.M.E.

Use of the Butted Seam by Textile Manufacturers[1]

By JOHN M. WASHBURN,[2] HARTFORD, CONN.

Increasing use is being made of the butted seam by textile manufacturers, and this paper defines and illustrates the butted seam, describes many of its inherent properties, and outlines some of the advantages to be gained by its use in the finishing of woolens and worsteds, cottons, silks, and rayons.

IN THE processing of piece goods, it is usually desirable to join one or more "cuts" or pieces in endwise relation. Comparatively recently, the overedge stitch or seam has been adapted to this work, and because of the fact that this stitch permits of end-to-end relations between the pieces of goods, at the seam, it has been termed the "butted seam." Either a one- or two-thread type of overedge stitch may be employed for making these seams, and the appearance of each side of the two types of seams is shown in the accompanying illustrations.

Fig. 1 shows the covering or looper thread side of a seam of the two-thread type. The reverse or needle thread side of this seam is shown in Fig. 2, and the goods are usually seamed so that these threads are showing on the face of the goods. The appearance of the two sides of the one-thread stitch is shown in Figs. 3 and 4. The side shown in Fig. 3 corresponds to that shown in Fig. 1 for the two-thread type of stitch. It will be noticed that the appearance on one side of each stitch, Figs. 2 and 4, is identical.

In making the butted seam the ends of piece goods are usually placed face to face and fed through a machine which trims and feeds the goods simultaneously with the stitching operation. The stitching is applied loosely to the edges of goods so that subsequent processing opens the seam and the goods lie end to end with the overedge threads uniting the pieces, thus producing a flat seam, wherein the cut ends of the goods are confined under the uniting threads.

A processing seam of this type has many inherent properties, among which are:

1 The seam is very little thicker than the goods, particularly when fine threads may be used or when the processing and fabric are such as to make the threads embed themselves in the goods.

2 The uniting threads lie in the direction of the warp threads of the goods, and if shrinkage of the thread occurs, it does not reduce the width of the piece goods at the seam.

3 The formation of the stitch is such that the seam is elastic and may be stretched laterally with the goods without fear of breaking.

4 The seam will thoroughly dry with and in the same drying time as the goods.

5 The practical strength of the seam under longitudinal tension is dependent upon the resistance of the filling threads to pulling from the warp threads of the goods, for a reasonable width of seam. Various widths of seams are available that will hold the great majority of goods through common processes.

6 The one-thread seam may be readily unraveled.

7 The width of the seam is comparatively small, and little goods are wasted.

Inasmuch as the quality of the butted seam is such that it approaches the character of the goods, it may readily be realized that its use for process seaming offers many advantages, but like all developments, its practical advantages and limitations have to be determined by experiment. To gain its full use, it has frequently been found practical to combine processes or slightly revise equipment and method of handling the goods to the end of reducing labor and saving material.

A number of processes in various textile fields, where the butted seam has been found to save labor, material, to speed up the process, or to give a superior result, are hereinafter enumerated and certain advantages given in each case.

Woolens and Worsteds

In the process seaming of woolens and worsteds, the butted seam has been used to advantage for speck dyeing. With the end-to-end relation permitted by the seam, this process may be made continuous, without difficulty in the drying. The seam may be dried in the same time and under the same conditions as the body of the goods, which obviates any shading in color near the seam. For speck dyeing in vat, a number of cuts may be joined, dyed, and subsequently dried as one piece.

In fulling or scouring, joining the ends of the piece goods with a butted seam is not of any particular advantage to the material itself. However, advantage lies in the action of the butted seam on the fulling mill. The elimination of extra bulk avoids the pounding of the rolls as a seam passes through them, thus increasing the life of the rolls and their bearings. If two or more cuts are fulled in the same mill for piece dyeing, this elimination of the pounding of the rolls has a decided advantage in preventing the possibility of wrinkles being pounded into the cloth as the seam of an adjacent cut passes through the rolls.

Use of the flat butted seam through the various extractors permits the liquid to be drawn out at the seam as well as the body of the fabric, which makes it of particular importance in the subsequent drying process. Its use through the crab extractor does not present any particular advantage other than the fact that there are no loose ends to throw liquid as the seam passes through process.

Certain advantages to the material, as the butted seam goes through the drier, have been enumerated. In addition, the seam is elastic and will give filling-wise with the goods, and therefore will not shrink or break as it goes over the tenter frame of the drier. This non-shrinking property helps to keep the selvage

---

[1] By permission, certain parts of this paper are being taken from an article to be copyrighted.

[2] Treasurer, The Merrow Machine Company. Jun. A.S.M.E. Mr. Washburn was graduated from Pennsylvania State College in the year 1918, receiving his bachelor's degree in Industrial Engineering. Shortly after graduation he was enlisted in the United States Navy and was commissioned ensign (Engineering Division), upon completion of the officers' training course at the U. S. Naval Steam Engineering School. After relief from active duty, early in the year 1919, he was employed as experimental engineer by The Merrow Machine Company.

Contributed by the Textile Division and presented at the Hartford, Conn., Meeting, June 1–3, 1931, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

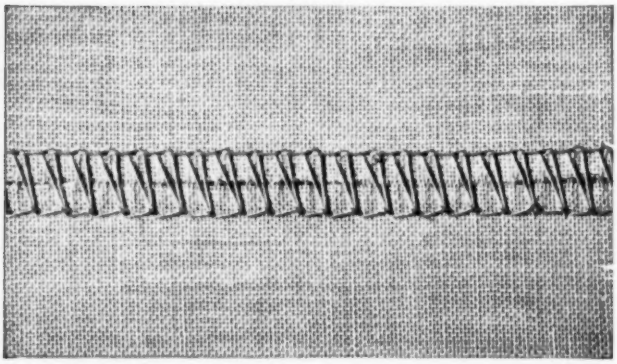


FIG. 1

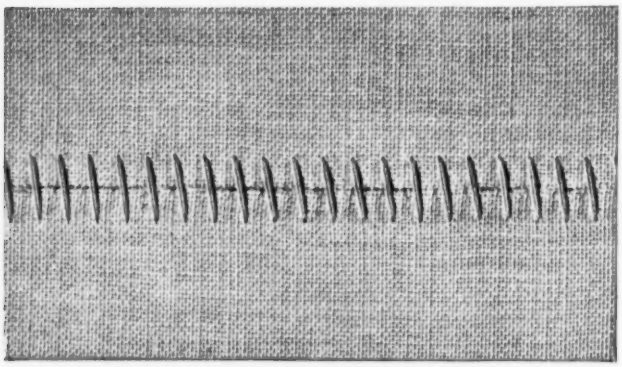


FIG. 2

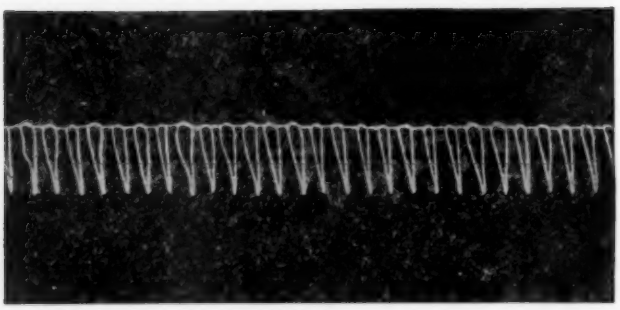


FIG. 3

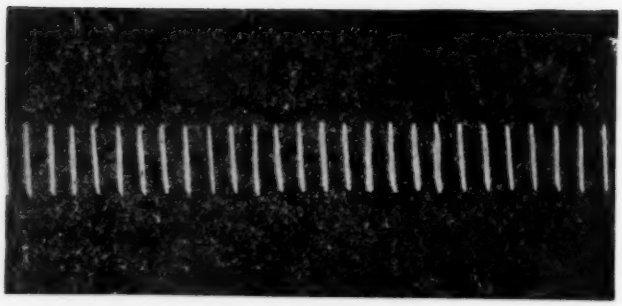


FIG. 4

edges of the goods in line, relieving the tenter-frame operator for other work.

The greatest monetary saving of the butted seam is found in seaming for the process of shearing. The great majority of goods seamed with this seam may be processed directly through the shears without raising the revolvers, making it practical to shear the goods to and over the seam, eliminating all waste of material due to incomplete shearing. On certain classes of light-weight, very closely sheared worsted serges, it is necessary to slightly tap the shears when shearing on the back of the goods. In such cases a block of rubber may be inserted between the shear-raising device so that the normal action of raising the shears will only raise them a small fraction of an inch, permitting the shears to go quickly back to place.

Certain manufacturers have found it practical to place four two-blade shears in series, the first shear shearing the back of the goods, a turning cradle being located between the first and second shears, and the remainder of the shears operating on the face of the goods. This entire unit is handled by one shear operator, and the shearing of the goods is completed as the fabric leaves the last shear.

As a rough example of the material saving of a mill shearing 20 cuts per day by the use of the butted seam as compared with a seam which requires raising of the shear blades, it is not unlikely that with this latter seam, at least 9 in. of goods are wasted at each seam due to incomplete shearing. The 20 cuts, requiring 20 seams, would waste 180 in. of goods per day, which, if made at a cost of $2 per yard, would represent a waste of $10 per day. This waste would be eliminated, to which could be added a substantial labor saving.

The elimination, by the use of the butted seam, of the necessity of reducing the weights or raising the cylinder of the press as a seam passes through, permits even pressing throughout the cut, the ends of the piece goods being pressed equally as well as the center.

If the goods are to be decatized, two cuts may be joined and rolled without the use of a leader. A well-butted seam with the proper thread will not "mark off."

COTTON FINISHING

On cotton finishing in general, although there is a distinct saving of material effected by the use of the butted seam itself, this saving does not assume the importance that it does in the processing of more expensive piece goods, such as woolens and worsteds. However, the improvement it brings to the handling of the goods through the various processes saves much material by the reduction of damaged work, and in such cases as the elimination of the pasted seam, a considerable saving in labor is effected.

The fact that the cut ends of material are enclosed within the formation of the stitches is of practical value in singeing operations. This enclosure of the cut ends prevents them from catching fire and "sparking back" on the goods as they are processed over a flame or red-hot rolls.

In bleaching, the butted seam has proved its value due to the fact that the uniting threads lie with the warp threads of the fabric, which prevents reduction in the width of the goods at the seam and at the same time provides an elastic seam, which may be stretched laterally, without fear of breaking during the tentering process. This non-shrinking, elastic seam is of further value in reducing "wrinkle backs" at the selvages of the material as it goes through this latter process. These same properties are of similar value to the processes of mercerizing, dyeing, and drying, where also the butted-seam property of uniform drying is a distinct asset.

The butted seam is processed directly through printing opera-

tions with uniform printing to and over the seam, and the fact that the seam has not shrunk during process and that it is elastic permits printed goods of square or diamond design to be uniform throughout the length of the piece. This seam may be processed without leaving an imprint on the "back gray" material, which might subsequently "mark off" on the printed cloth. The butted seam is also used in the seaming of the "back gray or bump cloth" material itself, making it practical to cut out damaged or heavily inked sections of this material and reseam into a continuous piece.

In the general process of calendering, including such subdivisions as friction, embossing, chase, and shreiner calendering, the inherent properties of the butted seam are of distinct value. For certain calender work, particularly of a drastic nature, it has become usual practice to break the process seams as they come to this operation and replace them with a pasted seam, consuming about 8 in. of cloth, and is of necessity a comparatively slow operation. In some cases, dependent upon the character of the goods, the butted seam is calendered directly as it comes from previous processes, without breaking the seam. In other cases it has been found necessary to reseam for calendering, making a careful butted seam with fine thread. This latter operation is a speedy one and much labor is saved in comparison with the pasted seam.

The butted seam has been found of value to the process of cotton-goods napping, when it is desired to nap both sides of the material. The uniting threads of this seam lie in the direction of the warp threads of the piece goods and do not disturb the wires of the napping brush as the material passes over them, thus increasing their useful life.

When the process of shearing for slubs and loose thread ends is involved on cotton goods, the butted seam is processed directly through such shears without raising the shear blades, thereby shearing all the goods to and over the seam.

This seam is also of particular advantage in the processing of cotton velours, mohairs, and other plush goods, especially for drying operations following dyeing, bleaching, or washing. On these goods the shearing process is rather similar to that employed in worsted shearing, with the addition of the various pile-raising processes between shearing operations, and the advantages of the butted seam for this work are similar to those gained on worsted materials.

Silk Finishing

In the processing of silks for finishing, the butted seam is used to advantage. Such properties of the seam as its flatness and lack of bulk permit proper "steaming out" of the gum from the silk inasmuch as the goods at the seam will respond to this process in a similar manner to the body of the goods. This thorough "steaming out" at the seams materially aids in the subsequent washing and drying processes. This drying process is also benefited by the fact that the seam will dry in the same time and under the same conditions as the goods themselves, making it unnecessary to keep that portion of the goods including the seam at a higher temperature or in the heat a longer time than the rest of the material. Advantages accrue to the printing of silks similar to those that have been described in the printing of cotton. Shearing for slubs and the loose thread ends is, like similar cotton shearing, carried out without the necessity of raising the revolvers as the seam passes through process, thus shearing the material to and over the seam.

Silk velvets and other pile materials are processed to advantage by the use of the butted seam because of its inherent properties, as has been described. Its use for the shearing of such goods makes a worthwhile saving in labor and material.

Rayon Finishing

The uniform drying qualities and the elasticity of the butted seam are of particular value to the processing of rayon materials through the drying and tentering operations. The butted seam is used for the processes of printing and such light calendering as rayon goods receive, gaining substantially the same advantages as have been enumerated under cotton finishing.

Discussion

George E. Bennett.[3] The paper certainly very completely covers the subject and confirms the experience of the writer's company to the extent that there is nothing that can be added to it that will enhance its value. However, one might suggest for the sake of clarity a few slight changes in the description concerning the printing method. Perhaps an explanation of terms used in printing would better serve to illustrate how useful this seam is in printing operations. In printing, the chief advantage of the butted seam is that it prevents "scrimping" at the seam. Scrimping is caused by the seam pulling in to a width that is less than the normal width of the fabric. A "scrimp" is a crease in the fabric that starts at the seam and runs from 1 to 5 yards through the piece and parallel with the selvage. As the printing ink is applied to the face of the goods it does not get into or under the crease, and consequently when the fabric is finally pulled out to normal width a bare or unprinted portion is produced in the area previously occupied by the crease.

Concerning the butted seam in back gray material, since a butted seam is flat it does not in the majority of cases produce a "seam mark" on the printed piece. A "seam mark" is the imprint of the seam in the back gray left upon the printed piece. One also might mention that where scrimps develop in the back gray material they also leave their impression upon the printed piece in a similar way as in the case of seams.

[3] Glenlyon Print Works, Phillipsdale, R. I.

Design of Furnaces and Fuel Feeders for Burning Refuse

By M. A. HOFFT,[1] INDIANAPOLIS, IND.

This paper includes a discussion of (1) a recent development in the design of a furnace for burning dry wood refuse in suspension; (2) factors influencing the design of wood-burning furnaces in industrial plants; (3) the development of inclined-grate furnaces for burning refuse fuels; and (4) the development of automatically controlled dry wood-refuse feeders and storage bins.



THE problems of refuse fuels are usually visualized, especially by wood industries engineers, as the burning of dry wood refuse from furniture plants, body plants, and sash and door plants with a moisture content varying from 6 to 10 per cent. Sometimes the problem includes the burning of sawmill hogged fuel with a moisture content of from 45 to 50 per cent; however, engineers engaged in the design of general refuse-burning furnaces are often expected to provide for the burning of unusual fuels. The wood refuse may have a moisture content as high as 65 per cent. There also are such fuels to be handled as bark from paper-mill logs, tanbark, sugar-mill refuse, fiber and pulp from palmetto palms, the refuse from brush making, and the wet coffee grounds from commercially operated percolators. The moisture content of some of these fuels runs as high as 75 per cent. Sometimes these installations are needed to dispose of a refuse that has been a nuisance; but often this refuse may furnish sufficient steam for the power and process requirements of the plant, and thus become a source of profit instead of causing expense for its removal. In consequence of the large range of problems presented, the design of refuse-burning furnaces is probably less standardized and there are more different conditions to be met than in any other branch of combustion engineering.

The early methods of refuse burning are too well known to require more than passing mention. Woodworking engineers are usually familiar with the flat-grate Dutch-oven setting for burning wet refuse in cone piles. These are still much in use and give fair results. There is also the flat-grate ordinary setting so typical of the furniture factory of a dozen years ago, where the sawdust and shavings were blown into the furnace from a cyclone and piled up unevenly on the grates, leaving many bare spots for cold air to rush through. Such installations are rapidly passing out of existence, and present methods in woodworking plants show much improvement. The amount of waste per unit of production has materially decreased, and the amount of power per unit of production has increased, until refuse that formerly was a nuisance has come to have such a monetary value as makes it profitable to use furnaces of higher efficiency and greater steaming capacity. Inefficient wood-refuse burning was inevitable so long as blowpipe collected refuse was irregularly mixed on the grate with hand-fed waste. A long stride in the right direction was taken when refuse hogs were introduced. It then became possible to grind assorted wood refuse to a reasonably uniform size and to distribute it evenly over the grate surface.

Burning in Suspension

Dry-refuse-burning furnaces are of two principal types—one for burning finely hogged dry wood and sawdust in suspension (this furnace resembles the regular pulverized-coal furnace) and the other burning refuse partly in suspension, but mostly on inclined grates.

The design of an outstanding burn-in-suspension furnace is outlined here to show its adaptability to certain conditions and to show some of the factors that enter into the decision as to which type of furnace should be used. The inclined-grate furnace is more adaptable to the average woodworking factory; hence it will be more completely discussed in this paper.

The burn-in-suspension furnace requires finely ground refuse that is fed in mechanically at a uniform rate and never blown in from a cyclone. This type of furnace is used for extensive woodworking plants where usually there are other boilers for burning an auxiliary fuel or where the auxiliary fuel, if burned in the same boiler with wood, is pulverized coal, gas, or oil. Fuels that are suitable for burning in pulverized-coal furnaces must be used. Raw coal, blocks, sticks, and rubbish cannot be burned in this type of furnace.

The burn-in-suspension furnace is rather expensive, requiring an unusually high boiler setting, with sufficient vertical distance to burn practically all of the fuel before it drops to the floor of the setting, and a large furnace volume for suspension burning, at least as large as is required for pulverized coal for the same ratings. This necessitates a building of corresponding size for housing this unusual boiler setting.

The maintenance of the refractory walls has been a big problem due to the high furnace temperatures and possibly to the erosion caused by the larger particles of fuel which drop part way to the bottom of the furnace and are picked up by the draft and carried back up to the boiler. The flying particles are doubtless partly responsible for the deterioration of these walls. Air-cooled hollow-wall settings improved the conditions materially. For some time after the development of the burning of dry wood in suspension it was thought that water-cooled walls could not be used because they would reduce the furnace temperature to the point where the refuse would not burn completely before dropping to the floor.

The water-wall setting illustrated in Fig. 1 has solved this problem. This setting is at the Mengel Company plant in Louisville, Ky., and is an excellent example of the burn-in-suspension furnace. It has three improvements over the majority

---

[1] President, M. A. Hofft Company. Assoc. A.S.M.E. Mr. Hofft was born in Indianapolis in 1881 and attended Shortridge High School. From 1897 to 1909 he was with the Chandler & Taylor Company, of Indianapolis, and then with the Hofft Company, of which he is the head. This concern has specialized in equipment for handling and burning all classes of refuse and low-grade fuels. Mr. Hofft has patents on stokers and flat arches.

Presented at the Fifth National Wood Industries Meeting, New York, N. Y., Oct. 16 and 17, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

of the burn-in-suspension furnaces; first, the water wall; second, the hollow floor with air inlets to eliminate the piling of partly burned refuse on the floor; and, third, the location of the roof of the furnace relative to the first bank of tubes. The boiler was set sufficiently high in relation to the roof of the furnace to force a longer travel of the fine particles before passing the first baffle of the boiler. An early practice of this type of furnace was to put the roof of the furnace up near the top drum of this type of boiler. The Mengel Company design assures a more complete burning in suspension, even with the water-wall setting, and also prevents some of the carrying over of fine dust. The author spouts and is a factor in the success of the water-wall setting as it helps to maintain a higher furnace temperature. This also eliminates the puffing of the furnace and the possibility of back-firing into the bin. These boilers are 850 hp. each and easily develop 2200 hp. The auxiliary fuel is pulverized coal from a unit pulverizer.

Burning in Suspension Only Adaptable to Large Installations

The plants in which the burn-in-suspension furnace can properly be used are limited. Most woodworking plants are

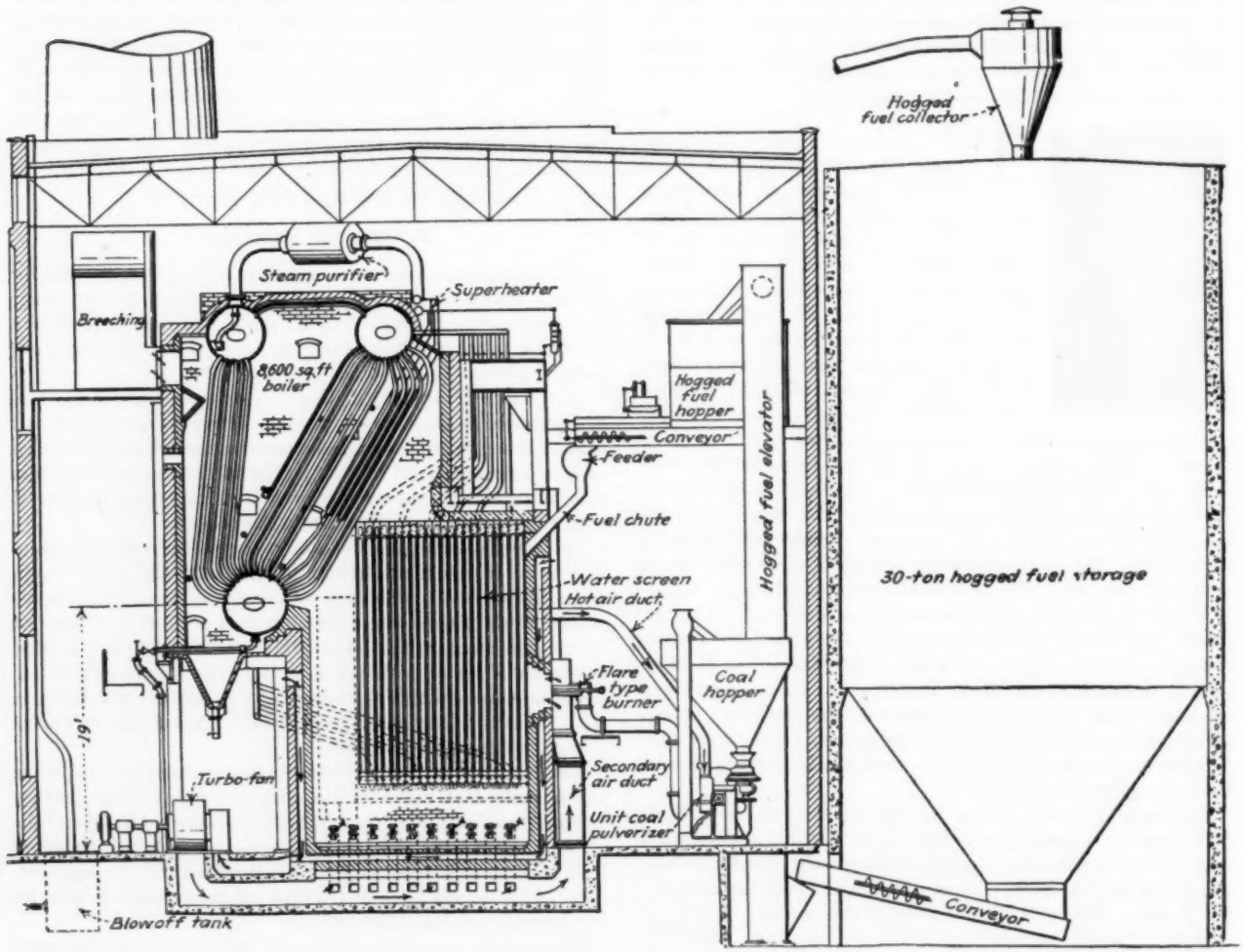


Fig. 1 Design of Burn-in-Suspension Furnace

(For burning kiln-dried refuse in suspension with water-cooled side walls and unit pulverizers for auxiliary fuel.)

believes that this furnace could be further improved by the introduction of air jets at high velocity through the nose of the arch. These small air jets, pointing down, would still further prevent carrying over and would give turbulence.

The sawdust, shavings, and hogged dry fuel are fed into this furnace from the storage bin by an automatically controlled screw feed. Instead of the fuel being delivered by the screws directly into the furnace, with charges of approximately a cubic foot of refuse per turn of the screw, it is fed from the screw through a rotary feeder, the rotary feeder breaking up these comparatively large charges of fuel into a fine spray, allowing an intimate contact of the air and fuel for complete combustion while in suspension. This prevents the introduction of cold air at the feed

not large enough and cannot profitably afford these settings and the equipment necessary for supplying the auxiliary fuel. Further, they must be able to burn blocks, sticks, and rubbish in case anything happens to the hog. The amount of edgings, blocks, and sticks that must be burned may be a fairly large percentage of the fuel. These plants are often unable to accumulate surplus refuse sufficient for holiday and shutdown periods, necessitating the use of coal at these times, even if not during regular plant operation. The installation of good furnaces and feeding equipment would save enough refuse in many of these plants for this night and Sunday firing; however, they must anticipate burning raw coal during holidays and shutdown periods. There also are many plants where the moisture content of the

fuel may be too high or the hogging too coarse to permit burning-in-suspension. For such plants the author has for a number of years been developing the inclined-grate furnace, described herein in detail.

Inclined-Grate Furnace

The inclined-grate furnace does not require the large furnace volume necessary for burning in suspension, since only the fine dust burns in suspension, the larger particles burning on the grates, producing excellent results even on low-set boilers. Fig. 2 illustrates the setting for three 500-hp. boilers. The first boiler in this battery ran a large part of the time for a couple of years at around 300 per cent of rating as it was the only boiler in the plant. In emergencies, steam was obtained from another factory across the railroad track. When the plant was enlarged, two additional boilers were installed. This setting has always met all the requirements of the Smoke Inspection Department and did not have an excessive maintenance cost. A large part of the maintenance cost on this type of setting when operating at a high rating is in the repair to the bridgewall. Some water bridgewalls have been installed with excellent results, holding down the furnace temperature slightly. Water-cooled side walls would be very desirable and would eliminate the maintenance on refractories.

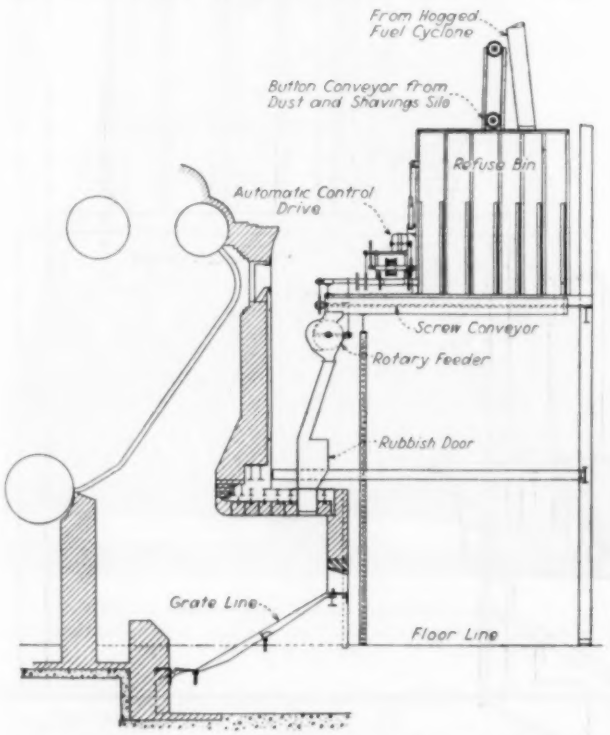


Fig. 2 Dry-Refuse Furnace, Showing Method of Feeding
(Rubbish is fed by hand through the roof of the arch and hogged fuel with an automatic feeder.)

When this plant was operated on one boiler, there was no hog in use; the blocks and sticks were fed through the roof of the arch to the grates, being shoved through a balanced rubbish door. Sawdust and shavings were fed through the same opening in the arch by means of the same type of automatically controlled feeder. During holidays and shutdown periods raw coal was used for fuel. The coal was fired either through the rubbish opening above the arch or through the firedoors as desired. It was fired on the top end of the grates, and the coking method of operation was used. In firing coal these boilers are easily operated at about 125 per cent of rating. The plant now has hogging equipment in use. The hogged fuel is fed through the automatic feeder. The furnace volume on this is 1280 cu. ft., or 2.53 cu. ft. per rated horsepower. The arch on this setting is 18 in. higher than standard.

When this type of furnace is run at high ratings, it is necessary to introduce extra over-air, especially when being fed with the automatic feeder shown, as there is little air admitted with the fuel. Over-air is introduced through a ratchet-operated vent door in the firedoor, or if battleship doors are used, they are tilted slightly. Air admitted in this way is forced by the arch to mix with the gases before coming into contact with the boiler, so there is no appreciable loss in efficiency. The CO_2 may be maintained approximately as desired. However, when resinous woods are used for fuel, it will smoke at approximately 12 per cent CO_2. Oak and similar wood refuse may be fired with a higher CO_2 without smoke.

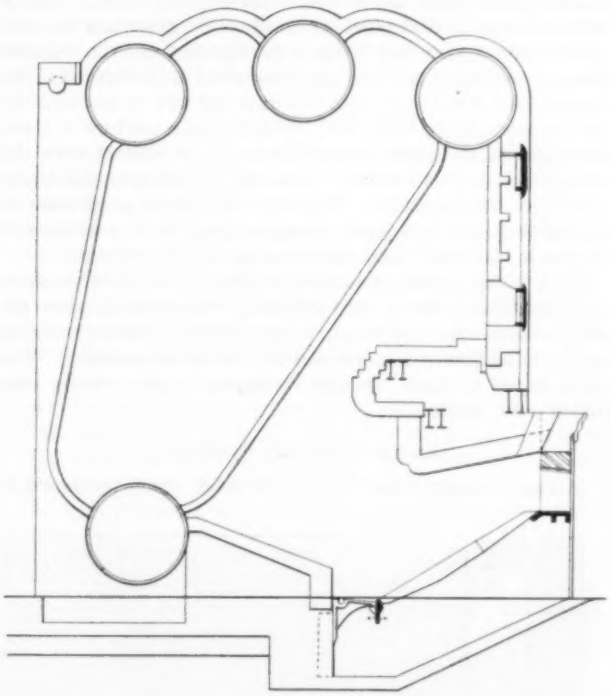


Fig. 3 Efficient Smokeless Dry-Refuse Furnace
(This has an extremely small furnace volume.)

Low Boiler-Setting Installations

While it is not desirable to set a boiler low and a good furnace volume is desirable, it is possible to get smokeless and efficient results with a low setting and restricted furnace volume. The setting illustrated in Fig. 3 is for four 350-hp. boilers at the plant of the M. B. Farrin Lumber Company in Cincinnati. These boilers had been in operation with flat grates for a number of years. Large quantities of charred and partly burned fuel were being carried out of the stack and deposited over the surrounding buildings, causing complaint and legal difficulties, not only with the Smoke Inspection Department, but with the neighbors. It was impossible to raise the boiler, or to make an extension Dutch oven or to lower the floor. The furnace, as illustrated, was installed, and it completely eliminated the carrying over, and incidentally was enough more efficient to make it unnecessary to use coal as an auxiliary fuel. Approximately

$8000 worth of coal had been used each year. These boilers have often operated at 200 to 250 per cent of rating, and normally at 150 per cent of rating, to the entire satisfaction of the Smoke Inspection Department. One would expect to have a high maintenance cost on this restricted setting, but the reverse has been the case. The inclined-grate setting has been in service for 10 years, and the maintenance cost for brickwork and furnace castings for the last three years on these four 350-hp. boilers was $1879.08 for materials and labor.

The furnace volume on this setting is 586 cu. ft., or 1.67 cu. ft. per rated horsepower. This is outlined here to show that unusual results may be obtained even with small furnace volume. This is due to the fact that the fuel is burned on the grate and the arch construction is such that it forces the fine dust and the volatiles that are distilled off the fuel that is deposited at the top end of the grate to pass down over the hotter fuel bed and to mix with the thinner gases at the lower end; thus forcing a mixture of these fine particles and gases before they come in contact with the cooling surface of the boiler. The arch on this setting is about 1 ft. lower than standard. This was done partly on account of limited headroom and partly to more surely force a mixture of the gases, and rather high maintenance cost was expected.

There is also another installation of 500-hp. boilers of the same type and which are set very similarly, with approximately the same results. As mentioned, these restricted furnace volumes are not to be desired, but are used only in case of necessity. The refuse is fed to these furnaces through an automatically controlled screw feeder.

Return-Tubular Settings

A return-tubular boiler setting for small plants is shown in

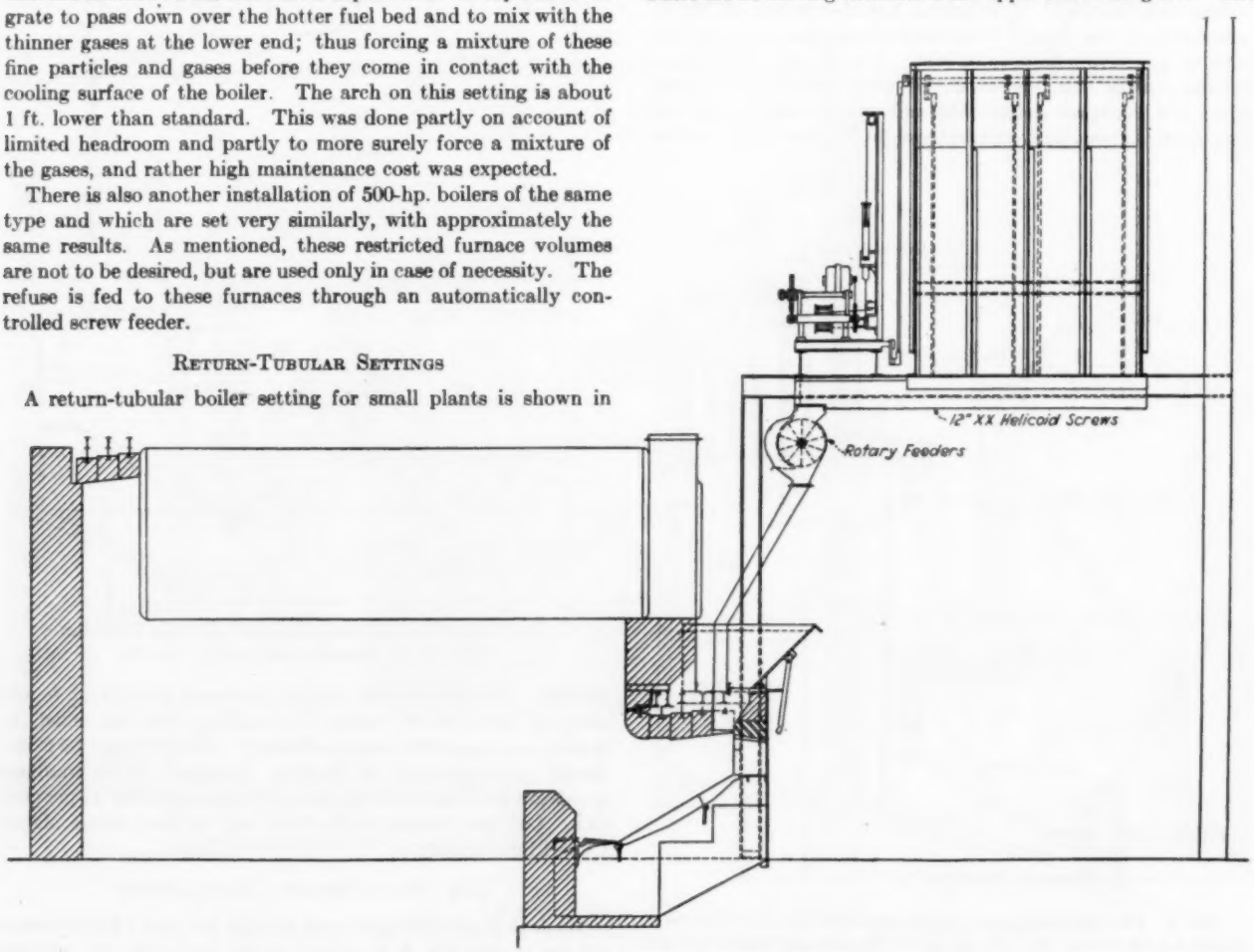


Fig. 4 Furnace for Return Tubular Boiler
(This shows the method of feeding coal and wood refuse through double-feed spout.)

Fig. 4. This illustrates the method of feeding coal through the top of the extension Dutch oven on the inclined-grate furnace. With this setting there is a coal hopper on top of the extension Dutch oven and a double-opening feed spout extending through the arch to deliver the fuel to the top end of the grates. Sawdust shavings, and hogged fuel are fed through an automatically controlled screw conveyor delivering the fuel through one of the openings in the double-opening feed spout. Coal is fed through the other opening of the same feed spout by means of a slide gate which drops it through the arch. The rotary feeder in the automatic feeding device makes an air seal between the feed bin and the furnace. The slide gate permits dropping the coal from the bunker to the top end of the grate when needed and also shuts off the air from the coal bunker. The coking method of firing the coal is used. The firedoors below are still available for firing coal or for blocks and sticks if desired. The firing of the blocks and sticks in this type of furnace is much like throwing them into a hole, as they drop to the rear end of the grates and it is not necessary to shove them through a cone pile of wood as in the case of a flat-grate setting.

Grate and Dump Construction

The construction of the grate, fuel pusher, and dump is shown in Figs. 5 and 6. The grates are longitudinal and stationary, and for dry wood are set on an incline of approximately 30 deg. There are no moving elements in the upper end of the grate. The air spaces at this point are horizontal, so as to prevent dripping of fine fuel into the ashpit. The lower section of the grates is provided with two to six rows of fuel pushers, depending on the length of the grate. Each row of pushers is individually operated from the front by a separate lever. Since the grates are longitudinal and stationary, there are no projections or cross-ridges to hold up the fuel. The refuse fuel deposited at the upper end slips slowly down the grate as it burns, avoiding the necessity of shoving the refuse fuel down the grate. In burning refuse fuels it is very objectionable to agitate the fuel bed, especially at the upper end. This agitation is eliminated with this type of grate.

With the proper angle to the grate, the fuel does not avalanche and make a big pile at the bottom.

There are dumps at the lower end of the grates for the removal of ash. The cleaning of the fire is accomplished by raising the lower row of pushers operating between the stationary grates and dropping the dumps.

The raised lower row of pushers prevents the slipping of good fuel into the ashpit and at the same time kicks off the ash and clinker through the dump opening. The fuel bed on the main body of the grates is not disturbed in this operation. The fuel pushers are 12 in. long and travel through between the stationary grates approximately 10 in., which is ample travel to kick off the ash and clinker without the necessity of poking through the cleanout doors. The dump openings are the full width of the furnace, having no bar across the opening to hold up the ash and clinker. Any sand slag that may be formed will be on the lower

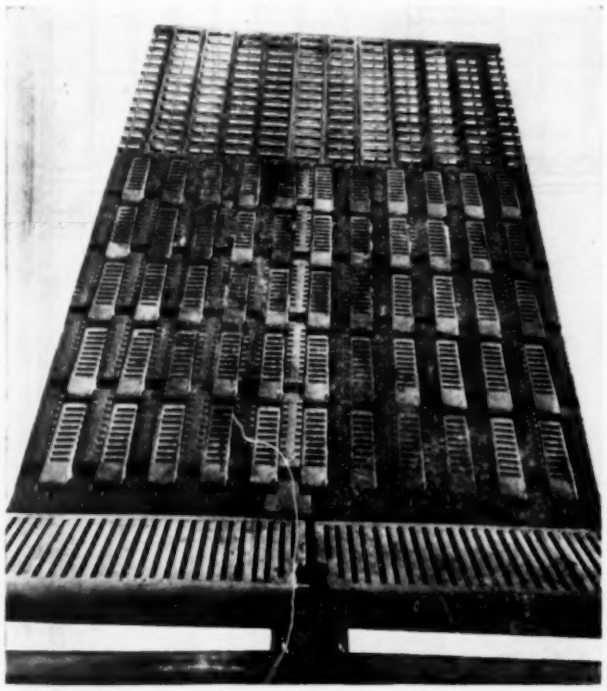


FIG. 5 STATIONARY GRATE CONSTRUCTION
(This has horizontal air spaces in the upper section and fuel pushers in the lower section.)

section of the grate and is removed by the operation of the other rows of pushers.

When clean wood refuse is being burned, the cleaning and pusher operation is only occasional, usually once in several days. At that time it is often advisable to keep the lower end of the grates covered with ash so that the air entering the furnace will be forced to enter at the upper end and mix with the gases under the arch, and not be allowed to short-circuit through the lower end of the grates or dumps. If it becomes necessary to use coal for the auxiliary fuel, this ash can be dumped out quickly and the coking method of firing coal used. The coking method of firing coal and the arch construction which forces the volatiles to travel down over the fuel bed make a practically smokeless fire, even with high volatile coal. Of course with coal firing it is necessary to use the fuel pushers and dumps more frequently, their use depending on the kind of coal and the rate of firing.

It is best to fire coal and wood refuse separately. When firing both at the same time, the temperature created by the burning wood on top of coal will fuse the ash and cause heavy clinkers unless the coal has a high fusing ash.

The illustrations, of course, show but a few types of boilers. There are a great many of these installations under all types of boilers and operating under varying conditions. It is usually possible to very greatly increase the capacity of the boiler, sometimes doubling the previous capacity; also to get practically smokeless operation. The increased efficiency obtained from the available wood refuse is often sufficient to eliminate the use of coal.

The tendency now in designing boiler plants for woodworking industries is to design a boiler setting for the highest possible efficiency on the major fuel, which is wood, and not to burn the wood in a furnace designed for the auxiliary fuel, coal. This often means that very little, if any, coal is needed.

WOOD REFUSE FEEDERS

The result from any refuse-burning furnace largely depends on the feeding method. With the old method of feeding, sawdust and shavings were blown from a cyclone directly into the furnace as they were made in the factory, regardless of the needs for steam. The furnace would be flooded with shavings at one time, and in a few minutes it might become necessary to put in coal, often the cyclone blowing in nothing but cold air. This led to the development of equipment for regularly feeding the refuse to the furnace; first, the use of the chain drag and, later, the automatically controlled screw feeder. In handling the dry refuse with the screw feeder, the sawdust, shavings, and hogged fuel are blown into a bin and fed to the furnace by means of heavy screws, the speed of these screws being controlled by the steam pressure.

The early problems of this method of feed were, first, the fire



FIG. 6 DUMP AND FUEL PUSHER OPERATION FOR CLEANING FIRE

risk caused by possible backfiring and puffing of the furnace and, second, the arching of the fuel in the bin. The first problem was overcome by the use of a rotary feeder between the end of the feed screw and the furnace. This made an effective air seal, keeping any back draft and any sparks and backfire from getting to the bin. In addition to this it shut out the air entering the furnace with the feed, which air was found undesirable. It was found also to effectively break up the comparatively large charges of fuel delivered by the screw spreading it more evenly in the furnace. The second problem, the arching of the fuel in the bin, was overcome by the use of a bin with a flat steel bottom and either vertical or slightly tapering sides. The bottom of this

bin is practically covered with a series of heavy helicoid screws. A typical installation of this kind is illustrated in Fig. 7. The screws covering the bottom of the bin are connected in series of three, any series or all the screws may be operated as desired. These screws slip the fuel forward into a trough extending at right-angles to the screws. This slipping forward and the slight agitation caused by the turning of the screws effectively overcome the tendency of the fuel to arch, even with the use of coarse hogged fuel and ground veneer strips having a considerable quantity of long slivers. These bins have been built as large as 7000 cu. ft. capacity, and there seems to be no reason why bins of much larger capacity cannot be built without arching.

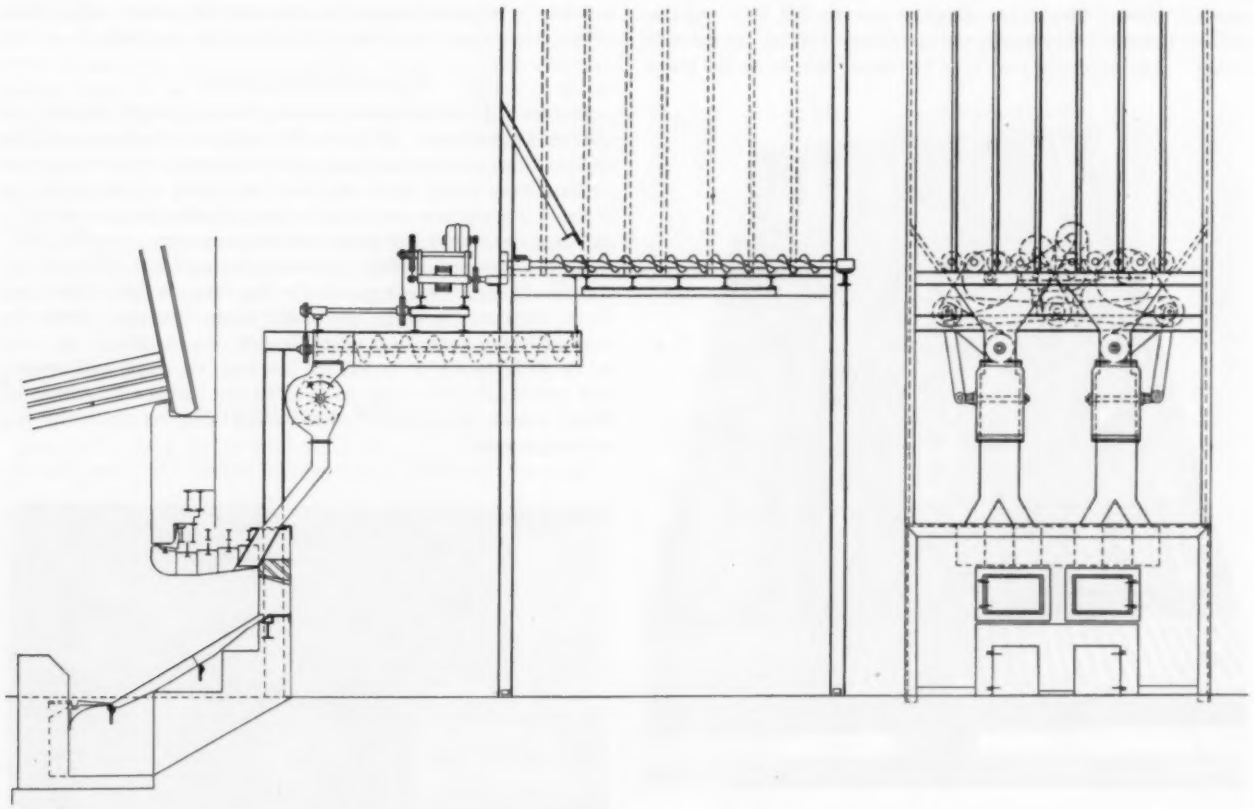


FIG. 7 AUTOMATIC SCREW FEEDER WITH SCREW BOTTOM BIN
(This prevents the arching of hogged fuel.)

This feeding equipment, even for large boilers, is driven by a 3-hp. constant-speed motor driving through the proper reduction gearing a variable-speed transmission. The arm controlling the variable-speed transmission is connected to an automatic pressure-control valve, which moves this arm back and forth, controlling the speed of the unloading screws on the bottom of the bin, and it is able to so control this feed as to hold the steam pressure to within 3 or 4 lb., even with a widely fluctuating load, the speed of the unloading screws on the bottom of the bin being controlled by the steam pressure. The capacity and the speed of the lower feed screws and rotary feeder are sufficient to prevent the fuel from accumulating in the trough, avoiding the possibility of plugging up the feed screws or the rotary feeder. The sides of the rotary feeder are counterbalanced so as to discharge any blocks or pieces of metal that might accidentally get into the machine. A small steam engine may be substituted for the motor, in which case it is advisable to control the speed of the variable-speed transmission rather than by throttling the engine, as the control is closer and the operation more satisfactory.

The automatically controlled screw feeder is not well adapted for burning wet refuse, as the quantity of fuel to be handled is very great and heavy, and there is no fire risk in connection with its handling. This class of fuel can be handled with much lower cost and as satisfactorily with an ordinary drag conveyor in which the control of the feed is accomplished by controlling the depth of the fuel on the grates, as will be described.

FACTORS IN THE DESIGN OF THE FURNACE

In the design of the furnace for burning these widely varying fuels, the moisture content, of course, is the first consideration. The heat value of the fuel, the size of the particles of the fuel as delivered to the furnace, the freedom with which the fuel burns, the tendency to pack densely on the grate, the draft conditions, and the capacity desired are a few of the factors that enter into the design of furnace.

The angle of the grates for this wet refuse varies and is from 30 to 50 deg. The length of the grates also is determined by the moisture content and rate of firing and the width of the boiler setting. The length and contour of the arch also are governed by the same factors. The depth of the fuel fed at the top end of the grate is governed by the nature of the fuel, coarse-ground wet hogged wood refuse requiring about 18 in. of fuel at the top, sawdust about 15 in., and other fuels varying up to bagasse, which requires about 3 ft. 6 in. of fuel at the top end, on account of its being light and porous.

For this type of furnace the volume in the firebox proper need not be as great as on the old-style Dutch-oven furnace because the inclined-grate furnace firebox is really a gas producer, com-

bustion being completed more in the back combustion chamber than is the case of the old-style setting.

It is very necessary to have as much refractory surface as possible to keep drying out the incoming refuse and to help drive out and ignite the volatile gases. In order to accomplish this it is advisable to use two or more furnaces for wide boilers. Boilers of 700 hp. and larger have been equipped with three furnaces per boiler, and settings are now being designed for 1200-hp. boilers with four furnaces to the boiler. This multiple-furnace construction is obtained by building the grates in sections and putting refractory division walls between the sections, not only giving the additional refractory surface needed, but causing the fuel fed to the top end of the grates to spread evenly over the entire section, overcoming the tendency to have thick and thin strata down the grate. With this furnace the cleaning of the fire is done so quickly on each individual section that there is practically no loss of steam during the operation. The ash and clinkers are dumped into the ashpit by the dumping and pusher operation described for dry-refuse furnaces.

Any sand slag that may be formed on the grates is usually formed on the lower or pusher section and is quickly removed by the operation of the fuel pushers between the stationary grates.

It also is possible to operate with one section of the furnace idle or to burn one class of fuel on one section of the grate and another on the other section of the grate. When one section is being cleaned, the draft is shut off of that section, preventing the cooling down of the furnace by the incoming air.

Grate Design

Longitudinal stationary-grate construction with fuel pushers operated between the grates at the lower end has overcome the difficulties with the early inclined-grate furnace. These early furnaces were known as a stepladder design and had grates extending crosswise of the setting, which had a tendency to hold up the fuel and keep it from sliding evenly down the grate, thus developing holes in the fire and causing some avalanching of the fuel.

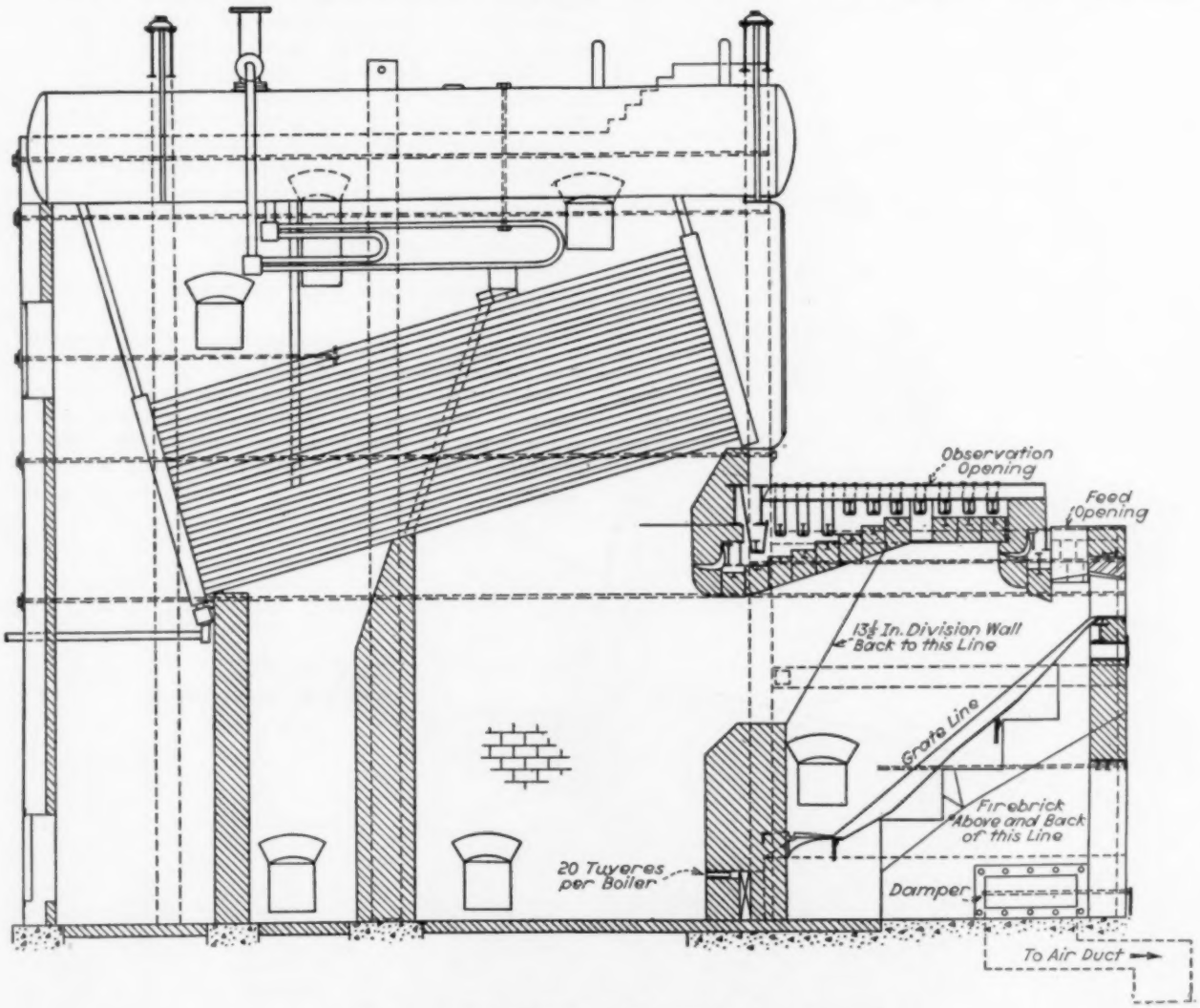


Fig. 8 Typical Design for Sawmill Hogged Fuel

Design for Wet Hogged Waste

A typical setting for burning a good quality of hogged fuel of 45 per cent moisture content for a 600-hp. boiler to operate at 160 per cent of rating is illustrated in Fig. 8. The depth of the fuel bed at the upper end of the grate is regulated by the location of the feed opening in the arch; the arch is then stepped up in as

short a distance as possible to allow the radiation to strike the drying pile of fuel as quickly as possible and to get ignition high up on the pile of drying fuel. The arch is extended back over the grates to just past the bridge wall, and the nose of the arch is dropped at the bridge wall. This arch contour is to force a mixture of the heavy volatile gases distilled off the drying pile with the hotter, lighter gases rising off the lower end of the grate. These gases are finally burned in the combustion chamber back of the bridgewall. The angle of the grate is 40 deg., the furnace volume in the furnace proper is 680 cu. ft., the volume of the back combustion chamber is 2058 cu. ft. This setting is provided with dampers for forced draft. Forced draft is used at the higher ratings. Tuyères are placed in the bridgewall for air at this point.

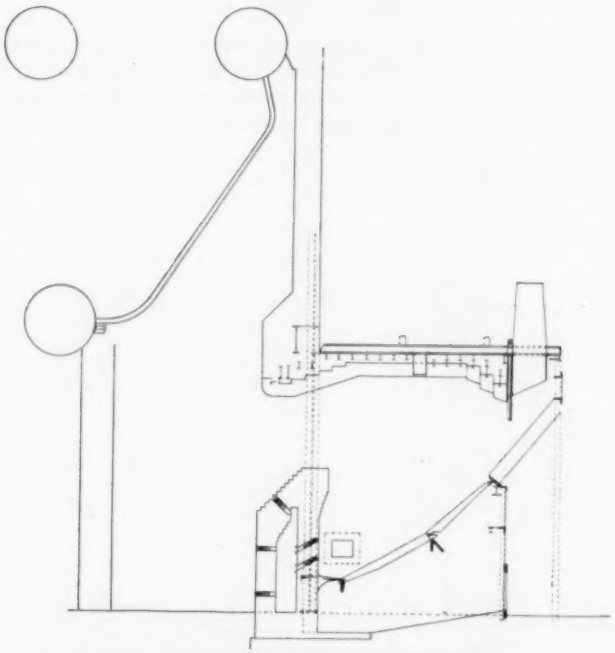


FIG. 9 DESIGN OF FURNACE FOR VERY WET HOGGED FUEL OR BARK

A furnace designed for a wetter fuel and to operate at higher ratings is illustrated in Fig. 9. Under these conditions it is not possible to get sufficient air through the fuel bed to get double rating even with forced draft, as the draft will pick up the fuel and carry it over the bridgewall. This setting also is a 600-hp. boiler. The grate area is much longer than on the furnace for drier fuel. In this design the fuel is dropped on a refractory hearth, the thickness of the fuel bed being regulated by the gate. When once regulated to the fuel to be used, this gate need not be moved unless the class of fuel is changed, the steam output of the boiler being controlled by the air damper. This refractory hearth is set on 50-deg. incline. The next section of the grate is stationary, with horizontal air spaces, and is set at approximately 45 deg. This change in angle is made because the fuel is slipping more freely this distance down the grate. The lower section where the fuel pushers may be operated between the grates is set at approximately 30-deg. incline. This change in angle is to prevent the possibility of too heavy a fuel bed at the bottom and on the dump at the lower end. With this furnace the bridgewall is tilted forward to force the lighter, hotter gases from the lower end to mix better with the heavy volatile gases from the drying pile of fuel at the top end.

Provision is made for a large quantity of over-air or auxiliary air through air ports placed through both the front and rear faces of the bridgewall, and air openings also are provided in the side walls above the grates. This very materially assists in speeding up combustion, especially if this air is put in under pressure. Suitable dampers are provided at these air openings so that the auxiliary air may be used as desired. The air ports through the back face of the bridgewall are to burn any unburned particles that may carry over the bridgewall. With some fuels it is practically impossible to get sufficient air through the fuel bed. It is also inadvisable to allow the air to enter with the fuel on account of cooling down the arch. This is the reason for putting the air

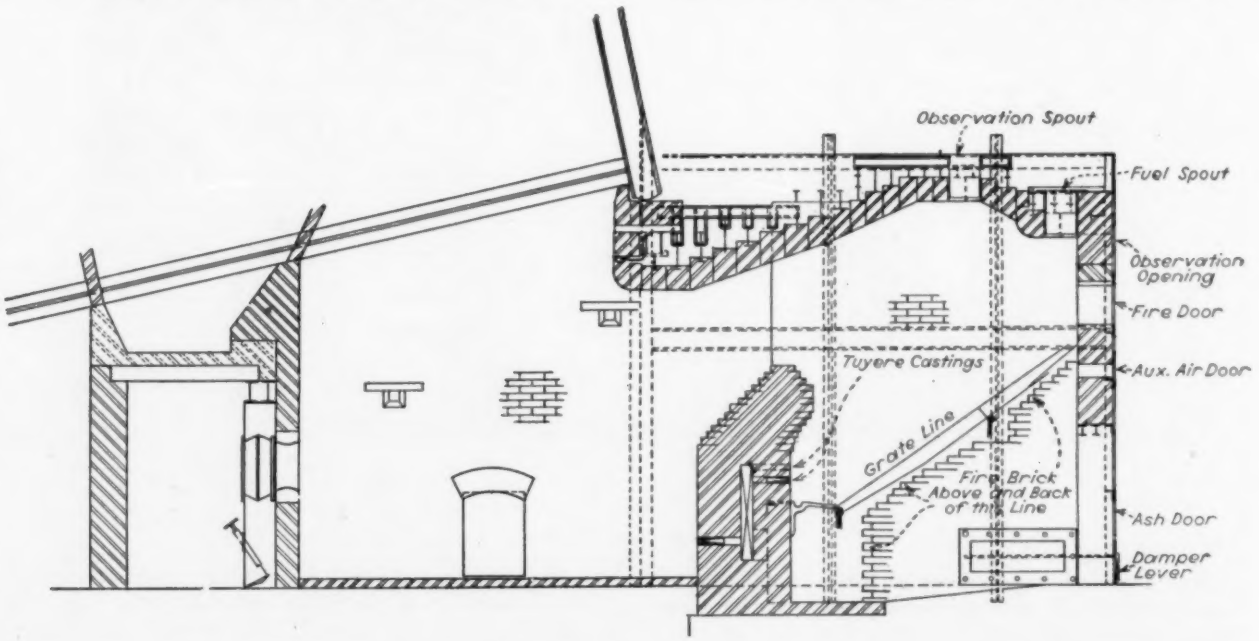


FIG. 10 TYPICAL DESIGN FOR BAGASSE FURNACE
(This shows the oil burners in the rear bridgewall.)

through the sides and bridgewall, preferably under pressure. This gives the necessary turbulence to speed up combustion. The regulated fuel depth and control of the air permit the operation of this furnace at very high CO_2, and almost all of these furnaces will be found operating at from 14 to 15 per cent CO_2. It is possible to get a higher CO_2 on wet refuse without deterioration to the refractories than on dry. On dry refuse a very high CO_2 of course is very hard on refractories. For wet refuse it is not possible to obtain sufficient temperature in the furnace to seriously damage the refractories, and these furnaces will run for years without any maintenance on the furnace, probably due to the fact that the highest temperatures are in the combustion chamber and not in the furnace proper.

Furnace for Burning Bagasse

A typical setting for burning bagasse refuse from cane-sugar mills is shown in Fig. 10 and Fig. 10a. This setting is for three 900-hp. boilers with multiple furnaces, three furnaces to each boiler. As this plant is also a sugar refinery requiring more steam than for the making of raw sugar, provision is made for the use of oil as an auxiliary fuel. This oil is fired through the back bridgewall as shown. This furnace may be operated on bagasse with natural

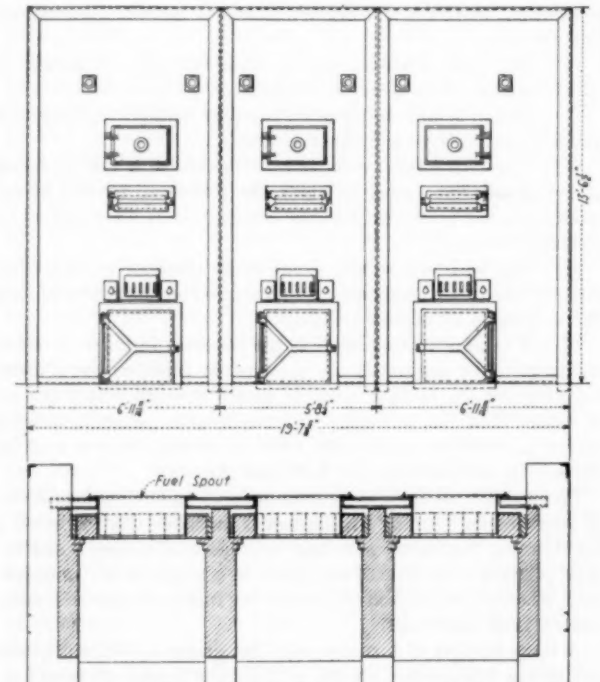


Fig. 10a Cross-Section Through Furnace as Illustrated in Fig. 10, Showing Multiple Furnaces

draft at 150 per cent of rating, and with forced draft it will run 200 per cent of rating or more. The forced-draft air connections are through the bottom of the division walls, with dampers arranged so that the air may be blown to any or all of the divisions of the furnace.

Bagasse is a light and porous fuel requiring a heavier fuel bed and has a greater tendency to burn holes in the fire than has wood. The multiple-furnace construction and grate design have solved this problem of keeping an even fuel bed across the grate. The small opening in the top of the arch, as shown in the illustrations is a treadle-operated observation door over each division of the furnace.

The setting illustrated in Fig. 11 was originally designed for burning a large quantity of very wet coffee grounds of about 75 per cent moisture content to be used with about 20 per cent coal. It was found that with the addition of the coal too much steam was made. The furnace was altered by dropping the feed-spout openings as shown, to regulate the fuel depth to the proper height, and the bridgewall was altered to give more refractory and to create the proper throttling effect between the arch and the bridgewall to bring up the furnace temperature. This alteration permitted the operation of this furnace on the coffee grounds taken from the percolater without the use of the auxiliary fuel. There are 17 or 18 tons of coffee grounds burned per day on one of these furnaces for a 150-hp. boiler.

Other extremely wet fuels and fibers are burned. In some cases where the waste fuel burned is not sufficient to supply the steam required for the operation of the plant, coal is used as an auxiliary. For extremely wet fuels this coal is put on the dumps through a side opening and acts as a backlog to help bring up the furnace temperature. This makes possible the burning of any refuse that is combustible.

Conclusion

In the designing of the furnace for efficient dry-refuse burning consideration should be given the moisture content of the fuel,

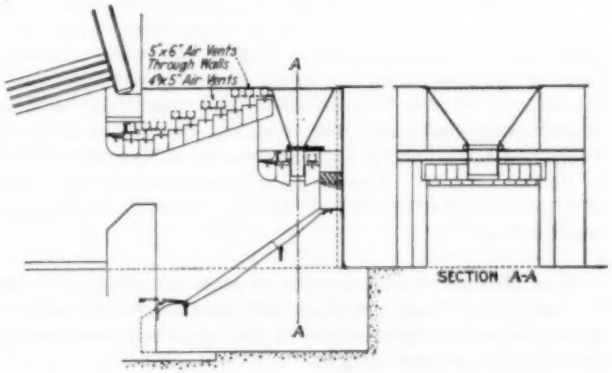


Fig. 11 Furnace for Burning Extremely Wet Refuse
(This design takes care of material from distillation or cooking processes.)

the fineness of the hogging of the fuel, the disposition of the blocks and rubbish, the kind and quantity of the auxiliary fuel, and the method of handling and firing the auxiliary fuel. For both fuels it is very essential to have an even and controlled feed of the fuel.

In designing a furnace for burning wet refuse the first consideration is the moisture content of the fuel; consideration also must be given to the size of the fuel as delivered to the furnace, the type of boiler, the density with which the fuel packs on the grate, its burning qualities, the draft conditions, and the rating desired.

The tendency in recent years in designing furnaces for industrial woodworking plants has been more to design the furnace to burn the major fuel with the highest possible efficiency, and in this way to reduce the auxiliary fuel to the minimum, and to use a furnace that is adapted also to the burning of the auxiliary fuel available.

Discussion

H. W. Brooks.[2] My own experience while connected with the design and operation of about 300 power plants has included 5 wood-burning plants. This led to the conclusion that the utilization of wood as a fuel was a rather insignificant matter when compared with some of the other energy sources in the United States with which we are more familiar. It has been, therefore, a cause of great astonishment to happen across a chart furnished by the United States Geological Survey which shows the sources of energy in the United States. This gives 0.1 per cent for wind power, 2.5 per cent for animal power, 6.3 per cent for water power, 5.5 per cent for firewood, 8.1 per cent for anthracite coal, 50.3 per cent for bituminous coal, 21.4 per cent for oil, and 5.8 per cent for natural gas. It is astonishing to learn that firewood produces, according to this chart, nearly 80 per cent as much energy as is produced in the United States by water power. It is almost unbelievable that firewood produces around 70 per cent as much energy in the United States as does anthracite and that firewood produces approximately as much of the total American energy as the total of natural gases.

There is hardly a subject whereon dependable data are less available than the utilization of wood and wood wastes as fuel. The statistics of the Geological Survey show that it is a hugely important subject, yet the bibliography is woefully lacking. There are a few publications along the general line, most of them obtainable through the courtesy of the National Committee on Wood Utilization of the Department of Commerce. The Bureau of the Census has included wood fuel in its 1929 schedule, but these figures will not be available for a year or more. In the October issue of *Combustion* there is an article by A. R. Smith, of the Combustion Engineering Corporation of New York, entitled, "The Utilization of Waste Wood for Steam Generation," which may be of some interest. The Forest Products Laboratory, of Madison, Wis., has issued two technical notes under the subjects of "Fuel Value of Wood" and "Fuel Value of Wood Waste Overestimated." The National Committee on Wood Utilization has a bulletin entitled, "Survey of Nonutilized Wood in North Carolina." There is a particularly interesting bulletin of the same committee entitled, "Sawdust, Wood Flour, Shavings, and Excelsior."

Sern Madsen.[3] This paper approaches the subject of refuse burning from a broad viewpoint and gives due consideration to all the individual characteristics of the fuel to be burned and the size and type of plant using it.

Referring to the burning of dry fuel as accumulated in most of the furniture factories, body plants, sash and door factories, etc., it has not generally been appreciated what economies can be effected by using the proper types of furnaces and feeding equipment for burning this refuse.

Old customs have influenced these industries in the belief that wood refuse was something to be disposed of rather than to be used as a valuable fuel. It is not generally appreciated that this dry fuel when burned on proper grates, with ideal admission of air through the fuel bed, produces a fire of such intense heat that it requires that special attention be given the refractories. It is highly desirable that some of the water- or air-cooled side walls, arches, and bridge walls developed for central-station operation should be adapted for use in furnaces burning dry refuse. Installations which have been designed for maximum efficiency produce intense temperatures far above those thought possible a few years ago.

The woodworking industry has improved its production machinery until it compares very favorably with that used in the metal-working industries. Now let us get the utmost out of the valuable fuel which has formerly been called and treated as waste. It is not unusual for these installations to pay for themselves by the savings made the first year. This is indeed a happy opportunity—to make improvements that not only pay their way but that produce dividends.

W. E. Allington.[4] The author has stated many interesting facts in connection with the subject, but he has, perhaps inadvertently, made some misstatements and omitted others and has not given a clear understanding of the principles of suspension burning and furnace design, type of grates, and automatically controlled wood-fuel feeders and storage bins.

These subjects have ramifications which run into almost endless complications, and with an experience of over forty years, the writer still is learning and still is coming in contact with problems that present new factors which need special consideration. A full treatment of these subjects so far as now understood would require a book instead of a short discussion such as contained in the paper, and no doubt it was difficult to know just what problems to discuss and where to leave off.

Instead of two types of furnaces for burning wood fuel in suspension, there are at least five principal types of furnaces, as follows:

(1) One with large combustion chamber without grates, of which General Motors and subsidiaries have over twenty.

(2) One with large combustion chamber with grates of various kinds, including several inclined grates.

(3) One with smaller combustion chamber with various forms of practically flat grates in which the fuel is all ignited before landing on the grates so that the combustion is not confined to surface burning.

(4) One with still smaller combustion chamber in which the inclined grates are used and the fuel is piled on the grates so that the burning is all surface burning.

(5) A type of furnace such as the Stratton furnace, in which the large upper portion of the combustion chamber has a lower tapering portion leading down to grates through which the air is heated and fed upwardly at considerable velocity, holding heavy particles in suspension which otherwise would land on the grates and be consumed with less efficiency.

Information on the evaporation secured in the various forms of furnaces for burning in suspension indicates that for such a fast-burning fuel as dry shavings, sawdust, and hogged wood the type of grate or the use of any grate is comparatively unimportant, whereas the method of introducing fuel to the furnace under control is all important.

A large number of furnaces with flat grates or without grates are giving evaporative results of 6 lb. per pound of wood fuel. With no grates the ashpit is cleaned about once in three months, and it has been said that a layer of ashes about 2 in. deep is found.

Our patented, automatically controlled wood-fuel stokers are operating in connection with furnaces with inclined grates and with similar furnaces with flat grates, and apparently there is not much, if any, difference in evaporative results. A comparative test between flat grates and inclined grates with all other conditions unchanged would be interesting. The Hofft grate is a good grate, and the writer does not wish to be understood in condemning it in any way, but our experience in a great number of cases indicates that changing flat grates to inclined grates would not yield anything like the improvement in evaporative results that the same expenditure would give if applied to increasing the size of the combustion chamber.

[2] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

[3] Curtis Companies Incorporated, Clinton, Iowa.

[4] President, Allington & Curtis Mfg. Co., Saginaw, Mich.

The difference in expense between a small and a large combustion chamber is not great, as both would have suspended arches, which would cost the same for either type.

The author's Fig. 3 illustrates a low boiler setting with inclined grates, four of which are in use by the M. B. Farrin Lumber Company in Cincinnati. It is true that charred and partly burned fuel was carried out of the stack before the inclined grates were installed, and this was caused by the blast of air from the cyclone which fed the furnaces at that time, but the author fails to state correctly what stopped the partly burned fuel being carried out. This result was secured by installing, simultaneously with the Hofft grates, Allington wood-fuel stokers which automatically deliver the fuel to the grates without any air blast; and in our opinion, if the grates were dropped level, at least as good if not better results would be secured because of the increase in size of the combustion chamber and the greater distance the fuel would drop before striking the grates, thus giving the particles of fuel time to become ignited before landing.

It is not a fact that blocks, sticks, and rubbish cannot be burned in furnaces designed for suspension burning, provided they are supplied with grates.

The refractory walls of a furnace are not injured by erosion of larger particles of fuel which drop part way to the bottom of the furnace and are picked up by the draft and carried back up to the boiler. This is, first, because these particles do not come in contact with the walls and, second, because if they are heavy enough to drop part way to the bottom of the furnace, but are only partly consumed and are picked up by the draft and turbulence of the gases, they are held in suspension and burned in suspension and are not forced against the walls of the furnace, as there the gas velocities are the lowest.

If high furnace temperatures are produced which injure the refractories, it is not indicated that the efficiency of suspension burning should be sacrificed, as a lesser fuel supply will reduce the temperature, but if the highest ratings are to be secured from the boilers, high temperatures must be maintained, and this is true of all furnaces and not confined to furnaces designed for suspension burning.

In suspension burning, whether complete or partial, it is important that the fuel be fed in a steady, open stream so that the particles are separated from each other the instant they enter the furnace. This we accomplish by a comparatively high-speed screw which feeds a steady, continuous stream of small cross-sectional area and does not feed with charges of approximately a cubic foot of refuse per turn of a screw, as stated in the paper. Our present method was adopted and our former paddle-wheel feeder-reel method was abandoned because the feeder reel empties each pocket in a mass to the feeder chute and not in a fine spray as stated.

There are comparatively few woodworking plants, large or of any size, having boiler settings in which the combustion chambers cannot be largely increased at reasonable expense considering the added economies and higher efficiencies to be secured. Of course there are very small plants where there is little efficiency in any department, but oftentimes a considerable enlargement of combustion chamber can be secured merely by lowering the grates and so obtain the semi-suspension combustion results for the wood fuel. An immediate coal saving has been the result.

ARTHUR D. SMITH, JR.[5] In regard to the smoke nuisance, our boiler plant consists of two 235-hp. return-tubular boilers, equipped with Hofft grates and furnaces and Allington-Curtis wood refuse stokers. Since the installation of the furnaces and stokers, we have had no difficulty in complying with the City of Philadelphia smoke ordinances, although prior to their installation we were in constant difficulty with the city authorities. It is our opinion that the even rate of feed of the wood refuse is of paramount importance in eliminating smoke.

As to furnace refractories, we have had some trouble with maintaining the refractory nose tiles in the Hofft furnaces. We find that very high temperatures are formed in these furnaces When operating at 150 per cent rating, we have developed a furnace temperature high enough to clinker a 2800-deg.-fusion-point coal.

Due to the high furnace temperatures developed, it is our experience that unless a coal is used with a sufficiently high fusion point, no end of difficulty will be experienced with the Hofft furnaces and grates. If clinker is formed, it is very difficult to move the rocker arms which operate the pusher grates, and if one is successful in breaking the clinker, a great deal of combustible coal will be carried down with the clinker into the ashpit. For this reason, we have insisted on buying coals with the highest possible fusion point.

We have found, however, that the Hofft furnace will handle coal with a volatile content as high as 35 per cent, without causing objectionable smoke. To be assured of no smoke trouble, we would recommend limiting the volatile content to 25 per cent, although the higher content may be used if extreme care is taken in coking and pushing back the coal. With the higher volatile coal, it has been found helpful to admit quite a little excess air for a few minutes after each firing.

Due to the slanting ashpit, it requires considerable labor to take out the ashes. Under the conditions in our boiler plant, the firemen are not able to handle coal of over 10 per cent ash content.

There is one question in regard to the mechanical hog which the writer would like to raise. Both the knife type and hammer type have been described, but the power requirements of each have not been discussed.

RAYMOND E. CALLER.[6] It is common practice in all power plants to think nothing of spending power enough to pulverize coal, and to pulverize coal in a plant will run all the way from 15 to 25 kw-hr. per ton. To crush coal instead of pulverizing, requires from 3 to 5 kw-hr. per ton.

THOMAS D. PERRY.[7] On the question of the hog that has been brought up, it is not always possible, as we have discovered in our own experience, to exercise choice. For instance, the knife hog will work well on veneer or thinner substances and also on a number of types of wet wastes. When, on the other hand, we come to relatively large blocks or if we run into fairly substantial chunks that may come from one cause or another, we find that a hammer hog is almost essential. On the other hand, the hammer hog will not handle wet veneer. In other words, the choice of hogs is to quite an extent dependent on the type of material that one has to reduce.

We found with a large waste bin in a Hofft apparatus that it is often more efficient to build up a full bin and burn it into the night. We built in our plant a bin calculated to hold enough fuel for three days' firing, and in that way we can largely eliminate the overloading of feed apparatus during the day. In other words, we run this surplus fuel off at night when we are not using other power.

MR. CALLER. How do you get the three days' supply out of that bin?

MR. PERRY. We have never had three days' supply yet. It is calculated to carry three days' supply. We have never yet had any cause of difficulty in being unable to feed from the underside instead of the incoming drop on top. What is the size of the bin referred to by Mr. Puckett, speaking for the author?

[5] Geo. W. Smith Woodworking Co., Philadelphia, Pa.

[6] Allington & Curtis Mfg. Co., New York, N. Y.

[7] New Albany Veneering Co., New Albany, Ind.

N. P. PUCKETT.[8] It was 21 ft. high and held about 21 tons.

MR. CALLER. We have been unable to prevent arching in a bin over 10 ft. high.

MR. PUCKETT. We have carried adzer shavings, which are very much like a straw pile, as high as 20 ft., without any difficulty.

M. H. TEAZE.[9] Several years ago we were employed as engineers out on the Pacific Coast, and Mr. Hofft gave us a proposal of one of his grates which was turned down principally because it was felt that it was not long enough. We finally installed a grate which was designed for burning coal, and we made that furnace 12 ft. long. We used preheated air for the firing. Practically all the combustion was on 6 ft. of grade surface. The grates function perfectly and gave an excellent fire, but for some reason there was a clinker in the perforations, which are similar to those that Mr. Hofft uses. Those clinkers are so hard that they could not get the stuff broken out. We finally decided to abandon those grates and have adopted a step grate which is somewhat similar to the upper Hofft grate. The principal purpose of this is to enable the operators to get in the furnace over the week-end and break up the slabs.

MR. BROOKS. One often wonders whether refractory problems are entirely a question of temperature or whether they may not be a question of the chemical constituents of the fuel that we are burning, possibly forming low waste with the refractories themselves and thereby producing the deleterious effects that give the trouble. In one installation of the Ford Motor Company in northern Michigan that question came up, so that it is likely that many have had similar problems.

MR. CALLER. It is found in Detroit plants that when they go beyond a furnace rating of about 175 per cent, sufficient heat is generated from the combustion, so that the very chemica combination that is referred to takes place and starts to work on the refractories. Running wood waste at a high rating necessarily makes high temperatures. That has been so serious in the furnace upkeep that it is a hard rule with some of our customers that they will not run wood over 175 per cent.

MR. BROOKS. In the coal-burning line, we are out of the habit of speaking in terms of per cent of rating. Today we think in terms of boiler horsepower per cubic foot. I wonder what this 175 per cent rating means. I wonder if any one ever has isolated any of the chemical elements that are so harmful to the refractories. What chemical element is it that is released at these high temperatures that produces that effect?

MR. CALLER. I have no information to go on record with, but I believe that the chemists and engineers from the General Motors could very well reply to that question.

MR. BROOKS. The matter of silica alone does not seem to enter into it, because silica itself is one of the best refractories. There must be some combination with silica. There are certain of the siliceous metallic combinations which have very low ash fusion; ferrous silica will run down to 600 or 700 deg., so that probably there must be some combination of the silica with some metallic form producing a fusion.

MR. SMITH. Very often a part of our work is made from the so-called fireproofed wood, which has been treated by impregnation with certain salts. We have found that where even a small percentage of the shavings or refuse is from fireproofed wood, the fusion of the ash within the furnaces was very greatly increased. The writer has seen our furnaces with the ash fused to the refractory walls until it projected out into the furnaces 12 in. or more.

It has been stated that the knife hog is not suitable for large blocks or heavy lumber. We have a knife hog in our plant which is now successfully hogging materials up to 6 in. thick, as well as the thin veneers and lighter woods. We are also using it for reducing all of our crating materials, including nails and light strap iron. Unintentionally, some heavier metals have got into the hog and left their mark on the knife, but our hog operates on an average of three hours per day and the knives are changed only once in every two months. From this, it does seem that the knife hog is capable of handling the heavier material and also crating materials without much serious damage.

MR. PERRY. One fact comes home from the discussions, and that is the lack of thorough and proved information on the value of wood waste as a fuel. With that also, we must consider that the utilization of this waste is not in all cases a primary power-production problem. It often becomes a problem of disposal of a rather miscellaneous type of waste accumulating from a variety of woodworking operations. In our factory we have a large amount of glue, wrapping paper, and various other things that accumulate in a factory, and while Mr. Smith has made an interesting plea for the knife hog, I still continue to feel that where there is a considerable amount of metal coming in with the waste, one cannot sort out for the hammer knife as the other type. In the hammer hogs, they generally have a metal pocket which, by centrifugal force, throws off the heavier particles. I imagine that two-thirds of the metal will be segregated and thrown off.

MR. PUCKETT. With regard to the inability to operate the pushers, probably the low-fusion ash and the treated wood have brought about those conditions.

AUTHOR'S CLOSURE

In burning coal and wood in the same furnace the best results are obtained by firing them separately, although this is not always necessary. However, it is not advisable to burn dry wood and coal, with a low-fusing-point ash, in the same furnace at the same time; for the burning of the dry wood, especially at high ratings, will bring up the furnace temperature to a point where it will fuse the ash of the low-fusing-point ash coal. It is even difficult to burn this low-fusing-point ash coal at high ratings, alone, on account of the melting of the ash and heavy clinkers. These difficulties of course are not experienced with coal having a higher fusing point to its ash.

The stoker Mr. Teaze refers to as having air spaces similar to the one described in the paper probably was of the shaking-grate type with grates 6 or 8 in. by 3 or 4 ft., extending crosswise of the furnace. This grate would have a tendency to slag over the air spaces. That is the reason for the design of the grate with the pushers 3 in. by 12 in. operating between stationary grates like a rake. The operation of a row of these pushers between the grates cracks off the sheet of sand slag that sometimes forms in a furnace burning refuse from logs having considerable sand in the bark or that are delivered by salt water. This glassy slag would also come in plants where there is a considerable amount of sander dust. This slag is rather brittle and would be removed with the raking of the pushers between the grates.

With regard to arching of fuel in the bins, the screw-bottom bin described was developed to overcome serious trouble caused by the arching of fuel in the bin. This fuel was part dry sawdust and shavings and part wet veneer strips that were not finely hogged, but came in rather large pieces. The ordinary agitators would not break up this fuel. After a lot of experiments with this fuel, this screw-bottom bin was developed, and there never has been any arching since, and even more difficult adzer shavings are being handled with the same type of machine.

The author limited this paper to his personal experience in this type of equipment and did not attempt to describe all types of wood-handling equipment and refuse-burning furnaces, as these designs have been ably discussed in many other papers.

[8] Engineer, M. A. Hofft Company, Indianapolis, Ind.

[9] H. S. Ferguson Company, Consulting Engineers, New York, N. Y.

WDI-53-2

Longitudinal Shrinkage of Wood

By ARTHUR KOEHLER,[1] MADISON, WIS.

Longitudinal shrinkage of wood ordinarily is so small as to be considered negligible, and no allowance is made for it for most uses. There are uses, however, for which even the small longitudinal shrinkage of normal wood is an important factor, and others in which the shrinkage along the grain of abnormal wood is so large as to be a serious handicap. This paper gives information on the shrinkage of wood along the grain about which very little has been published.



SHRINKAGE along the grain of wood may be objectionable under the following conditions:

1 When it causes undue shortening in length, particularly in posts, so as to throw the adjoining members into serious disalignment. As a rule, no allowance is made for longitudinal shrinkage as is usually done for transverse shrinkage, and consequently when it does occur to an appreciable degree it often causes trouble.

2 When it causes exceptionally wide opening of butt joints.

3 When it causes bowing or cross breaks in pieces which shrink unevenly longitudinally.

4 When it weakens wood in longitudinal shear on account of stresses set up as a result of variations in shrinkage potential of different layers in timber or other members.

5 When it causes slivers to loosen and raise up from the face of a board.

Variations in Shrinkage

The data here presented are the result of an initial investigation by the Forest Products Laboratory into causes for the variation in the shrinkage of wood. There are large variations in the shrinkages of the different species of wood, in the individual pieces of the same species, and in different directions on the same piece. These variations can be accounted for primarily by differences in the cellular structure.

The longitudinal shrinkage of wood normally ranges from 0.1 to 0.2 of 1 per cent. Unless otherwise stated all shrinkage values given in this paper represent the total shrinkage from the green or soaked to the oven-dry condition, that is, a condition of zero moisture content. In drying to an average air-dry condition of about 12 per cent, the shrinkage would be about one-half as much.

Cases have been observed in which the shrinkage along the grain was abnormally large. The greatest longitudinal shrinkage so far recorded at the Forest Products Laboratory is 5.78 per cent in wood from the lower side of a limb of western yellow pine. This is more than the average transverse shrinkage for the species. It is equivalent to a shortening of 11.1 in. in a 16-ft. board. On the other extreme, slight elongation has been observed in certain woods in drying.

Fig. 1 gives shrinkage values for 34 different types of wood at three different percentages of moisture content; namely, about 12 per cent, about 7 per cent, and at 0 per cent. Shrinkage is assumed to have begun at a moisture content of 24 per cent, which is taken as an average fiber-saturation point. Different types of wood were selected so as to give as wide a variation in shrinkage as possible; consequently the values given in Fig. 1 are not intended to represent averages for any species.

Conditions Under Which Excessive Longitudinal Shrinkages Occur

No sharp limits can be set above which longitudinal shrinkage in wood may be called excessive. A total uniform shrinkage potential of 0.3 of 1 per cent, however, is near the maximum permissible for most uses. It would cause a shortening of 0.58 in. in a 16-ft. board if oven dried. For numerous uses, 0.2 of 1 per cent would be the maximum permissible. Fortunately most wood falls below these limits. If the longitudinal shrinkage is not uniform from side to side, an even smaller shrinkage may be objectionable on account of the bowing which it may produce. Shrinkage beyond these limits has been found to occur in the following types of wood:

1 *Compression Wood.* Compression wood is a hard, heavy, brittle type of wood formed on the lower side of branches and leaning trunks of coniferous trees. It is characterized by rather wide annual rings containing wide summerwood that is not so dark and hornlike as in normal wood. Fig. 2 shows a cross section of a spruce log in which compression wood is well developed. Within any annual ring compression wood is confined to one side of the pith, that is, it does not extend continuously around the trunk. It may be continuous from the pith to the bark, or it may occur in intermittent layers. In some annual rings it may be in one direction from the pith and in others in a different direction. Therefore, it frequently happens that boards or dimension stock have streaks of compression wood in them. Compression wood, having a greater tendency to shrink longitudinally than normal wood, usually causes a bowing of lumber in which it is unevenly distributed. (See Fig. 3.) Occasionally streaks of compression wood that adjoin normal wood will pull themselves apart so as to form cracks across the grain.

Compression wood is a source of considerable trouble in the drying and use of softwood lumber, principally on account of the bowing which it produces. Sometimes the direct shortening causes trouble. For example, butt joints in redwood and cypress siding occasionally open up on account of the longitudinal shrinkage of compression wood in the siding. If the siding is well-dried before it is nailed down, however, there will be no occasion for any further shrinkage to take place. No sharp line of distinction can be drawn between compresion wood and normal wood the two grade into each other imperceptably. Therefore, no fixed shrinkage value can be ascribed to compression wood. The maximum longitudinal shrinkage so far found in compression wood has already been given for wood from a limb of western yellow pine. The maximum shrinkage so far determined for compression wood from a tree trunk is 5.42 per cent in a southern

---

[1] In charge of Section of Silvicultural Relations, Forest Products Laboratory, Forest Service, U. S. Department of Agriculture, maintained in cooperation with University of Wisconsin. Mr. Koehler has been engaged in the study of the cellular structure of wood for identification purposes and to explain the variations in the behavior of wood since 1911.

Presented at the Fifth National Wood Industries Meeting, New York, N. Y., Oct. 16 and 17, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

yellow pine. More often the shrinkage of compression wood is less than 1 per cent but over 0.3 of 1 per cent.

2 *Abnormally Light-Weight Wood.* Wood below the average weight for a species shrinks more along the grain than heavier wood of the same species. This is shown in Fig. 1 for high- and low-density wood of longleaf pine (1 and 2), tupelo gum (22 and 23), ash (25 and 26), and for red oak of low density which shrunk over 0.8 of 1 per cent. Fig. 4, which shows the shrinkage of redwood of different specific gravities, clearly indicates a general trend toward a reduction in shrinkage with an increase in specific gravity. All of the specimens represented in Fig. 4 above the 0.400 per cent line and about one-half of those between the 0.300 and 0.400 per cent lines consisted of compression wood. With this fact in mind, the downward trend of the remaining points which represented normal wood almost entirely, is more apparent. Balsa, on the other hand, which normally is a very light wood, did not shrink so much as most of the other specimens

3 *Springwood.* In many species of wood each annual growth layer is differentiated into an inner, softer layer of low density, the springwood, and an outer, harder layer of high density wood, the summerwood. The springwood invariably shrinks more along the grain than the summerwood. This is easily demonstrated by whittling out a piece of wood only one annual ring in thickness and observing the way it bows lengthwise of the grain as it dries.

The greater longitudinal shrinkage of springwood as compared with summerwood is shown in Fig. 1 for loblolly pine (3 and 4), southern yellow pine (5 and 6), abnormal southern yellow pine (7 and 8), Douglas fir (10 and 11), and redwood (13 and 14). Fig. 4 shows that the specimens of redwood of lowest specific gravity, which consisted entirely of springwood, shrank considerably, whereas those of highest specific gravity, which consisted entirely of summerwood, shrank very little. Figs. 1 and 4 show that some pieces even elongated in drying. Such elongated pieces invariably consisted of summerwood only.

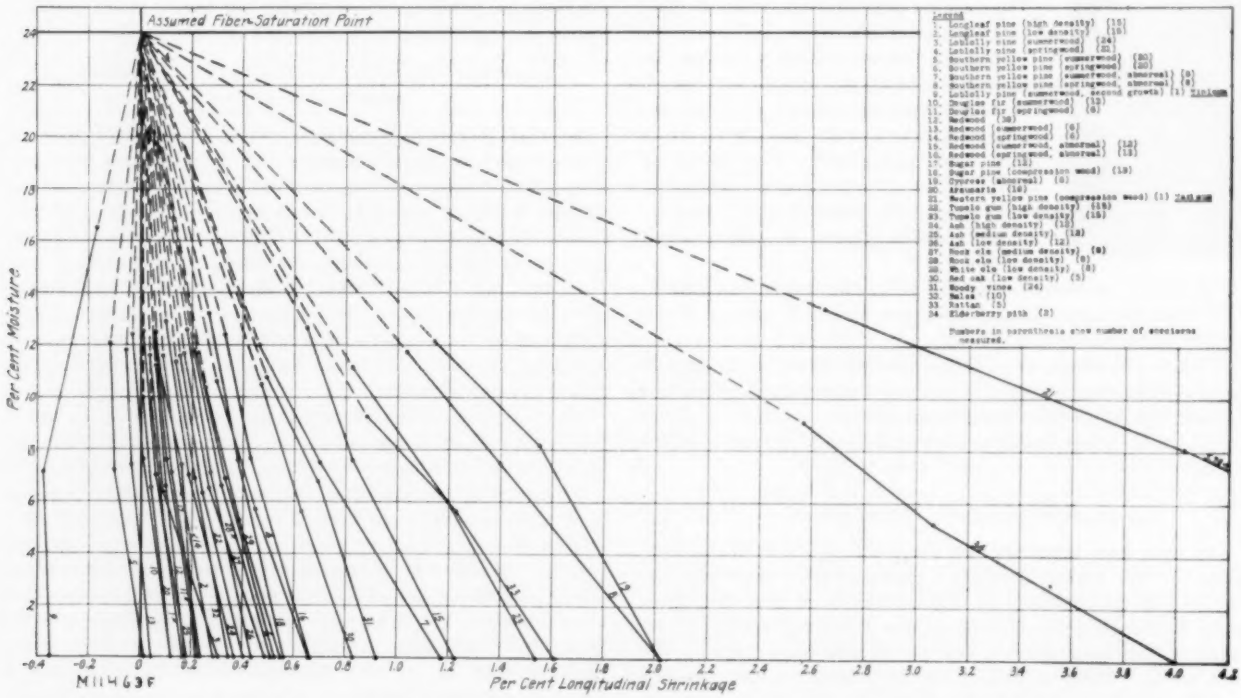


FIG. 1 LONGITUDINAL SHRINKAGE OF VARIOUS TYPES OF WOOD AT THREE DIFFERENT PERCENTAGES OF MOISTURE CONTENT

that were light in weight. (See Fig. 1.) It is only wood which is considerably below the average weight for the species that shrinks excessively along the grain.

This peculiar relationship is due to the fact that in normal-weight wood most of the fibrous cells, or wood fibers, are constructed so that they do not shorten much in drying, and since the wood fibers predominate, as a rule, they control the shrinkage of wood as a whole. In abnormally light-weight wood, however, either the wood fibers have a different type of structure or they are so scarce that the other kinds of cells predominate, particularly those that make up the pores, which have a high shrinkage potential. The relationship between longitudinal shrinkage and density is just the opposite of what it is for transverse shrinkage and density; that is, in general the heavier the wood the more it shrinks across the grain. The relationship between density and shrinkage, whether longitudinal or transverse, is caused by the fact that the woods of low and high densities have different cellular structures, and not by the fact that there is a difference in the amount of wood substance present.

Within a piece of wood several annual rings in width, the springwood, of course, cannot shrink more than the summerwood since the two are so closely and alternately bound together. The difference in shrinkage potential, however, undoubtedly sets up stresses which may possibly produce shear failures either directly or when supplemented by stresses due to external loads.

In flat-grain lumber the greater longitudinal shrinkage of springwood frequently causes slivers to loosen and curve away from the surface, especially on the pith side. This condition frequently is exasperating in flat-grain flooring. It can be largely avoided by dressing pattern stock so that the face side is on the bark side of the piece. In that event, the summerwood of each annual ring is toward the outside, and consequently the tendency of the annual rings to curve up is eliminated.

Rotary veneer occasionally happens to be cut so as to be one annual ring in thickness. Under such circumstances, any difference in shrinkage potential of springwood and summerwood causes local curvatures.

4 *Wood Near the Pith of the Tree.* Boards and planks of the

southern and western yellow pines, and occasionally other species, which happen to be sawed lengthwise through the pith, frequently show cross breaks extending for a short distance at right angles to the pith. (See Fig. 5.) Such breaks are due to the greater longitudinal shrinkage of the wood near the pith. For the same reason, narrow strips cut out so that the pith runs along one edge will bow as they dry.

On the other hand, it has been reported that the wood on the bark side of some hardwood shrinks more than the wood away from the bark, and thereby causes bowing. The explanation probably is that the density of the wood in such pieces decreased toward the outside of the tree, as frequently is the case, and as previously stated, low density wood shrinks more longitudinally than that of high density.

5 *Wood in Very Fast Growing Conifers.* In loblolly pine and red-



FIG. 2 CROSS SECTION OF SPRUCE LOG SHOWING COMPRESSION WOOD—THE DARK, WIDE-RINGED PORTION

wood relatively high longitudinal shrinkage has been found in second-growth trees with very wide annual rings containing summerwood which is not so dark and hard as in normal wood. Such wood resembles compression wood in some respects, but differs from it in extending entirely around the tree trunk and in some microscopic features. Wood of this type was found mostly in the butt portion of the trunk. Probably it also occurs in other species than the ones mentioned.

6 *Cross-grained Wood.* In any piece of wood in which the grain does not run parallel with the long axis, there may be excessive apparent longitudinal shrinkage due to a transverse shrinkage component being effective along the length of the piece. If the grain ran at an angle of 45 deg. across a board it would shrink as much on a percentage basis along its length as along its width. Wood with spiral, diagonal, interlocked, wavy, or curly grain may have excessive apparent longitudinal shrinkage on account of the presence of cross grain. A knot along one edge of a narrow strip will cause bowing of the strip due to the cross grain which it produces.

7 *Long Pieces of Wood.* Although a long piece of wood should not shrink any more than a short one of the same kind on a percentage basis, the actual shrinkage is proportional to the length. The greatest shortening effect therefore, would be expected in long boards, planks, timbers, or the like. The present tendency toward increased use of short lengths will, therefore, help to break up the longitudinal shrinkage into smaller units, which will be particularly advantageous in keeping down the width of openings in butt joints in siding and flooring.

RELATION OF LONGITUDINAL SHRINKAGE TO MOISTURE CONTENT

No attempt has been made by the author to determine the moisture content at which wood begins to shrink longitudinally in drying. It is hardly conceivable that it would be any different from that at which transverse shrinkage occurs; namely, the fiber-saturation point, which averages about 24 per cent moisture.



FIG. 3 BOWING OF LOBLOLLY PINE AS A RESULT OF EXCESSIVE SHRINKAGE OF COMPRESSION WOOD (DARKER PORTION) ADJACENT TO NORMAL WOOD

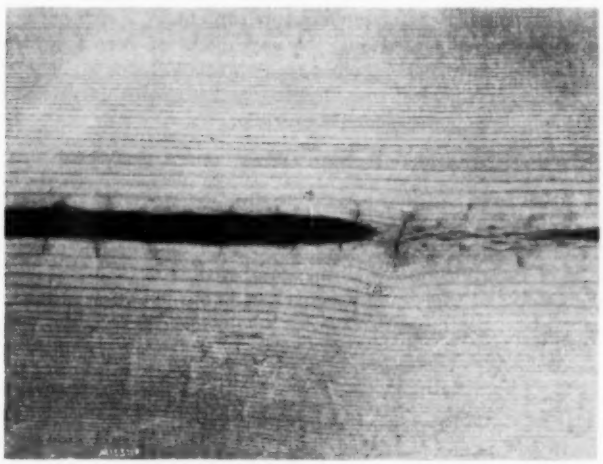


FIG. 5 CROSS BREAKS IN WESTERN YELLOW PINE DUE TO GREATER LONGITUDINAL SHRINKAGE OF WOOD NEAR PITH

Although most of the points representing shrinkage for different percentages of moisture content in Fig. 1 are not in a straight line or characteristic type of curve for each kind of wood, they agree fairly well in pointing to about 24 per cent as the moisture content at which shrinkage began.

Among the specimens of low shrinkage there is a slight tendency to curvature in the shrinkage lines, indicating that the shrinkage rate increases as the moisture content decreases. Some of the specimens showed no shrinkage or even elongated in the first and second stages of drying, but when all moisture was removed they were shorter than when wet.

Longitudinal shrinkage, like transverse shrinkage, depends on the change in moisture content below the fiber-saturation point. If wood is dried so that it will not change a great deal in moisture

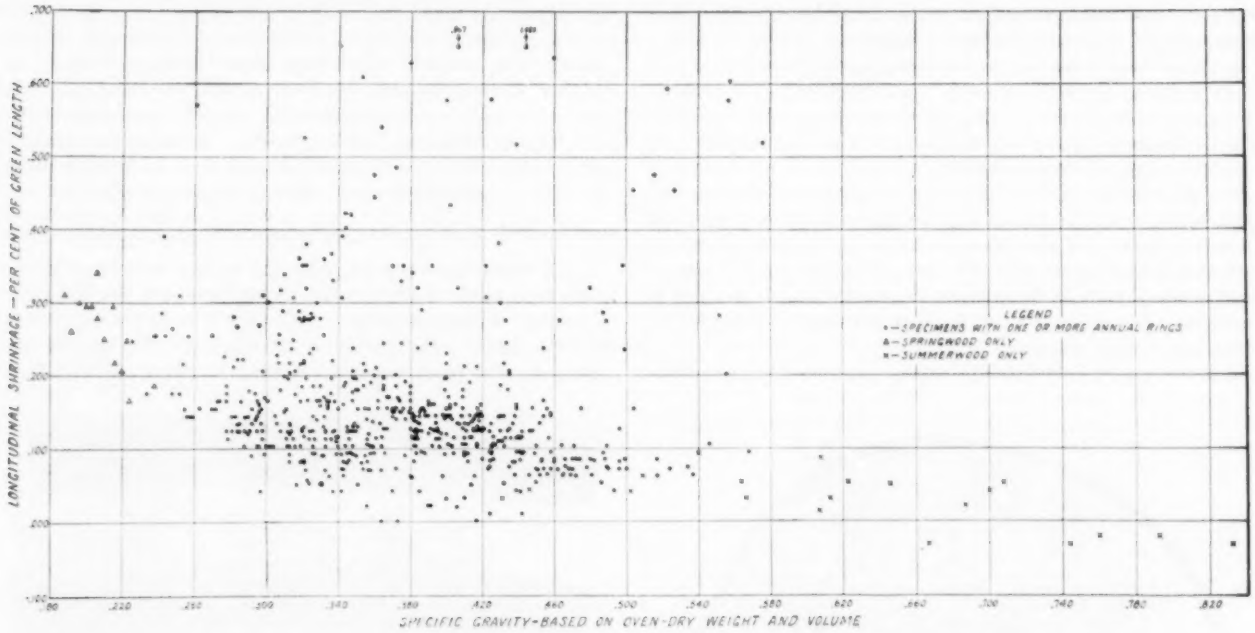


FIG. 4 RELATION OF LONGITUDINAL SHRINKAGE OF REDWOOD TO ITS SPECIFIC GRAVITY

content when used, then it will not shrink appreciably along the grain, no matter how abnormal its shrinkage potential may be. For uses in which the wood becomes alternately wet and dry, previous drying, of course, does not prevent subsequent changes in dimensions with changes in moisture content.

As previously pointed out, the uses discussed here in which longitudinal shrinkage becomes an important factor are far in the minority; for most uses the longitudinal shrinkage of wood is so small as to be considered negligible.

FUTURE WORK

Further investigations are under way at the Forest Products Laboratory to determine how the cellular structure of wood affects its shrinkage. It is believed that information on the cause of variations in the shrinkage of wood will be of value in diagnosing and explaining peculiarities in the behavior of wood, and in learning how to recognize wood which has an exceptionally large shrinkage potential, so that difficulties that might result therefrom may be guarded against. It is also believed that an understanding of the mechansim of wood shrinkage will be of value in developing treatments for reducing the shrinking and swelling of wood in use. It has already been discovered that the pitch of the spirally wound cellulose fibrils of which the cell wall is composed is one of the controlling elements in longitudinal shrinkage, the less the pitch the greater the shrinkage in the length of the cell. Theoretically, the reverse should hold for transverse shrinkage, and it was found to do so in many cases. For example, compression wood shrinks less than normal wood, and springwood shrinks less than summerwood across the grain. Evidently other factors also come into play in transverse shrinkage, the importance of which should be determined by the further investigations now under way at the Forest Products Laboratory.

Curved and Bent Plywood

By THOMAS D. PERRY,[1] NEW ALBANY, IND.

Curved surfaces on solid wood reveal unattractive wood grain and are seriously lacking in strength and rigidity. The veneering of curved surfaces on solid wood is an ancient art. A description of the curving of plywood for modern designs is timely, since curved surfaces are passing through a period of style popularity. The paper gives a concise description of the building of curved plywood, as well as of the bending of plywood by processes just emerging from the experimental stage. Suggestions are given for the necessary tolerances in size and shape, the trimming and sanding of the curved shapes, and the accessory equipment required therefor. Attention is called to the fact that, pound for pound, wood is the strongest material known; hence curved plywood has an opportunity to become an industrial product with wide applications and substantial economic value.



FROM time to time curved veneered surfaces have been conspicuous in wood-product designs, and just at present they are unusually popular. There are several approved methods of producing such curved veneer and plywood; methods that depend on the quantity, on the complexity and abruptness of the curve, and on the strength and rigidity required. The three best developed processes are as follows:

(1) Curved veneering on solid lumber
(2) Building curved plywood cold (with rigid male and female dies) during gluing
(3) Bending plywood with heat after gluing.

Historic Methods. Surviving examples of early artistic veneering consist almost wholly of single-ply veneer glued by hand, practically the first of the aforementioned processes, on flat or curved foundations of an inert and stable wood like pine or poplar. This veneer was applied with what is known as a veneering hammer (see Fig. 1), which really was a hand pressing tool drawn or pushed over the surface of freshly glued veneer to produce complete adhesion and to squeeze out the glue, while making the close and tight surface joints required in high-grade veneering. Such veneered work has endured for hundreds of years, and has not only been proved durable by time, but also has been judged genuinely artistic by leading designers of past and present wood products.

Mechanical Limitations. Given plenty of time and skilled artisanship, it is possible to attach veneer to almost any curved surface. Even long circular columns and complicated re-entrant curves can be veneered. As a practical manufacturing problem for quantity production, however, only such curves can be successfully and economically veneered as can be formed under platen presses with vertical pressure. There is a definite limit to the curved segmental angles that can be glued effectively. Reference to the pressure diagram in Fig. 2 will indicate that, with a direct pressure A of 75 to 100 lb. per sq. in., a 90-deg. segment will have approximately two-thirds of this pressure at its extremities, while one of 120 deg. will have only one-half of this pressure, as at C. It is possible to veneer the latter successfully, and it will include sufficiently more than a quadrant to produce grooved and well-fitted corners. It is obvious that a segment of 180 deg. would have no vertical pressure component, and its only pressure would be derived from the snugness of the forms. This limit of 120-deg. segmental angle applies to both simple and compound curved surfaces. Bending machines, under the third process, may raise this limit to 240 deg.

No definite size limits can be stated as conditions vary so widely. In general, curves more than 2 ft. wide (around the curve) and more than 2 ft. long (parallel to the cylindrical axis) are quite exceptional and are rarely required on a quantity-production basis. Hence their methods of manufacture have little practical background. Grand-piano rims are invariably made of curved plywood and are an exception to this classification. Probably 90 per cent of curved plywood falls within the foregoing size limits, and the methods outlined will be particularly applicable to this majority portion.

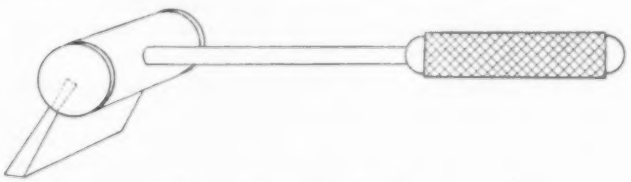


FIG. 1 VENEERING HAMMER
(Used for pressing veneer to base and squeezing out surplus glue.)

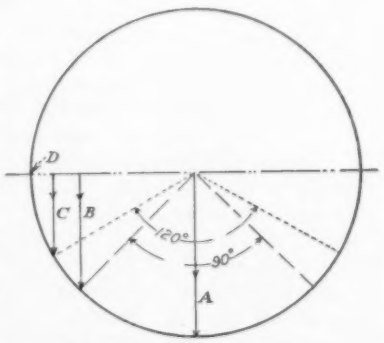


FIG. 2 PRESSURE DIAGRAM
(Indicating distribution of vertical pressure for various sized segments.)

Glue Requirements. The principal wood adhesives of commerce are each adapted to certain kinds of curved work, but none of them is universally applicable to all kinds of bent and curved plywood.

Successful plywood glues for use in curved products must hold their fluidity well—i.e., must not evaporate too rapidly in bulk or while spread, waiting for pressure; at the same time they must set quickly under pressure, so that gluing equipment can be used repeatedly during the daily working schedules. Ordinarily such quick-setting glues are low-moisture-content

[1] Works Manager and Engineer, New Albany Veneering Company, New Albany, Ind. Mem. A.S.M.E. Mr. Perry has devoted many years to engineering in the wood-industries field and has served as vice-president and manager of the Grand Rapids Veneer Works and as consulting engineer with Bigelow, Kent, Willard & Company, of Boston, Mass. He is chairman of the Wood Industries Division of the A.S.M.E., and the author of many articles dealing with lumber-handling equipment, woodwork, and engineering practice with veneer, plywood, and solid wood.

Presented at the Fifth National Wood Industries Meeting, New York, N. Y., Oct. 16 and 17, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

brands, requiring from 66 to 75 per cent of the normal amount of mixing water. They are usually rather above average quality and price. Ordinary glues or cheap glues are hazardous on curved and bent work.

A satisfactory working schedule is three hours under pressure or about four hours between successive gluings. This permits from two to six charges, according to the length of the working day. Where 1000 units are required daily, and the forms cost $10 per set, the charges per day become an important economic item.

Animal glue was the original standard for veneer work, but

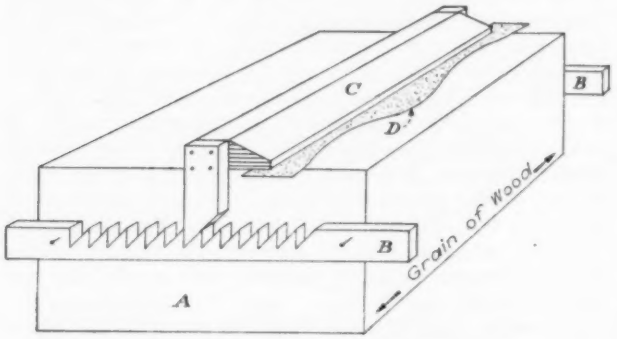


FIG. 3 MARKING TEMPLET, WITH METAL EDGE
(*A*, block of laminated wood; *B–B*, racks for spacing; *C*, templet holder, of plywood; *D*, metal templet.)

is of such a cost that its principal application is to the curved tasks where skilled labor is required and volume is very moderate. It is an excellent manual glue, but after setting it is neither moisture proof, nor does it resist the action of heat above 140 deg. fahr. It is the only one of the glues mentioned that requires heated wood and 80 deg. fahr. workroom temperatures for its successful application.

Vegetable glue is made from a starch base, usually of casava or tapioca flour, and is especially adapted to a large volume of standardized production. It is moderate in price, and when properly blended can be removed from pressure in a minimum of two and one-half hours, although three hours is safer. Due to its heavy, tacky qualities, it must be mixed and spread mechanically. It has no material advantages over animal glue in either moisture or heat resistance. Room and product temperature may be normal workroom conditions.

Casein glue is a milk product that can be mixed and applied cold. It has water- and heat-resistant qualities and is suitable to use in dry-bending operation with a short exposure to temperatures around 200 deg. Its water-resistant qualities depend on lime content, but high water resistance requires a large lime admixture and it becomes very brittle. It must be machine-mixed, but can be spread manually or mechanically. The best grade for dry bending is a low lime content, and the plywood should be bent within 24 to 48 hours from laying. Casein costs are about midway between animal and vegetable glues.

Blood-albumin glue is a packing-house product and consists chiefly of dried blood. Its use is fully described under the wet-bending process later in this paper.

CURVED VENEERING ON SOLID LUMBER

Fundamental Principles. The fundamental principles of success in this process of curved veneering are three: first, the selection of wood of such quality as will tenaciously hold glue and serve as a suitable foundation for the thin, fragile sheets of pliable veneer to be attached thereto; second, the preparation of thoroughly stable and inert bases of suitable curved shapes with sufficient strength to carry the strains of hard usage in a piece of furniture; third, gluing the veneer to such carefully selected and designed bases with proper regard to figure, joints, and freedom from surface breaks.

In the past such work was ordinarily glued with animal glue, but modern practice approves several other more recently developed adhesives.

Preparing the Block. The curved bases on which the veneer is to be glued are most advantageously made from large blocks of wood, band sawn to the desired shape. These blocks of wood are preferably laminated out of 4/4 in. stock and must be thoroughly kiln-dried, as well as square on every face, in order that the marking and saw cuts may be free from irregularity and imperfection. The grain of laminations is usually parallel without cross-banding, since the cross-veneer increases the difficulty of sawing, but does prevent warping. Such blocks must be of a convenient size for ready handling on the band-saw table and must be of such dimensions that the band saw will readily cut without binding, running, or whipping in the cut.

It is possible to make successful band-saw cuts as deep as 22 in., but accuracy of form and speed of cutting are sacrificed in such thick blocks. It is much better practice to confine the thickness of such blocks to a range between 6 and 12 in., permitting more rapid sawing and eliminating to a large degree the tendency toward irregularity between the desired curve at the bottom and top of the block.

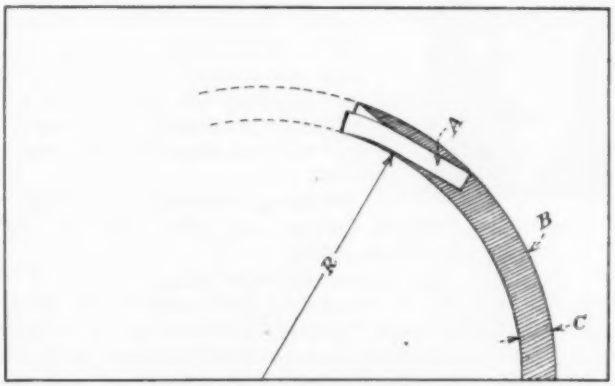


FIG. 4 LIMITING CONDITIONS FOR BAND SAWING
(*A*, section of band-saw blade; *B*, saw cut from which kerf has been removed; *C*, must equal veneer in thickness; *R*, radius of saw cut, inside.)

Several woods are in current use for blocks of this kind, and each has its special qualities. White pine is ideal in that it works easily, glues excellently, and retains its shape. It is, however, decidedly costly, as well as rather soft and easily dented. Yellow poplar is not quite as stable as pine, but is denser and rather harder. Sound wormy chestnut makes an excellent block and is, from the standpoint of economy, stability, and gluing qualities, one of the best blocks available. It is largely used by discriminating manufacturers of curved veneered parts.

Templets for Marking. If only a very few pieces are to be produced, it is of course possible to scribe the curve directly on the block with drawing instruments or to make wooden templets for a limited number. Quantity production demands standardization of shape, and a method of utilizing metal templets is shown in Fig. 3, in which two exactly similar racks *B-B* are nailed to either end of the block and serve as spacing guides to insure uniformity of thickness in the sawn curved bases.

For moderate production quantities the metal templet *D* can be made from zinc, brass, or aluminum and sawn to shape on the woodworking band saws. For large quantities and unusual

accuracy this templet should be made of steel. It is obvious that great care must be exercised in making this metal templet, as it must be true to curve, reversible, and free from irregularity.

It is recommended that the templet holder *C* be made of multiple plywood so that its shape may not be seriously altered by climatic conditions.

Sawing the Curves. There are a number of sturdy makes of band-sawing machines that are rigidly built, easily adjusted, and conveniently operated. The best types have electric motors built into saw frames and are equipped with ball bearings.

The problem of selecting a suitable band-saw blade is subject

TABLE 1[1] STANDARD SAW SIZES

Width of saw, in.	Points to inch	Standard Specifications: Thickness, Inch	Thickness, Gages	Minimum radius[2] to be cut, in.
1/8	8	0.028 or 0.025	22 or 23	3/8
5/32	7	0.032 or 0.028	21 or 22	1/2
1/4	6	0.032 or 0.028	21 or 22	3/4
5/16	5 1/2	0.032 or 0.028	21 or 22	1 1/2
1/2	5	0.032 or 0.028	21 or 22	2 1/4
5/8	4 1/2	0.032 or 0.028	21 or 22	3
3/4	4	0.035 or 0.032	20 or 21	4 1/2
7/8	4	0.035 or 0.032	20 or 21	6
1	3 1/2	0.035 or 0.032	20 or 21	8
1 1/4	3 1/2	0.042 or 0.035	19 or 20	12
1 1/2	3	0.042 or 0.035	19 or 20	20

[1] From Disston's Tool Manual (1927).
[2] In ordinary construction there are few exterior curves less than 1 1/2 in. in radius or few interior curves less than 3/4 in.

to a number of considerations. An exaggerated example of the limiting conditions is shown in Fig. 4, in which the set of the saw teeth is clearly illustrated. The saw teeth must be set to produce not only a reasonably smooth surface, but to also properly clear and relieve the heel of the cut of the saw in a cut of the radius required. Standard saw sizes are indicated in Table 1, published in Disston's Tool Manual (1927). These specifications, however, particularly apply to band sawing ordinary 4/4 in. to 8/4 in. lumber. On thicker blocks it will be necessary to reduce the points per inch

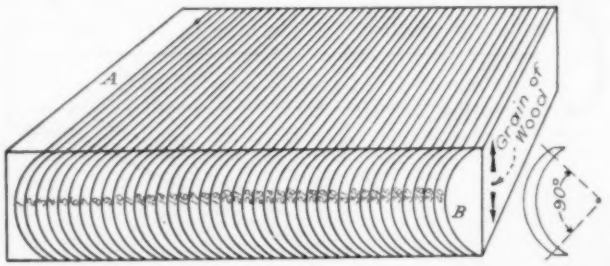


FIG. 5 BLOCK SAWN INTO SIMPLE CURVES
(Showing the sawn segments numbered and laid together in original sequence. *A*, top segment, or male form; *B*, bottom segment, or female form.)

to provide additional sawdust-removal capacity. The width of the saw cut is determined by several important factors, as follows:

(*a*) Gage or thickness of the saw
(*b*) Set of the saw teeth
(*c*) Clearance of the heel of the saw
(*d*) Thickness of veneer to be applied.

The ideal amount of lumber removed in saw kerf should be equal to the thickness of the veneer to be inserted and glued to the sawn surfaces. The width of the saw blade should be as great as possible for strength purposes with due regard to desired radius and necessary width of the cut. The gage of the saw depends partly on the size of wheel used, i.e., thinner saws on smaller wheels.

Authorities vary as to the proper lineal cutting speed of the band saw, but the best authorities recommend from 3000 to 5000 lin. ft. per min. An adequate range of speeds, wheel sizes, and revolutions per minute are given in Table 2.

TABLE 2 OPERATING SPEED OF BAND SAWS

Lineal speed of saw, ft. per min.	Diameter of wheel in inches: 20	24	30	36	42
3000	573[1]	477[1]	382	318	273
3250	621	517	414	345	296
3500	668	557	446[1]	371	318
3750	716	597	477	398	341
4000	764	637	509	424[1]	364
4250	812	676	541	451	387
4500	859	716	573	477[1]	409
4750	907	756	605	504	432
5000	955	796	637	531	455[1]
5250	1003	836	668	557	477
5500	1050	875	700	584	500[1]
5750	1098	915	732	610	523
6000	1146	955	764	637	546

[1] Recommended speed for wheel (Disston's Tool Manual, 1927).

Formula:

$$\text{R.p.m.} = \frac{\text{Lineal speed}}{(D + 12) \times \pi} = \frac{\text{Lineal speed}}{\frac{D \times \pi}{12}} = \frac{\text{Lineal speed}}{0.262 \times D}$$

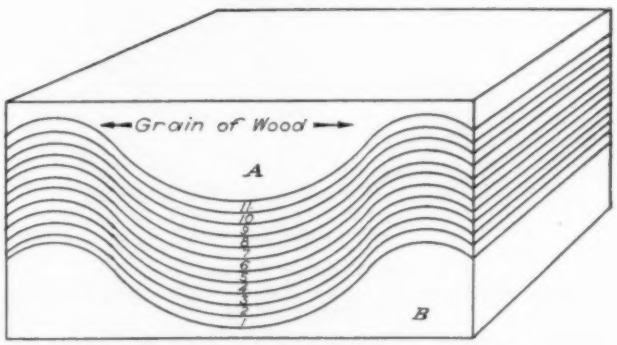


FIG. 6 BLOCK SAWN INTO COMPOUND CURVES
(Sawn sections numbered and laid together in original sequence. *A*, top segment; *B*, bottom segment.)

One manufacturer, making a large volume of sawn curved shapes, has adopted as most satisfactory 620 r.p.m. on a 36-in. band saw, using a 3/8 in. wide, 3-point, No. 19 or No. 20 gage band saw. This gives a saw travel of approximately 5800 lin. ft. per min., indicating that the recommendations in Table 2 are quite conservative.

Laminated lumber blocks, band sawn to simple and compound curves, are shown in Figs. 5 and 6. It is to be noted that when the simple band-sawn curves are nested, the thickness at the center will be slightly greater than at the edge, due to the crescent shape, resulting from the moving center and fixed radius. It is therefore customary to select and use the desired radius for either the inside or the outside of the curved pieces, whichever requires accuracy, and to allow sufficient thickness at the thinnest point. Manufacturing conditions may vary these requirements in certain instances. The same variable thickness occurs in a compound-curved block as a result of the lack of concentricity in the circles. This variable thickness cannot entirely be avoided, but it is seldom a detriment under actual working conditions, as one surface can always be made true and uniform.

The off-bearer at the band saw should be equipped with an efficient hand brush so that all sawdust may be brushed off each successive segment as sawn and the saw table be kept free from any sawdust that might cause tilting of the block during sawing.

Obviously the limiting width along the curve is the depth of the saw cut or about 22 in. If curves of larger dimensions are needed, simple curves may be made on a lathe, but compound curves are exceedingly difficult. Usually such compound curves are made all plywood (rather than veneered on solid lumber), as described under the second process.

Applying the Veneer. In applying the veneer to these sawn

curved shapes, the top and bottom segments (*A* and *B*, Figs. 4 and 5) are used as forms, the veneer being designed to occupy the space between the segments from which the saw kerf has been removed. If it is desired to put $^1/_{28}$ in. face and back on a segment, it will be necessary to use a saw kerf removal of $^2/_{28}$ in., or $^1/_{14}$ in. There are instances, also, where the strain demands a crossband, in which case the thickness of the saw kerf must be increased to permit the insertion of one or two crossbands. More often, however, the segment has sufficient strength for all structural requirements, and frequently sufficient resistance for the strain occurring in the gluing of the veneer to one side of the segment, to permit the use of a single ply of veneer on the face of the segment and none on the back. In such instances the saw kerf need only be $^1/_{28}$ in., although usually somewhat greater in practice.

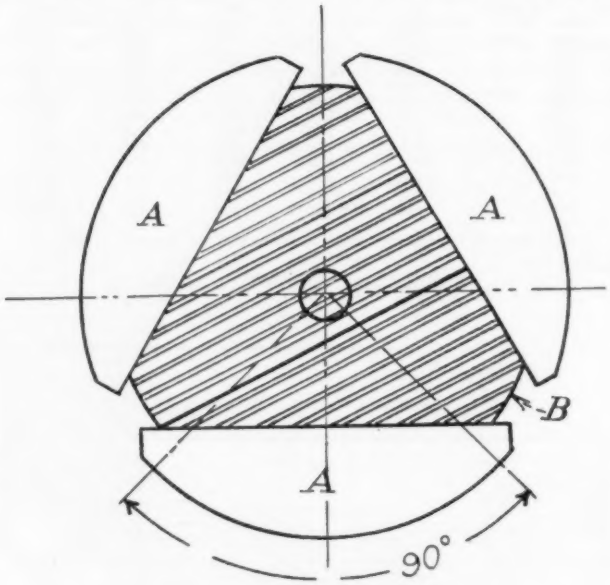


FIG. 7 CIRCULAR SEGMENTS
(Another process of producing circular segments, *A*, *A*, *A*, using a lathe and triangular arbor, *B*.)

The sheets of veneer are usually glued with a mechanical spreader and the wet glued faces placed against the segment, followed by a piece of oil paper and another segment; this is two-ply construction, the base and one thickness of veneer. If a face and back veneer are used, two sheets of veneer would be inserted with unglued sides of veneer against an oil-paper separator. If crossbands are to be used, they would be mechanically glued on both sides and placed between the dry face veneer and the sawn lumber segment, still using oil paper between unglued faces. It is usual to use hand clamps in clamping together the entire block.

The position in which the veneers are laid up will depend somewhat upon the size. In general the vertical pile is quite unstable, and either an angling or horizontal accumulation is preferable.

Gluing With Fabric Bands. There are instances in which the nested form does not meet manufacturing conditions. Frequently, simple curved segments need to be longer than 22 in., the limit which can be band-sawn successfully. Segments up to thirds of a circle can be turned on a lathe, as is indicated in Fig. 7, in which three quadrant segments are shown mounted on a triangular arbor. These quadrants are usually attached at the ends where the hole will be trimmed off after veneering. It is customary to make the back side of these quadrants flat, rather than curved, since the hollowing out of the curved back is rather a difficult task. In the making of bow ends for cabinets or beds these segments usually need to be enough more than a quarter circle to provide a proper lip for framing. The back side is often out of sight and requires only strength and manufacturing convenience.

In the making of these lathe-cut quadrant forms, no female members result, for forcing the veneer in place. Female forms may be made on a lathe and used repeatedly, but ordinarily a stretched canvas clamp is used, as shown in Fig. 8. It is usually advisable to use crossbands in this type of gluing, although some authorities do not favor it. Tension in the canvas band produces sufficient pressure to bring the face veneer evenly against the curved base. The proper designing of clamp lever *E* will permit taking up the slack in the canvas belt as it stretches in service.

This process of clamping veneer to convex surfaces with flexible bands is a principle used variously in veneering round posts and columns with spiral wrappings of tape or bands. This work requires trained artisans of individual ability and is not mechanized to any extent.

BUILDING CURVED PLYWOOD

While gluing face veneer on solid band-sawn or lathe-turned cores is the historic and perhaps the easiest method of producing curved veneered surfaces, it has certain serious limitations, such as uneven thickness, size limits, and often may be of such a shape as to lack strength; i.e., the band-sawn core may split under the strain of usage.

Bending Plywood While Plastic. A process has therefore been developed in which freshly glued plywood may be pressed immediately between curved male and female forms, having equidistant spacing between all parts of the form when the pressure is at its maximum. This process requires an adequate number of rigid forms that are used over and over again, with generous throats or clearances for protruding edges of veneer and sufficient strength to withstand not only the repeated pressure, but also to resist the rough handling of continuous use.

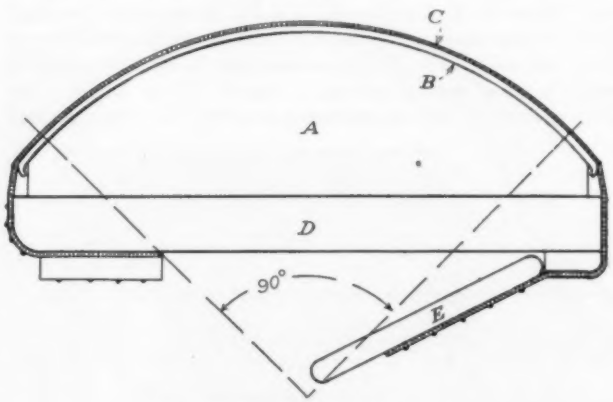


FIG. 8 CANVAS GLUE CLAMP
(To press veneer on lathe segments shown in Fig. 6. *A*, turned segment; *B*, veneer applied, with or without crossing; *C*, canvas clamp band; *D*, clamp block; *E*, clamp lever to hold canvas band tight.)

Veneer Requirements. Plywood bent under this process will consist of a number of laminations of relatively thin veneer, curved so that the grain of the principal laminations will be parallel to the axis of the cylinder represented in the face of the curve. This bending is called "with the grain." Thin veneer, $^1/_8$ in. and less, is preferable, since thicker veneer will show cutting checks. The resistance to bending "against the grain" of the veneer is considerable; consequently, the right-angle laminations must be as few as possible. It is important, however,

that the crossbands under the face and back should be bent at right-angles to the axis of the cylinder (against the grain), as this will add balance to the panel and will largely eliminate the tendency to "wind" which is so prevalent in curved shapes.

A typical design for a $^{11}/_{16}$-in. curved 9-ply door is shown in Fig. 9, in which $^{1}/_{8}$ in. is the maximum thickness and all veneers are laid parallel to the cylindrical axis except the two crossbands directly under the two faces and the center layer of $^{1}/_{20}$ in. The core is made of five layers, not true 5-ply, but an equivalent for a $^{9}/_{16}$-in. lumber core, that could not be bent.

In the case of $^{1}/_{4}$-in. curved plywood, the more conventional flat 5-ply construction can be used, as shown in Fig. 10. The grain of the face, back, and core are all parallel to the curve, with only the two crossbands at right-angles thereto. Adding another $^{1}/_{8}$-in. core, parallel to the face, would make a $^{3}/_{8}$-in. 6-ply.

It is preferable not to design a curved surface so that the face veneer has to be laid against the grain, since the wood permits little stretching or compression endwise. All veneers have considerable flexibility sidewise, due to their methods of cutting.

Face veneers, to be laid in curved forms, require more than ordinary preparation. Center joints need to be strongly taped as well as edge-glued. Cross tapes should be placed at both ends, and not more than 6 in. apart over the whole length. These will do much to prevent surface checking in bending and drying.

Standard commercial veneers will vary somewhat in thickness, due to shrinkage allowances made by veneer cutters, looseness of cutting, and pressure exerted during mechanical drying. The best manufacturers cut their green veneer about 10 per cent over standard, resulting in plump dry veneer. Mixing veneer from several sources is a hazardous proceeding, especially in plywood bending operations.

Male and Female Forms. The building of forms for shaping these all-veneer plywood curves requires careful attention and extreme accuracy, since any variation in the distance between the faces of the two forms under pressure will result in defective plywood. When forms are too close at any point, the pressure will be excessive, and plywood will have flat or hollow spots; when

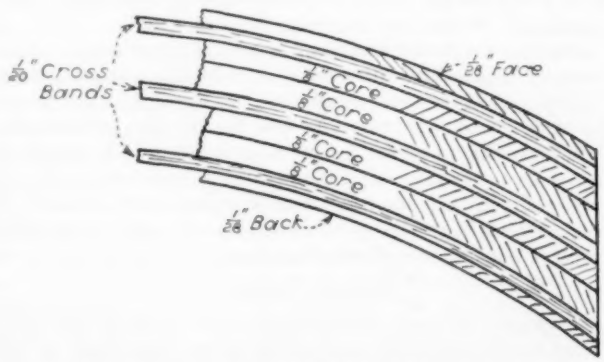


FIG. 9 CURVED DOOR, 9-PLY, $^{11}/_{16}$ IN. THICK
(All laminations bent with the grain, except the three crossbands; i.e., in the cross-section the end wood is exposed in face, back, and all cores, while the side wood is exposed in two outside and one center crossings.)

spots on the forms are too far apart, the veneer is likely to be loose at that point.

Lumber for use in these forms must be dense and hard, to withstand rough handling and to maintain even, smooth-faced curves. Maple and birch are ideal, but gum and poplar can be made to serve. The lumber must be sound and free from defects. End and edge joints, if well manufactured, are permissible, but must be carefully staggered.

Three types of forms are shown in Figs. 11, 12, and 13. In Fig. 11 the forms are made of laminated and crossbanded blocks, of a size and length that could be molded to curved shape on a four-side sticker, with a capacity of 12-in. by 6-in. stock. In order to successfully use this molder process, it is necessary to build these forms in multiples of length, not less than 4 ft. After the molding operation, it is necessary to carefully sand off the faces of the forms, with particular reference to removing any particles of end wood resulting from the cutting of the crossbands.

While it is possible to rely on glued joints to hold these forms together, it is preferable to bolt them firmly, as indicated, with $^{1}/_{2}$-in. bolts. These bolts do much to resist hard usage and to

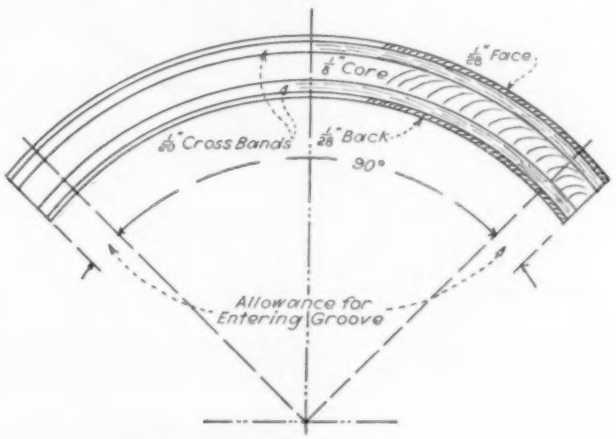


FIG. 10 CURVED PANEL, 5-PLY $^{1}/_{4}$ IN.
(Laminations have normal grain directions of ordinary flat plywood.)

keep the forms from becoming distorted in rough handling.

Authorities do not agree that the crossbanding is an advantage in this type of form. While it does increase the difficulty of obtaining a smooth cut in the molder, it also simplifies the gluing operation and adds much to the prevention of "wind" in the ultimate male and female forms.

The form shown in Fig. 12 is made with the grain of the wood running across the form rather than lengthwise, as in Fig. 11. The sections of this form can be roughed out on a band saw and brought to a smooth true curved face on a shaper. Some authorities advise gluing and bolting these laminations together, and others rely on bolts only, as indicated. There are instances, however, where a shaper might not be available, and such forms as shown in Figs. 12 and 13 might be made with a band saw and sanded after bolting together.

To secure accuracy in resulting curves, the distance between the forms is usually the thickness of a single unit of plywood, so the plywood curves will be uniform. There are cases, however, where plywood is relatively thin and accuracy is not so essential, so that the distance between the forms may be equal to two or three units of plywood. These units will have slightly varying curves in which the variations become less and less as the radii of the curves increase.

It is important in building male and female forms to provide sufficient and adequate clearances for excess veneer edges (to be trimmed off) and also wide enough throats at each side so that the surplus size of the veneers will not interfere with proper clamping under pressure.

Placing Veneer in Forms. Forms of the type shown in Figs. 11, 12, and 13 should be scraped and sanded smooth after bolting and thoroughly waxed before using. They should always be lined with waxed or oiled paper at each insertion of wet glued plywood to prevent the adherence of glue particles. Such glue particles should be immediately cleaned off, but at best are likely to mar the surface of the form.

Plywood to be bent in curved forms is glued and laid up as in ordinary flat plywood, but with two sheets of waxed or oiled paper between each unit. The units are carefully placed and centered in the lower concave form, with the convex form placed and centered above. Usually three sets of forms can be placed in a press retainer with flat caul boards between sets.

It is obvious that curved forms must be approximately balanced

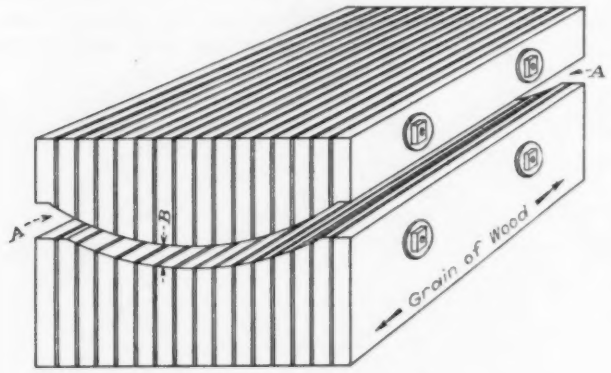


FIG. 11 MALE AND FEMALE FORM

(Curves cut on a sticker or molder, from blocks glued together with crossbands. Forms are bolted to resist hard usage. End grain of wood exposed in illustration. *A*, clearances for excess veneer to be trimmed off, sometimes called throat; *B*, net thickness of single plywood unit.)

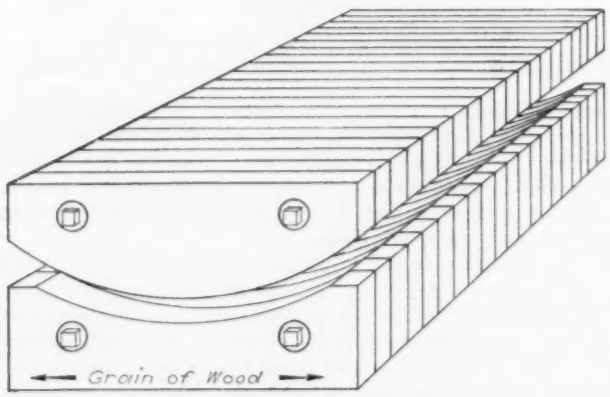


FIG. 12 FEMALE FORM

(Curves band-sawn and finished on a shaper. Laminations bolted together after shaping.)

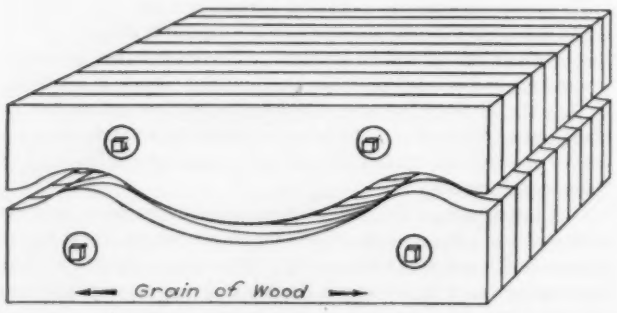


FIG. 13 MALE AND FEMALE FORM

(Curves cut on band saw, bolted together, and sanded smooth after bolting.)

on a center line in order that the wet plywood may not slip or slide under pressure. Attempts to anchor or provide guides for unbalanced forms have not been at all successful.

Multiple Forms With Crescent-Shaped Curved Cauls. There are instances where curved plywood may have shapes of approximate equality, such as opera-seat backs. In such instances it is found economical to design a series of male and female forms as shown in Fig. 14. Half a dozen plywood units may be placed in each opening, with curved cauls or separators between adjacent lots. These curved cauls are sufficiently crescent shaped to insure equal pressure at all points of the curved-plywood units placed between them. While it is conceivable that such multiple cauls could be made of wood, it has been found desirable to utilize the multiple system principally in those instances where production will permit the careful machining of metal cauls to true shapes. Such multiple cauls are more expensive than the single cauls, but they offer advantages in production where large quantities are required.

Drying Forms. The forms described have been those primarily designed for forcing the freshly glued plywood into the proper shape and holding it there for the initial set of the glue. Some manufacturers find that it is advantageous to provide sets of skeletonized forms for drying purposes, into which the curved plywood can be placed after removal from the gluing forms and rigidly held to shape during the complete drying out of glue water. These skeletonized drying forms will of course permit greater access to air currents and quicker drying, and will hold the curved form in the proper position until the drying out of the glue water has been completed. This practice is particularly necessary when curved shapes are large in area and relatively thin.

A common example of this practice is the gluing up of grand-piano rims, which are rather complicated pieces of plywood. Laminations are usually parallel without crossbanding, tending toward warping. The curve is quite complex, and drying forms are desirable.

Drying forms are sometimes used for curved veneer pieces, made from band-sawn, solid-lumber shapes, as described earlier in this paper. They are only necessary when a large amount of glue water needs to be dried out without distortion.

The use, however, of quick-setting, low-water-content glues has reached such a point that, in most instances, three hours in a gluing form will give a sufficiently permanent set, so that there is a minimum of distortion during subsequent drying. (See paragraph "Design and Inspection Tolerances.")

Crowned or Dished Plywood. It is possible to give a slight crowned or dished effect to plywood made by this process, in which the resulting plywood represents a slightly spherical shape. Such crowned plywood is frequently used for the tops of wardrobe-trunks, but if the crown is more than nominal, it is frequently necessary to remove narrow triangular sections from the different plywood layers to prevent crowding or lapping under pressure. A slight dish is usually given to veneered chair seats by the same process, but without removing segments.

BENDING PLYWOOD

Principles Involved. The bending of any piece of solid wood or of plywood involves compression of the wood fiber on the concave side and tension on the convex side. In other words, the fibers of the wood must all yield to a greater or less extent, either in compression or tension, except at the neutral axis.

The curving of plywood, previously described, was done when the glue was wet and the tension and compression had become slippage between the layers of veneer, and thus adjusted itself before the initial setting of the glue. The present problem of bending plywood after gluing differs in that the glue has had its initial set, so that it is necessary to stretch and crowd, not only the wood fibers, but the glue layers as well.

Most materials bend more readily and with less breakage when heated, and this applies to many organic as well as to inorganic substances. In the case of wood, its fibrous nature is an advantage in the bending processes, but heat must be used guardedly,

so as not to char the wood surfaces nor to make the wood brash and brittle.

Methods of Bending Solid Wood. It is essential to briefly mention the two principal methods of bending solid wood, in order better to visualize their application to plywood bending.

The steaming process of bending solid wood has been known for many years, and several approved methods have been followed in such products as agricultural implements, automobile and buggy tops, chair backs, wheel felloes or rims, and the like. Modern methods in this art, with attendant research, were fully covered in a paper on "Wood Bending," by T. R. C. Wilson, of the Forest Products Laboratory, published in Transactions of the American Society of Mechanical Engineers, Wood Industries Division, September to December, 1929. The problems there presented and the results accomplished are related almost wholly to the bending of steamed wood against the grain. One of the important fundamentals was the securing of the proper end pressure during the bending operation.

The dry-wood bending process is a comparatively recent development, not yet entirely beyond the experimental stage, but of proved practical worth. It is a process of selecting straight-grained wood, with a moisture content between 15 and 20 per cent (not kiln dried), and rolling it lengthwise between heated cylinders, in much the same way that sheet steel is rolled to produce curved boiler plates. It is obvious that some moisture will be developed and evaporated from the lumber itself, but no additional steam or moisture is applied as a pretreatment, hence the term, "dry bending." End pressure is not a matter of importance in dry bending.

Both the wet- and the dry-bending processes are applicable to plywood, but with considerable modification.

The Haskelite or Wet Process of Bending Plywood. This process of bending flat plywood into curved shapes in heated forms was named after its inventor and was first applied to the construction of plywood canoes, made from veneer with highly water-resistant blood-albumin glue. During the stress of the war period the process developed rapidly because of the suitability of such curved waterproof plywood for airplane construction. Commercial development since the war has been largely along the line of shaped and formed plywood for bus bodies and interiors, as well as for other curved industrial uses, where waterproof plywood quality was essential and where the cost of this process was not prohibitive.

In general the process is one of gluing plywood from sheets of veneer not more than $^1/_{16}$ in. thick, using blood-albumin glue as an adhesive. Blood albumin makes a two-stage glue; that is, the initial set results from the usual evaporation of glue water, and the final set is one caused by coagulation under heat. After the initial set of the glue in the flat plywood, the entire panel is soaked or steamed until the wood itself is highly pliable. This soaking can be accomplished without loosening the glue line, due to the qualities of the blood-albumin glue. The wet, pliable veneer is then placed between steam-heated male and female platens of the desired shape and forced into the curve desired under a temperature running between 160 and 220 deg. fahr. When this heat reaches the blood-albumin glue, it produces the coagulation or secondary set, which gives the plywood a permanent curve of the desired shape. The steam-heated platens also dry out the moisture in the plywood. This process is closely protected by patents and is not extensively practiced, but the essential steps are fundamentally as outlined. The resulting plywood is an excellent but costly product.

Dry Process of Plywood Bending. The wet process of plywood bending is limited to the use of blood-albumin glue, and cannot be accomplished with animal, vegetable, or casein glues. The dry-bending process, however, is fully as applicable to plywood as it is to solid wood, and with essentially the same equipment.

The roller bending machine is shown in outline in Fig. 15, in which cylinder *D* is rigidly held in position at its bearings, which do not require vertical adjustment. The interior of this cylinder is heated through perforations in an illuminating-gas pipe *E*, extending the whole length of the cylinder. The diameter of this cylinder may vary somewhat, according to the radius of the material to be bent, although the range of radii that can be bent on the one size cylinder is considerable.

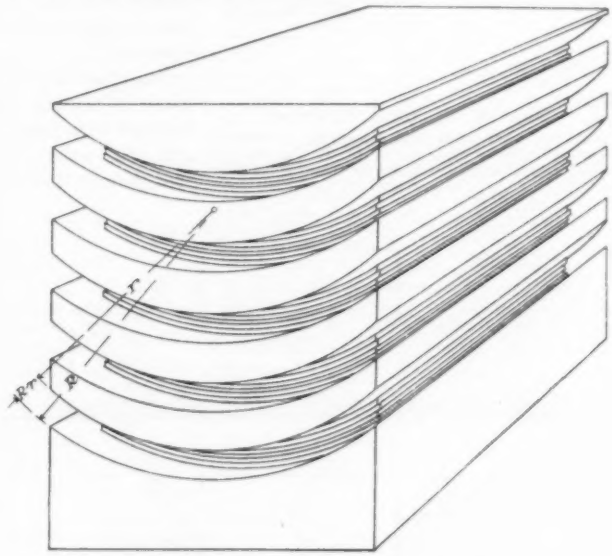


FIG. 14 MULTIPLE MALE AND FEMALE FORMS

(Producing approximately similar shapes. Preferably made of metal, R, outside radius of largest unit; r, inside radius of smallest unit; $r + R/2$ average radius; $R - r$, distance between cauls.)

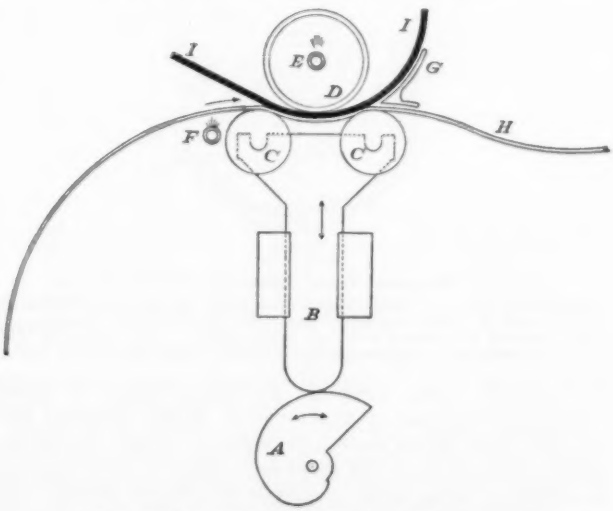


FIG. 15 ELLIOTT DRY-BENDING MACHINE

(*A*, pressure adjusting cam; *B*, adjusting bar; *C-C*, lower roller supports; *D*, upper heated roller; *E* and *F*, perforated gas pipe; *G*, guide; *H*, flexible metal band; *I*, plywood under bending.)

The pressure exerted on the plywood to be bent is regulated by cam *A*, which in turn forces the idle rollers *C-C* closer to the live roller *D*. Between idle rollers *C-C* and live roller *D* is an endless steel belt *H*, usually made in strips not over 6 in. wide. This belt serves to convey the plywood under live roller *D* and over idle rolls *C-C*. In some instances idle rolls *C-C* are heated with internal gas lines, and in other cases the heat is

applied at *F* under the moving steel bands. The plywood must be kept hot on both sides. The curved guard *G* is to keep the bent plywood from following the steel band and is adjustable; in operation it should touch the curved panel as it leaves the rolls. The piece of plywood to be bent is indicated at *I* in heavy black line. These bending machines are made in several sizes and, due to the slow speed of operation, are usually designed in duplicate or quadruple units, to take two, four, or more pieces of plywood simultaneously, thus economizing on the time of the machine operator.



FIG. 16 FLEXIBLE PLYWOOD

(Face and back bend with the grain. Center is made as thin as possible, 1/16 in. to 1/24 in., to offer a minimum resistance to bending against the grain.)

Great care must be taken in feeding the flat plywood squarely into the rollers, so that there will be no faulty spiral or wind resulting in the machine-curved pieces.

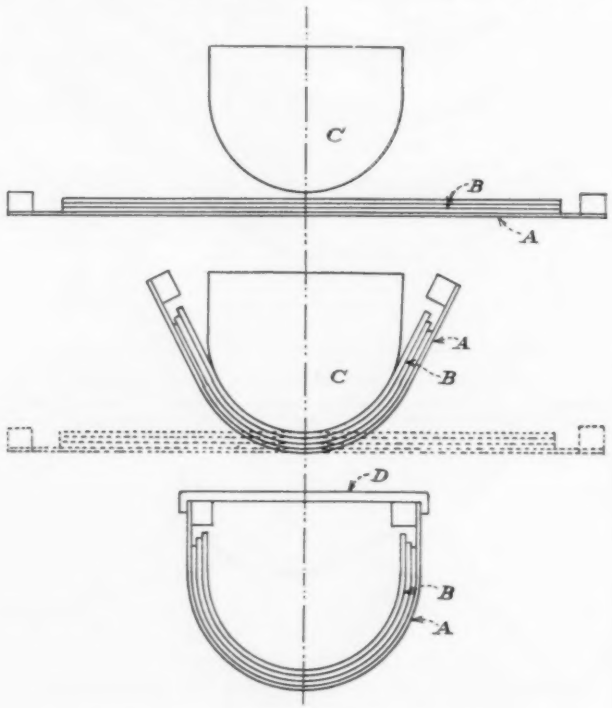


FIG. 17 BENDING WITH FLEXIBLE METAL BANDS

(Upper view shows plywood ready to be bent; center, plywood partly bent; lower, plywood completely bent, removed from the machine, and bands clamped as a drying rack. *A*, metal band; *B*, plywood to be bent; *C*, forming die; *D*, clamp to hold plywood *U* in shape until dry.)

Bending machines of this type are usually equipped with variable speed drives, since it is desirable to heat the machine gradually and to make the speed dependent on the temperature available, as well as on the thickness and moisture content of the plywood to be bent.

It is impossible to do bending of this character in a rapid manner, since the yielding (stretching and compression) of the wood fibers and glue strata must be gradual and without sudden jolts and jars, which would shatter the wood or tear it away from the glue lines.

Some authorities advise machine-bending plywood before sanding, although the lines of paper tape may make some indentations which will sand out. Others advise bending after sanding, in which instance some type of efficient cleaner must be attached to keep all small particles of dust away from the heated rollers and steel belts, and thus prevent denting or marring the surface of the plywood. Where the finished face of the panel is on the convex side, it is highly desirable to leave the tape in place and add its restraining influence to prevent either the cracking of the face veneer in stretching or the opening of the joints. Where the finished face is on the concave side, the absence of the paper tape will do no harm.

Machine bending under this process requires as high a temperature as can be used without either burning the wood or destroying the adhesiveness of the glue. In general the best success has attended the use of casein glues, which are low in lime content and therefore not as brittle as the more thoroughly water-resistant brands. The most successful bending is done within two days after gluing; further delay increases the hazard of tearing the plywood on the glue line. Animal glue will not endure such high temperature, and vegetable glue has not proved satisfactory.

It is desirable to have drying or tempering racks in which to place the bent panels after machine bending and during their cooling. Any tendency of the curves bent on this machine to straighten out occurs within this cooling period, and if the bent plywood is held true until cool, it will seldom change shape afterward.

It is a peculiarity of this type of bent plywood, as well as a distinct advantage, that it is more limber and more easily forced into mechanically shaped grooves than is plywood curved in the press while the glue is wet. Curves up to 240 deg. may be bent under this process.

The use of this bending machine results in substantial economy in utilization of material, since much narrower trimming allowances are required than in the case of plywood curved in forms while wet. On small panels this saving may be as much as 10 per cent.

The dry-bending process for plywood is too new to permit any comprehensive idea of its range of usefulness. It is entirely successful on 1/4-in. 5-ply panels and has been operated without serious difficulty on plywood up to 1/2 in. in thickness where all of the layers were of veneer rather than solid lumber. Whether it will operate with equal success on thicker plywood or plywood with lumber cores is a problem yet to be determined.

The development of bending machines of this type is in its infancy, the principal pioneer being C. W. Elliott, of Columbus, Ind., whose machine is described in the foregoing. The possibilities of further improvements along this line are decidedly challenging.

OTHER MINOR PROCESSES

Flexible Plywood. Flexible plywood is another type of construction in which only 3-ply can be used. The core or center is very thin, with the grain across the curve, as shown in Fig. 16. Two faces make up the required thickness with the grain parallel to the curve. Such flexible plywood must be made of very loose-cut veneer and is not suitable for figured veneer or highly finished. It has been extensively used for curved sleigh dashes, horns for phonographs, and other curved surfaces where flexibility is essential and appearances are not important. The principle of this flexible plywood is found in the 5-ply core of the curved door shown in Fig. 9.

Bending Plywood With Flexible Metal Bands. In Mr. Wilson's paper, referred to in the early part of this article, Fig. 8 illustrates a rim-bending machine particularly adapted to the bending of solid-wood wheel rims and other semicircular shapes. An outline of this process of bending, applied to plywood, is indicated in Fig. 17, the upper portion showing three strips of veneer, with either parallel or angling grain, laid on a thin metal band. The action of the machine is such as to draw this thin metal band and freshly glued plywood around a sturdy, rigid form resulting in

the bending of the plywood to the shape of the form or die. It is arranged so that a clamp or double hook can be placed across the U-bend, to hold the curve in proper shape until it has been thoroughly dried and the curve fixed in position. This type of plywood bending is limited in scope, but is applicable to the bending of plywood rims and subrails for circular table tops. It was in extensive use when round dining-room table tops were the prevailing style. The machine is expensive to install and operate and requires costly dies for the different curves; hence it is only suitable for large production schedules of standard curved units.

Design and Inspection Tolerances

Mechanical uniformity in curved plywood shapes, even those most accurately made by any of the processes described, can be only approximate. Physical differences in the lumber and veneers, as well as inevitable variations in manufacturing procedure, will make some curves subsequently flatten out or curl up more than others. Curves piled in a hot or moist place may change shape more than those piled in a cool or dry place. Designers and users of curved shapes should therefore make due and reasonable allowances for such unavoidable variations. Thin curved plywood in the ultimate product should be set in reasonably rugged frames or be firmly attached to a sturdy member, permitting a reasonable amount of forcing into place.

Inspection tolerances should be drawn with reasonable liberality, since too rigid limits will cause a high percentage of rejections and influence cost prices.

Trimming and Sanding Curved and Bent Plywood

Trimming Ends and Edges. The trimming of the edges and ends of plywood curves, produced by any of the processes described, presents some rather interesting and complicated sawing problems, particularly as the edges usually need to be sawed on radial lines, after making allowance for inserting in frame grooves. Both ends and edges must be trimmed with the minimum of tearing in the face veneer. Where edges are parallel, as in Fig. 9, the task is easy on standard twin saws, equipped with a saddle on the sliding table between the saws. Radial edges are much more difficult, and usually require saws located on the convex, rather than on the concave, side of the curved piece. Much ingenuity has been expressed in the type of cradles or holders to be placed on the double cut-off or equalizing saws, as well as on the double rip saws.

A large production would justify the building of swivel twin trim saws with sliding table between, since no machinery manufacturer seems to have provided adequate equipment for this operation.

Sanding Equipment. As in the case of sawing, this operation has required a variety of machine attachments to facilitate safe and speedy work. An oscillating saddle is shown in Fig. 18, which can be mounted on the sliding table of a belt-sanding machine and will adapt itself conveniently to the sanding of the outside of convex surfaces without unduly stretching or straining the sand belt. Pressure block *C* may have a concave surface, although such surfaces are dangerous and not desirable unless absolutely necessary. The ordinary flat under-surface is best for this block. Sander belts may be reduced to about 4 in. widths for curved surfaces and in some difficult instances may be made of cloth rather than paper. It is desirable to use the underslung rather than the overslung type of belt sander, since the latter interferes with the manipulation of the saddle and the curved plywood.

In some instances, however, the interior, or the concave side, of the curved surface must be sanded, and a cradle for doing this is shown in Fig. 19. This is practically the reverse of the saddle shown in Fig. 18, but needs careful operation to avoid a conflict between the sand belt and the oscillating shaft of the cradle.

Where doors are to be sanded on both concave and convex surfaces, it is possible to make a combination saddle and cradle,

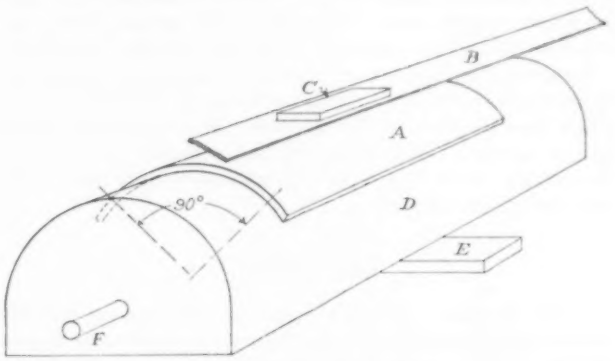


Fig. 18 Saddle for Sanding

(*A*, curved plywood to be sanded; *B*, sand belt; *C*, sander block; *D*, oscillating saddle; *E*, operator's handle for swinging saddle while sanding; *F*, saddle shaft.)

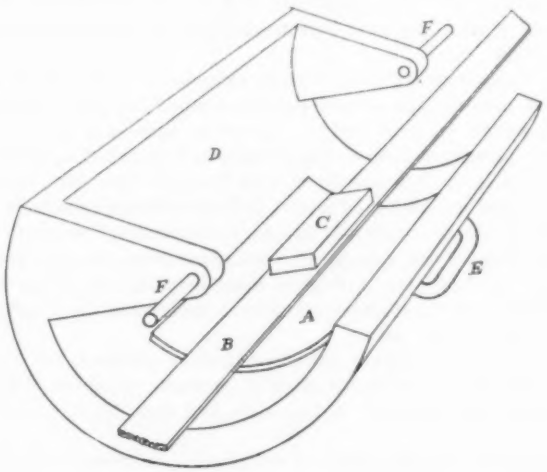


Fig. 19 Cradle for Sanding

(*A*, curved plywood to be sanded; *B*, sander belt; *C*, sander block; *D*, rocking cradle; *E*, operator's handle to rock cradle while sanding; *F*, cradle shaft.)

with suitable curved beds upon which each side rests during sanding.

The sanding of serpentine fronts can seldom be aided much by attachments and is best done by laying the curved plywood directly on the belt-sander table.

There are a number of special sanding machines made for curved chair stock, but in most instances these contemplate that the grain runs across the curve, and hence not in the proper direction for sanding curved veneer and plywood surfaces.

Conclusion

The various processes of producing curved veneered shapes have been outlined at approximately the present state of the art. The author realizes keenly that much research and experimentation remain to be done and that the present processes, now limited in their scope, may be greatly extended, and that prevailing cumbersome steps may be eliminated entirely by future developments.

It must be clear that there are great possibilities of extending

the scope and usefulness of curved plywood, not only in the making of artistic products for use in pianos, radio cabinets, and the like, but also in the design of a wide range of utilitarian products where curved plywood can be advantageously adopted.

Plywood can be adapted to many new and original purposes with its desirable qualities of bulk, reasonable resiliency, and high strength-to-weight ratio. It was repeatedly proved during the war that pound for pound plywood was stronger than steel.

Research along the line of making wood products water- and fire-resistant are rapidly overcoming these two limitations under which wood has labored for many years. Wood in its many forms has an unusual industrial appeal and certainly will hold an increasingly larger place in the utilitarian branches of the manufacturing industries.

Discussion

C. W. Elliott.[2] The description in this paper of the plywood-bending operation has not fully covered the range of possibilities that the writer's experience has revealed. While the development of the dry-bending machine has been largely along the lines of solid wood, the writer's experience has proved that practically all of the operations and products that can be made from solid wood can be made with even greater effectiveness from various types of plywood that are designed for bending-machine operations.

The wood-bending craft, with an extensive natural resource as its raw material, has not stepped up as rapidly as other branches of industry in past years. However, enhanced largely by canny inventions, time-saving devices, and perfected equipment, today it is making greater strides forward than ever before. As in other industries, advanced and more efficient methods are being introduced currently in the wood-bending business.

Possessed of many years' practical experience and trail blazing in this revolutionary method of wood bending, the writer feels qualified to give authentic information concerning the industry's progress in recent years.

Prior to inventing the dry wood-bending machine, the writer was engaged in the manufacture of products requiring the use of hoops and circles of various sizes and from $^1/_4$ to $^3/_8$ in. in thickness. The steam-bending process was used, and many difficulties were experienced, such as slow production, uneven curves, considerable breakage, and high cost of equipment.

While using the steam-bending process the writer conceived the idea of dry bending by machine, a radical departure from the conventional methods then in practice. He designed and built the first machine of this kind. When completed and placed in operation, the results far exceeded expectations, and the future possibilities were seen. It was apparent that it produced a better quality of bent work with truer curves. The loss by breakage was practically eliminated, and the cost was reduced to approximately one-half.

Later the work of developing a machine for the bending of heavier material was undertaken. Innumerable difficulties were encountered in solving the problems accompanying the bending of this heavier material. Means were needed for greater contraction of the inner curve and better control of the outer curve. Through a long period of experimenting, many devices were tried until the desired result was accomplished in a fairly satisfactory manner.

With machines in operation for bending the heavier stock along with the lighter material, the writer's company entered the field of custom bending, manufacturing bent-wood stock for many purposes. The material ranged in thickness from $^1/_8$ to $^7/_8$ in., and the radius from $3^1/_2$ in. to 7 ft.

In the custom-bending field both solid material and plywood were bent. However, the machines had insufficient scope to meet all requirements of bent stock needed in the woodworking industry. It was found that further development of improved machines for producing various shapes and compound bends would prove of great benefit to the industry, and were greatly needed.

Work on machines to produce this type of work is progressing. Many experiments have been made in recent months, and the results are proving quite successful. It is hoped to have these machines perfected in the near future.

A class of work is being produced by machine bending which heretofore was prohibitive, even by the method of gluing in male and female forms. This work includes complete circles with butt joint, semicircles or other shapes of uneven thickness, and complete circles with scarf joint. (See Fig. 20.)

Plywood and solid wood of $^3/_4$ in. thickness can be successfully bent to a radius of 18 in. on the new type of chair-back machine.

[2] General Manager, Elliott Wood Bending Machine Co., Columbus, Ind.

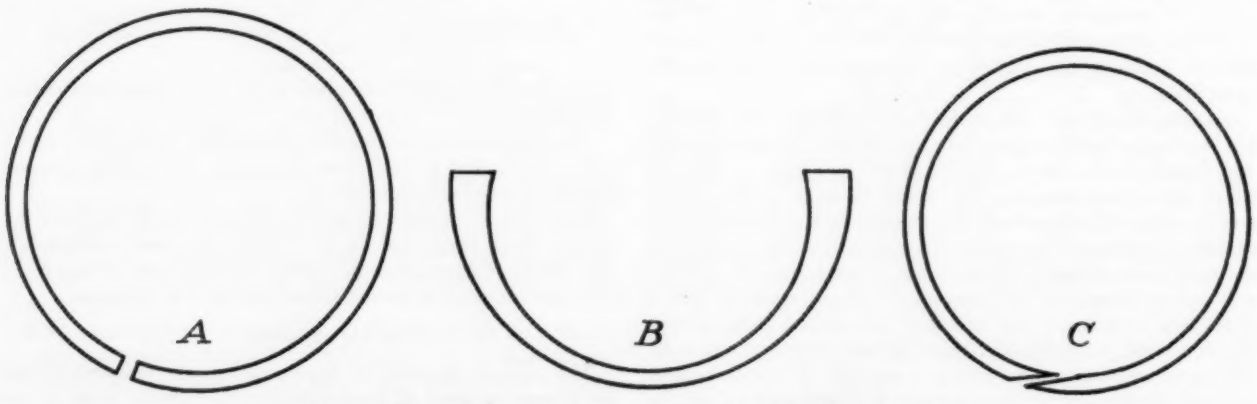


Fig. 20 Machine-Bending Work, Plywood Glued in Flat Sheets
(*A*, complete circle with butt joint; *B*, semicircle of uneven thickness; *C*, complete circle with scarf joint.)

WDI-53-4

Standards and Tests for Coated Abrasives

By E. C. SCHACHT,[1] TROY, N. Y.

The paper presents some of the standardization measures which have been undertaken by the sandpaper industry, tracing their evolution and influence on the product. Typical factory-control methods used in the manufacture of coated abrasives are described, together with testing media for evaluating the cutting efficiency of the finished product.



THE World War supplied the real centralizing, coordinating force in present-day industrial standardization. It stands today as an accepted economic institution, and yet generally there is little true realization or appreciation of its broad significance. The user of a commodity manufactured by an industry practicing cooperative standardization comes in contact usually with just one phase of its operation; namely, the elimination of unnecessary sizes, grades, items, etc. This contact in many instances produces an unfavorable impression because of the element of personal sacrifice, and it accounts for the occasional minority conception that standardization is a boon solely to the manufacturers of the article in question. By showing the purchaser or user the benefits which one particular industrial standardization confers on its trade, his cooperation can usually be obtained and real progress be assured.

In the sandpaper industry, the individual manufacturers have long appreciated the results of cooperative effort in eliminating unsound manufacturing practices and wasteful methods. That the user of sandpaper has profited by the development of the industry is certain, for there is a good way to judge the average improvement of the product and its increased production during the last 13 years. The quantity or reamage of sandpaper sold in the United States last year was nearly the same as in 1916. Therefore, over that period there has been little growth in the volume of sandpaper made. Over the same 13 years, however, the general increase in demand for basic materials has been about 70 per cent. The general economic growth of the country should have required about 70 per cent more sandpaper if the quality had remained the same. In addition to this, there have been new industries and uses that would have required, say, still 30 per cent more sandpaper. The total increase in sanding work to be done in 1929 was probably close to 100 per cent over 1916. It is certain that this increase in demand has been supplied by the improvement during 13 years of nearly 100 per cent in the quality of sandpaper. The increase in production per unit of area has been so great that no buyer of sandpaper used in production would today accept the product of 1916.

One specific instance of quality improvement will serve as an illustration. The fir veneer and door industry on the Pacific Coast uses six- and eight-drum sanders to finish and bring fir veneer to gage. The first or roughing cut is usually made with No. 3½ garnet combination, followed by finer grades of paper and combination coated with either garnet or aluminum oxide. Ten or twelve years ago as low as 4 hours' production was obtained from the first or coarsest cover, whereas today some plants have reported as high as 96 hours' continuous production from the present-day product which has been specially developed to meet the requirements of this very severe duty.

The decrease in the yearly consumption of sandpaper is not due to the fact that sanding is being replaced by other methods. It is not possible to prove this contention statistically, but close observers in the business can see no curtailment of old uses, but instead a development of new ones, notably floor-sander applications, waterproof sandpaper, and fiber disks for portable sanders in the automobile industry.

While the sandpaper industry has been cooperatively working out standardization measures, the individual manufacturers have been developing factory standards, control methods, and means of testing and evaluating the performance of the finished product.

Standardization in the Industry

Just a generation back each of the sandpaper manufacturers was making his product, which consisted of several hundred total items without any well-defined standards for his own plant and a complete lack of standards within the industry. The specifications for the product were continually being shifted and largely for competitive purposes.

The gradual development of the grading standard is interesting. Heretofore each manufacturer varied his grades or grain size for a given number as a competitive attack or defense. A single manufacturer would have in stock several different grades of the same number. Desiring to obtain the business on an item, his first move would be to select from his several variations of that number one slightly coarser than that of his competitor in order to increase the production. Or if the trouble was scratching, where possibly his competitor had been using a coarse variation, then he would recommend some of his finer gradations of the same grit number. The possibilities of error and trouble for his customer were obvious. The customer could not buy a competing brand without submitting a sample to match the grade. Under these conditions, of course, none of the grades was any too good. (See Figs. 1 and 2.)

The first attempt of the manufacturers to establish a uniform grade consisted in selling sheets of material by appearance and feel. That gave some pretense of reasonable grade spacing. When he chose to do so, the manufacturer would attempt to match this sheet. Later, silk testing sieves were designated by mesh number, and a rough specification for grade was thus set up. The difficulty with this method was the variation in sieves of the same mesh number. The next step was to establish standard sands for the most-used commercial brands. At this time there were still different grading standards for the various abrasive minerals used. Silk testing sieves were carefully selected, and a sufficient quantity was given to each manufacturer. A standard method

---

[1] Factory Manager for Behr-Manning Corporation. Mr. Schacht was graduated from the University of Michigan with the degree of B.S. in C.E. Subsequently, he pursued special studies at Carnegie Institute of Technology and at Stevens Institute. He served as Lieutenant (J. G.) in the Navy during the war. Since 1920 Mr. Schacht has been in the sandpaper business. He is a co-author of "Direct-Current Motor and Generator Troubles."

Presented at the Fifth National Wood Industries Meeting, New York, N. Y., Oct. 16 and 17, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

of screen analysis was set up so that different operators could check each other. Specifications were established for the amount of under grade, control grade, and over grade for each grit size. These terms are explained as follows: In a series of abrasive grades an individual grade is determined by two screens, one known as the fine screen and the other as the control. If the grain is shaken on the control screen, the portion that remains on the control screen is known as the "over grade," the portion passing through the control and remaining on the fine screen is known as the "control grade," and the remainder (i.e., the portion that passes through the fine screens) is known as the "under grade." The over grade is not greatly different in size from the control grade since it must be retained on the control screen of the next coarser grade. Likewise the undergrade must not pass through the fine screen of the next finer grade.

An attempt is made to keep the control grade as large as possible. The control screen of any grade acts as the fine screen of the next coarser grade, and the fine screen acts as the control of the next finer grade. (See Fig. 3.)

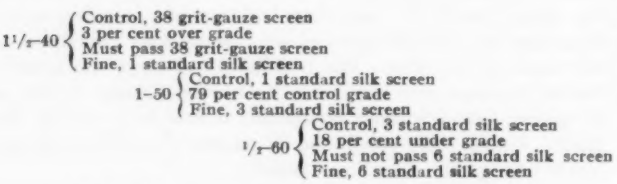


FIG. 3 DIAGRAMMATIC SCHEME OF CONTROL SCREENS

More recent refinements have come from the study of the size of apertures in silk sieves and specifications for sieve mode point. The method of determining the screen mode point consists of measuring the average size of 1000 openings by projecting the silk to be measured by means of a balopticon and measuring the projected screen image, averaging both dimensions of openings. Knowing the magnification factor of the lantern, these dimensions are converted back to ten-thousandths of an inch. A curve of size of opening versus per cent occurrence is then plotted. The mode point is the opening which occurs the greatest number of times. A mode curve of No. 11x–116 mesh silk is shown in Fig. 4.

Another method of particle-size measurement which gives reliable checks, but which requires a great deal of skill by the operator, consists in carefully calipering the vertical height of grain between surface plates.

By measuring the rate of fall of mineral in different fluids and assuming that Stoke's Law holds, an average particle size has also been determined. For fine grits methanol has been employed in order to decrease the time of the test, while for coarse grits a settling fluid of high viscosity such as glycerol has been used in order to decrease the time of fall to a measurable rate. For intermediate grits liquids having viscosities intermediate between methanol and glycerol are necessary.

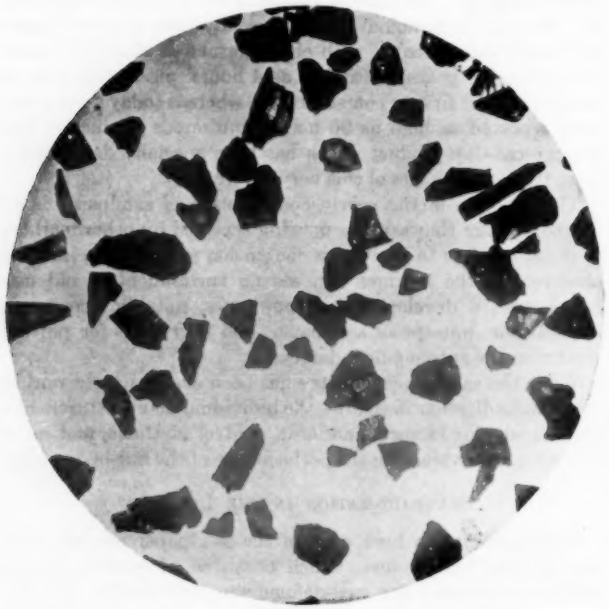


FIG. 1 A POORLY GRADED GRAIN

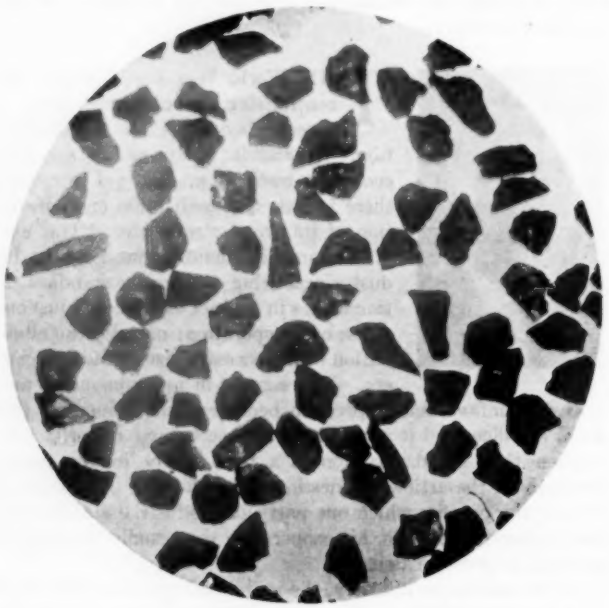


FIG. 2 A WELL-GRADED GRAIN

A uniform grade spacing for the important wood and metal-working abrasives has been worked out, so that now similar grade designations are employed on garnet, aluminum oxide, and silicon carbide. The grades as finally developed in the industry show a very fair grade spacing from one size to the next. For example, the following vertical height measurements were secured by calipering the maximum heights of aluminum oxide grain in the control portion of each grade. The measurements were made by calipering the grain between level steel plates and are expressed in ten-thousandths of an inch.

Mesh number	Grain height
400	12
320	20
280	25
240	33
220	40
180	45
150	50
120	64
100	80
80	105
60	140
50	175
40	215
36	260
30	310

Mesh number	Grain height
24	360
20	460
16	600
12	800

The standard sand and the analytical method have practically eliminated important grade variations as a problem to the user.

Similarly, paper and cloth backings have been studied, and the best practices have been determined. Fifteen years ago 14 different weights of paper backing in three different qualities, or a total possible combination of 42 paper items, were used. Today there are 8 standard weights of paper and a single variety for each class of product. The reduction in the number of items has increased the consumption of each class, so that the manufacturers of paper for the sandpaper trade can profitably devote research effort and provide special equipment to improve the product. Standardization programs have been conducted with the manufacturers of both rope and kraft papers, and limiting specifications have been adopted governing such factors as selection of samples, weight determination and tolerances, thickness, tensile tests, length and cross, air resistance, adhesion, color, moisture content, finish, and formation. By enlisting the cooperation of the paper manufacturers and acquainting them with sandpaper problems and the performance of their product in the field, remarkable results have been obtained. Rope paper, such as is commonly used for belts and drum covers, has been increased in length tensile strength from an average of about 85 lb. per in. to 150 lb. per in., and the adhesion of cylinder rope paper, which is the property which in a large measure controls the peeling of sandpaper, has been improved approximately 50 per cent.

At the present time a program is being worked out with the cloth manufacturers, and their aid is being enlisted in such measures as the adoption of weight, count and construction specifications, the reduction of stretch, the elimination of defects which must be cut out, and the increase in the length of cuts of cloth. The two latter measures will reduce the number of splices or breaks in the finished product. The work with the cloth manufacturers will necessitate their education in sandpaper and its uses. Customers of the sandpaper industry will eventually reap the benefits of improved cloth design and construction revealed to them in improved performance and condition of the finished product.

Some of the other industry standardization measures beyond those on abrasive grades and paper and cloth backings are as follows:

The elimination of dye in coatings to produce a uniform appearance, which makes it easier to substitute one brand for another without prejudice. Ordinary flint paper, which is normally light yellow in color, has in the past been dyed red to imitate garnet and has also appeared in shades of green and purple.

Coating classification and weight tolerances to guarantee a reasonable uniformity of coating from lot to lot and from one manufacturer's product to another. Tolerances within reasonable limits govern the standard weights of mineral used in making the two accepted coating variations, namely, closed and open coat.

Standardization of special shapes such as heel breasters, Websters, cones, etc. (In some of these cases it was necessary to cooperate with the machine manufacturers to get them to make equipment which conformed to the standard sizes.)

Improved ream and roll packings designed as a result of group experience and thought to more adequately protect the sheets and rolls of sandpaper in shipment to the customer.

Uniform quantities per bundle in ream goods.

Standard belt-joint specifications and many others.

Advantages of Standardization

The value of the standardization that has been carried out in the coated-abrasive industry has been very great to the user. Instead of being concerned with the determination and the analysis of a large number of factors of competing varieties of sandpaper, the user now has to concern himself with practically only one question which he must always determine for himself, and that is the cutting capacity or work-producing ability of the materials which he can obtain. The standardization program in the industry has almost eliminated the question of finish, although this must always exist to some extent due to individual skill and methods in cutting. The primary problem of the sandpaper purchaser is that of determining the cost of abrasive material per unit of product which he is turning out of his plant, which may be per square foot of veneer, or per desk, or per automobile, and the purchaser can put nearly all his effort into finding the right answer to this question.

The purchaser has in effect a set of rigid specifications, all prepared for him, that have taken more time and study to prepare than a single purchaser could devote to the subject. He need only specify in the briefest terms the material that he requires, and he is assured of uniform materials, weights, and gradings and that he will receive from practically every manufacturer a commercially comparable article so far as it relates to these several points.

There are, of course, among all of the manufacturers of sandpaper, differences in the use of materials, skill in manufacturing

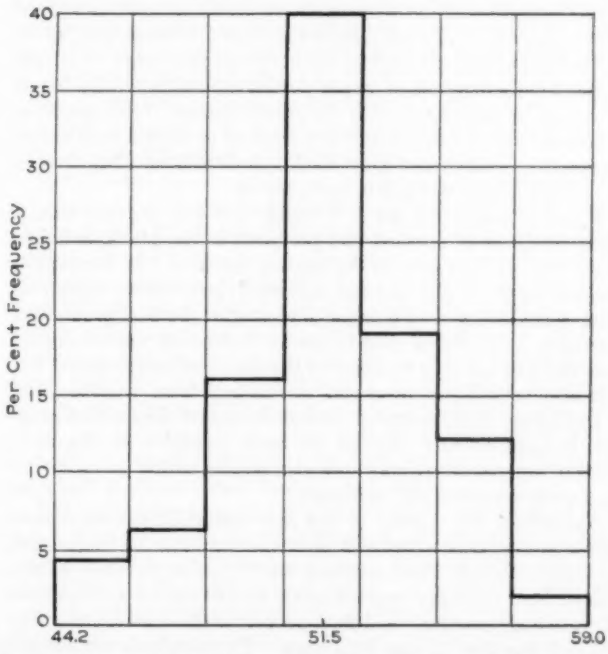


Fig. 4 Sample Mode Curve Illustrating Sharp Mode Point

methods, accurate control of the materials themselves, control of manufacturing operations, and the knowledge that comes from research and more indirect details of operations; differences in perfection of equipment—all of which make for very considerable variations between the working qualities of the product from one manufacturer and another, leaving a strong individuality of product in the industry that of course never disappears under any standardization program. The sandpaper manufacturer today finds himself in possession of excellent specifications for the principal materials, weights, and gradings. His

next problem therefore is the successful application of these standards to the most profitable control of operations in his own plant. This problem is worked out in different ways by the different manufacturers. The following will indicate the methods that are used by one manufacturer in putting his standards into actual operation.

Quality Control in the Sandpaper Plant

Paper backings are purchased which exceed the minimum limits adopted as industry standards. The competition in the paper industry affords the sandpaper manufacturer an opportunity to continually select paper whose tests far exceed the minimums established and to segregate for particular purposes paper having variations of such qualities as finish, softness, etc., which will suit it particularly well for use in certain products. A sample is cut from every roll of paper purchased and carefully tested to assure its agreement with the specifications. Grey goods as it arrives from the cotton mill is subjected to careful tests before acceptance. When the grey cloth has been finished or treated to prepare it for coating it is again tested.

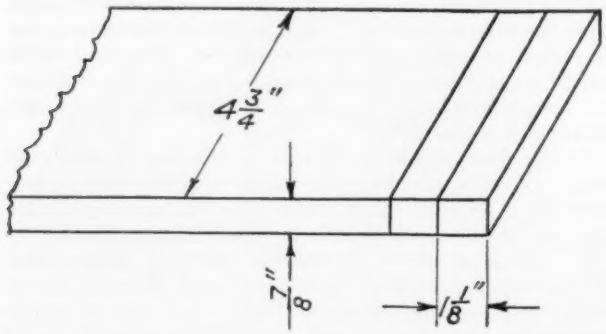


Fig. 5 Method of Cutting Test Blocks From Oak Boards

Abrasive grain, whether it is bought finished to grade from a grain producer or crushed and prepared in the plant, is finish-graded before it is sent to the coating machine. It is carefully checked for its ability to bond well with the adhesive employed. Each container of sand is check-graded after the final production screening by a control man to assure that every yard of coated material will not only conform to the standardization limits, but will also be uniform from run to run and from year to year.

Hide glue, which is one of the constituents of the finished product, is sampled and checked for such qualities as viscosity, acidity, deterioration, jelly strength, and its ability to be coated on a sheet smoothly and uniformly.

Assured of the quality of the raw materials, the sandpaper manufacturer must select and maintain standards for the finished product. An individual standard must be selected for each item coated (and there are approximately 478 items) by a committee of technical, practical, and field people in the organization who are well qualified to pass judgment. The standards are subject to a constant and continual revision based on performance reports of factory and field tests which insure a constant elevation of the quality. The standards are placed in charge of the Quality Control Department, manned by technical men. Each standard is in an individual envelope which carries complete and detailed specifications governing the manufacture of that item and is issued to the coating department immediately preceding the making.

After the proper standards have been set and a means provided for their maintenance, it is necessary to insure that the product is fabricated within the limits which have been imposed. A member of the Quality Control Department is present at each making and takes samples at regular intervals. The weight is continuously indicated by a continuous weighing scale, temperatures of all baths, etc., are automatically held constant, and the proper humidity and temperature regulation is automatically maintained in the drying compartments. Samples from the run are placed in individual envelopes containing all pertinent data and are kept on file by the Quality Control Department. Before filing, all samples are carefully examined, analyzed by the laboratory, and checked against the standard.

Periodically samples of all runs are spread out and reviewed for uniformity. All comments received from customers are considered and noted on the run envelope. Each month field performance reports, complaints, and general product experience is reviewed by the Quality Control and the Technical Departments with a view to the constant development of the product.

The foregoing outline gives merely the salient features of the control methods employed in sandpaper manufacture and in no way indicates the experimental, research, and developmental work which is constantly being expended on raw materials equipment, and manufacturing methods.

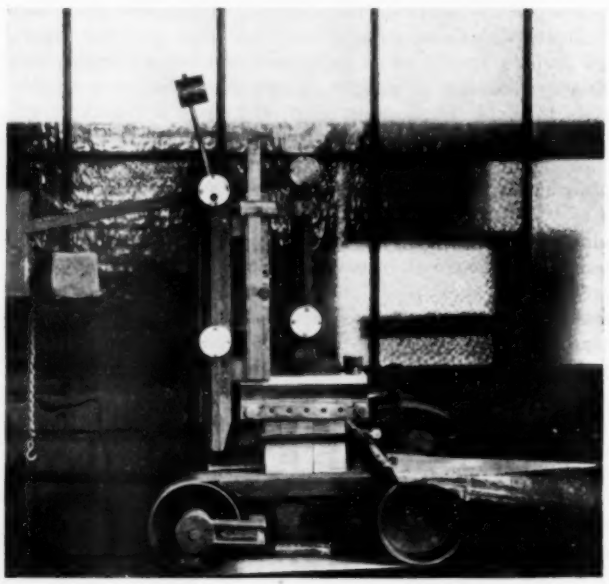


Fig. 6 Photograph of Belt-Testing Machine

Factory Testing of Sandpaper

From the Quality Control Department, the finished goods pass to the Testing and Inspection Department, which determines the performance rating and releases the goods for finishing and distribution if it is up to standard. The user is naturally most interested in the testing of the finished product, for he is concerned with the ability of the sandpaper to best do his job. He must, however, give due regard to the refinement of manufacturing operations, for it is this refinement which guarantees, not spasmodic making of a satisfactory product, but one which will constantly give uniformly good results. In the last analysis, sandpaper is only a tool, like a drill or a saw. Sandpaper is not generally used mechanically, but instead its use is attended with an expenditure of labor of far greater value. For this reason, either continued testing or performance checking should be resorted to, to evaluate the product, either by the sandpaper manufacturer or the user, or by both. Experience has shown that too often changes are made involving sandpaper as the result of a single test in which the variables in the conditions

surrounding the test were greater than the variations in the sandpaper itself.

In the factory tests a duplication of the conditions of field usage would at first appear to be the best method of testing. There are, however, in the case of woodworking materials several reasons why this cannot be done, the two outstanding, of course, being the time required and the material consumed as would be especially true in the case of drum sanders. It is necessary, therefore, to determine upon a test that will give a rating of the sandpaper which bears a relation to its rate of production in actual use.

Three varieties of testers are in common use on woodworking materials, namely, disk, belt, and reciprocal. The data secured on these machines have thus far proved very reliable, and while still the subject of refinement and development, the agreement

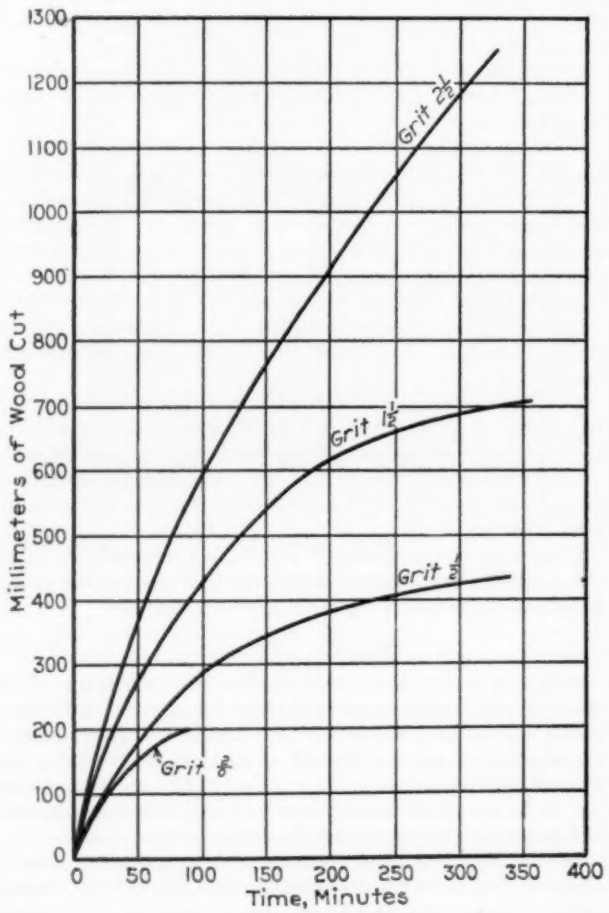


FIG. 7 BELT TEST CUTTING CURVES
(Amount of wood cut with different grits, using roll garnet paper.)

of results with those obtained in the field indicate that they represent good mediums for cutting determinations. Under controlled conditions in the testing laboratory, it is possible to get check results within a few per cent, and then almost without exception the results are substantiated in the field. Frequently, users of sandpaper are asked to conduct check tests of this kind, and their helpful assistance renders a great service, not only to the manufacturers of coated abrasives, but to all users as well.

BELT TESTERS

Two different types of belt tests have been used successfully which are described in detail for the benefit of any one who may wish to establish a belt-testing program.

(1) *Comparison With a Standard.* A Peerless belt-sanding machine, taking a sandpaper belt 4 in. wide by 86 in. long, is

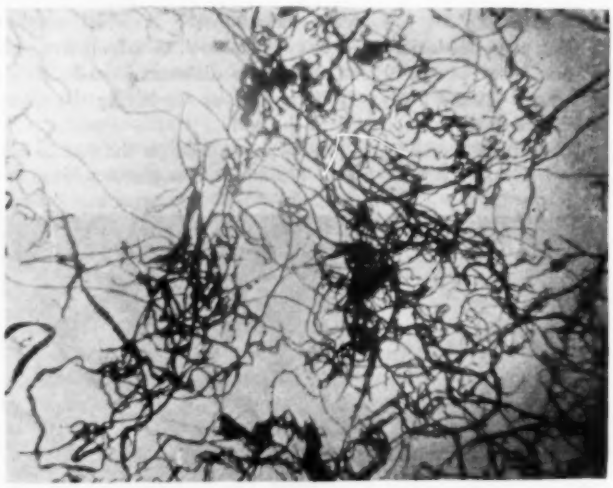


FIG. 8 WOOD ABRADED BY A FINE GRADE OF SANDPAPER
(The material removed is in the form of fine shavings, not sawdust.)

equipped with an arrangement for holding the wood test blocks to be abraded. The blocks may be counterbalanced so that any unit pressure within reasonable limits may be applied. For several years every run of woodworking material in the most generally used grits was tested by this method.

For a standard, a representative serial is selected and a good quantity put aside. It is not selected with the idea that it is 100 per cent, but rather as a reference mark.

Test blocks $1^1/_8$ in. wide, $^7/_8$ in. thick, and approximately $4^3/_4$ in. long are cut from well-seasoned oak boards. The length of the block ($4^3/_4$ in.) is determined by the travel of the block holders by which the blocks are applied to the belts and the economical width in which oak boards may be bought. The blocks from each board are placed in a separate container. Five blocks constitute a set, which is placed in a holder so that the ends of the blocks are applied to the belt and the sanding is done with the grain. These blocks are obtained by sawing $1^1/_8$ in. wide pieces off the end of a board $4^3/_4$ in. wide and $^7/_8$ in. thick, as shown in Fig. 5. One belt made from the standard material is tested on blocks from each container; that is, the standard is always tested once on each board.

Three tests (each one on a different path) are made on each belt. First, the belt is run 9 minutes. At the end of 9 minutes a new set of blocks is inserted. These are run 1 minute, and a reading in millimeters is taken. A millimeter scale and pointer are so arranged that the amount cut from the blocks being tested can be determined by taking a reading of the location of this pointer at the start and end of the test period. Five minutes later another reading is taken. The difference between these two readings represents the amount cut in millimeters in the interval between the 10- and 15-minute readings.

The blocks having been run 1 minute before the first reading is taken assures that the start of the test is always made under the same conditions. The result of the test is expressed in per cent of the amount cut by the standard, using blocks cut from the same board. This method helps to decrease any variation that might be caused by a variation in hardness of different boards.

A 5-minute test is used as a means of conserving the standard material. This test period, from the 10th to the 15th minute,

was chosen after extensive experiments had proved that this would give a reliable indication of what the material being tested would do during the average period of its commercial cutting life. It was found that the cutting curves rarely, if ever, crossed each other after this period within the economical life of the belt. A further method of conserving the standard material consists in using a set of blocks composed of one block each from five containers; that is, one block from five different boards. This also tends toward a more uniform test by modifying the effect of a board harder or softer than the rest.

A unit pressure of 16 oz. per sq. in. is used on the test blocks and a belt speed of 3000 lin. ft. per min. The test machines are



FIG. 9 WOOD ABRADED BY A COARSE GRADE OF SANDPAPER
(Showing that the cutting action is the same regardless of grade.)

arranged so that the unit pressure on the test blocks and the belt speed may be varied, but it has been determined, through a wide variation of belt speeds and pressures, that the greater the speed or pressure, or both, the faster the cut, there apparently being no critical speed or pressure that would give maximum cut.

Belt joints must be carefully made as a thick joint will cause the test blocks to jump and cause chattering that will affect the results. The successful operation of a belt testing machine depends to a large extent on the method of applying the test blocks. The use of a piece of sponge rubber back of the blocks will prevent chattering and vibration of the blocks and a consequent variation in the test results.

(2) *Direct Comparison Without a Standard.* This method of testing is more applicable to the comparison of several different runs or kinds of coated abrasives than to routine testing, although with some modifications it might be readily used for routine testing. One advantage of this method is the elimination of the standard and the possibility that the standard may change due to improper storage or other conditions.

For these tests a Peerless belt-sanding machine, taking a 4-in. by 48½-in. belt, is used. Fig. 6 shows this machine. The machine is enclosed in a small room in which the temperature and humidity are controlled. It is necessary to recirculate the air drawn out by the dust-removing fan, which is easily done by putting the dust-laden air through a bag filter, after which the air is returned to the test room. By proper manipulation of the heating unit and moisture spray, the temperature and humidity may be controlled within the required limits.

The belts and test blocks are conditioned at a constant temperature and humidity for 24 hours before being tested. This allows them to come to moisture-content equilibrium. A relative humidity of 35 to 40 per cent, with a temperature of 70 to 75 deg. fahr., is satisfactory for both conditioning and testing the belts.

A set of five oak blocks of the same size as used in the testing method previously described are used in the same type of holder. For this method, however, five blocks taken from five different boards are used to give a greater quantity of comparable wood. Great care is used to select uniform blocks; all those that have knots, hard places, or that appear soft are discarded. The selection of the blocks depends to a great extent on experience, but is quite reliable and susceptible of control.

Three tests are made in different paths on the same belt. Where possible, however, three belts are used, one test being made on each belt. The test is run from 30 to 90 minutes, depending on the grit. Readings in millimeters to determine the amount cut are taken every 5 minutes, and a cutting curve is plotted, time being plotted as abscissa and millimeters of wood cut as ordinates.

Oak has been chosen as a testing medium after experiments with a number of woods, including pine, mahogany, maple, bass, and gumwood.

A typical cutting curve is given in Fig. 7, showing the effect of a grain size or grade in removing material. The direct-comparison method was used in these tests. Oak test blocks, as described, were sanded under a pressure of 2 lb. per sq. in. This cutting curve was a part of a very extensive study of the cutting effect of various speeds and pressures. As stated before, it was found through quite a wide range of speeds and pressures that there was no critical speed or pressure, but that in general the greater the belt speed or unit pressure, or both, the more material was removed.

Two interesting microphotographs are shown in connection with the cutting action of coated abrasives on wood. The material removed is actually small shavings instead of sawdust, as is generally supposed. One, Fig. 8, shows shavings removed by a fine grit, while the other, Fig. 9, shows those removed by a coarse grit.

Except for the thickness of the shaving removed, the other characteristics are identical, indicating that the cutting action is the same regardless of the grade number of grit size.

RECIPROCAL TESTERS

A type of reciprocal tester is employed in the industry which is sometimes referred to as a grit retention or wear tester, consisting essentially of a machine for rubbing two pieces of sandpaper together under controlled conditions and weighing the amount of grain removed. Two pieces of the same size are dyed out of the sample of sandpaper to be tested, and these are conditioned overnight at a relative humidity of 65 per cent.

In running the test, one piece of paper is fastened to a table, abrasive side up, while the other is fastened to the rubbing block. The rubbing block is moved back and forth at a uniform velocity for 300 complete strokes, 50 complete strokes to the minute. At the conclusion of the test the loose abrasive particles are brushed on a watch glass and are weighed on an analytical balance to the nearest 5 milligrams.

This test is of value on grade No. 3/0 and coarser. The probable error is 8.8 per cent on samples taken side by side. The weight of the rubbing block is 10 lb.; area of contact on block, 4½ in. by 2 in.; length of stroke, 1 in.; speed of stroke, 50 per minute; number of strokes, 300, with uniform velocity.

The values obtained are expressed in milligrams. A test of 350 or less is considered satisfactory for most products. Certain products, such as metal cloth, which is designed for a high wear resistance, should test less than 175. In a few cases higher values are actually desirable.

The test reflects the quality and condition of the grain, the quality and condition of the adhesive, and the conditions of fabrication; that is, the construction of the coating. The test

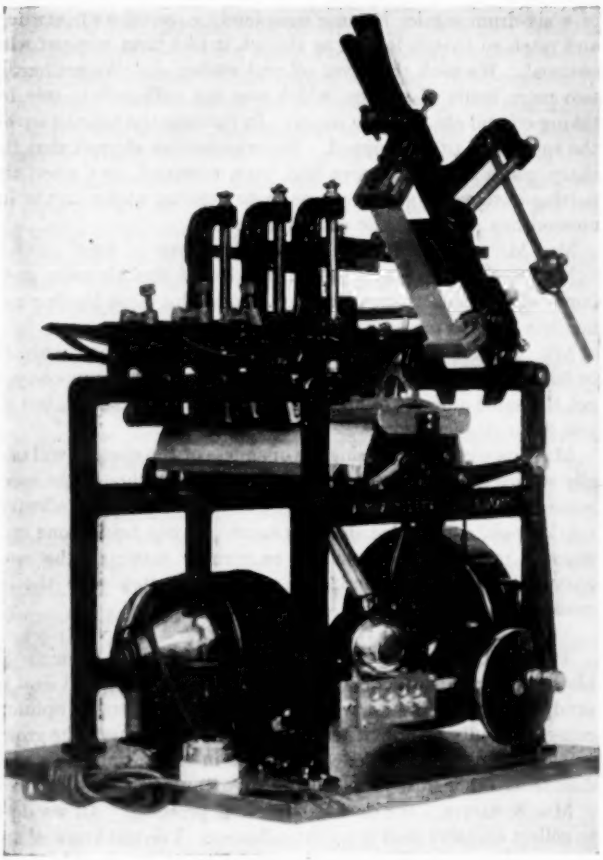


FIG. 10 WEAR TEST METER

is influenced by the amount of top-size adhesive employed, but within the normal top size range the test results are reliable.

The results of the grit-retention test bear a close relation to efficiency constants determined from belt-sander tests. Together they predict the amount of service that a sandpaper will perform.

Fig. 10 shows a wear-test meter which is manufactured commercially and used for testing textiles, leather, carpets, linoleums, etc., and which can be used to test coated abrasives. For testing sandpaper, a standard sheet is placed on the oscillating drum, and the samples to be tested can be stretched in the sample clamps. The amount of abrasive removed is again the measure of value. The illustration shows a four-arm machine, in which four samples can be tested simultaneously.

Reciprocal testing machines have also been employed in which a block or form covered with the sandpaper sample is caused to move backward and forward at a constant speed and pressure upon a piece of test material. The amount of stock abraded serves as the measure of cutting efficiency. Disk testing machines have been similarly employed in which the test piece to be sanded is fed to a revolving disk. Both of these methods have thus far proved more reliable in the testing of metal and composition and have not found much application on wood.

CONCLUSION

The development of standardization measures in the sandpaper industry has resulted in the general production of better coated abrasives which are uniform in specifications affecting interchangeability of competing products. Such specifications particularly refer to grit sizes and grade spacing, backing weights, coating tolerances, and color; and the purchaser, in addition to having reaped the benefits of quality improvement, can buy, without restriction as to individual manufacturer, on specifications as to these factors, which are probably more rigid than he would individually impose. He has only to concern himself with the selection of the product which will give most satisfaction in his particular application.

Industry standardization has been accompanied by the adoption of plant standards and refinement of operating control. The individual development among the different sandpaper manufacturers involving differences in the uses of materials, skill in manufacturing methods, accurate control of materials, research, and perfection of equipment has been such as to make for considerable variations between the working qualities of the product from one manufacturer and another, leaving a strong individuality of product in the industry. The refinement of quality control is designed to effect constant improvement and uniformity.

Factory testing mediums consisting essentially of belt, disk, and reciprocal testers provide a means of evaluating the cutting efficiency of the finished product.

Discussion

G. H. GOVE.[2] The great advances which have been made in the quality of coated abrasives during the past ten years can be attributed to a great extent to the greater use of technical control in this industry. As the author has indicated, the reduction in number of grades and backings has made it possible to concentrate the attention of the raw-material suppliers on the problems of higher grade materials of more uniform nature. The standardization of abrasive grades has clarified a most troublesome industrial problem, resulting in better service to consumers of coated abrasives.

This industry is one in which rapid advances in quality occur, caused primarily by better raw materials, better plant supervision, and better methods of testing the finished product.

We have found at this plant that the reduction in the number of items to be manufactured has increased quality and more particularly has increased our ability to give prompt delivery. Standardization of raw materials has increased the quality of the product. Technical control of plant process has assured uniform, high-grade products, with a constant improvement. The paper has covered these ideas in great detail and clearly shows how these standardizations and tests have been developed.

G. H. KOHL.[3] Conditions in a plant are sometimes such that a fair test is not always possible. In regard to tests, we are up against the fact continuously where an operator will feel better in the morning than he will in the afternoon, and therefore exert more pressure, or vice versa; and the result is that one never gets the same result from the sandpaper. That is one of the outstanding reasons why tests are not always fair. The conditions of the machine in the plant which is not operating to its best capacity will often affect the sandpaper. But we are anxious to hear from the men in the plants as to what way they think sandpaper should be tested in the plants.

J. A. MCKEAGE.[4] In an ordinary drum sander or belt sander,

[2] Superintendent, United States Sand Paper Co., Williamsport, Pa.

[3] Power Corporation, Montreal, Canada.

[4] McKeage Woodworking Machine Company, New York, N. Y.

is it in the testing machines that these various papers have been tried out? Is the cutting of any particular wood directly proportioned to the speed or not?

MR. SCHACHT. This is a question which a machine manufacturer always asks because he is very much interested in the speed of the machine and the amount of pressure applied. We attempt to find the answer on our factory testing equipment and in the field, where unfortunately the limitations of the machine often govern the results. On testing machines within safe working limits, we have found that the amount of wood abraded is directly proportional to the speed, pressure, and grit size. In field tests it was found that no increased cut was obtained in speeds beyond the 5000-ft.-per-min. mark. In regard to finish, I am a little hesitant to say what would be the result of increased speed and pressure, because finish cannot be evaluated on test blocks. Increasing speed and pressure will produce increased cut at least up to about the 4500 to 5000-ft.-per-min. range.

MR. MCKEAGE. Can you give the maximum speed recommended to sandpaper?

MR. SCHACHT. Four thousand feet per minute on garnet and 4500 ft. per min. on aluminum oxide seem to be maximum speed limits on woodworking operations as far as increased efficiency is concerned. These are general figures substantiated by laboratory tests and tests conducted in the field by manufacturers who have their plants equipped with direct current and could make observations as to square footages sanded with varying speeds.

M. J. MACDONALD.[5] Does that speed mean on all grains?

MR. SCHACHT. That represents some tests on intermediate grit numbers. I should like to ask some of the sandpaper people if their views coincide with mine.

MR. KOHL. I think the higher speed you have, the better finish you get.

J. C. DUKE.[6] I have had some experience in that field in running speeds and stepping up speeds. I think that Mr. Schacht has shown that in the control in the laboratory tests of sandpapers, the sandpapers have been increased in quality to the point where we are able to step up speeds. Some of the woodworking machines have run 3500 f.p.m. There are several well-known belt sanders that are today running 4300 and 4500 f.p.m. Some years ago that speed would have caused burning in panels. Sandpapers have become sharper in their manufacture. One particularly interesting case was that machines that were capable of standing greater speeds were being driven at a speed of 2500. This is a very slow speed. This was prolonged to a point where, perhaps, more heat was caused by the friction passing across the wood. In this particular instance, we stepped up the speed of the machine to 4000 f.p.m. At a speed of 4000, after marking the panels, the finish was pronounced far superior. The length of life of the product was increased by about 30 per cent. There have been several tests of this kind in plants, and in all cases the finish was much better and the length of the life of the product was a great deal better.

MR. MCKEAGE. In sanding varnished surfaces, of course all know that the paper fills up more rapidly. Is that due to the varnish or to the foreign material actually jamming in the grain of the sandpaper or is it due to the heat of the cutting? If you cut closer when you are cutting varnished floors, will that do away with the gumming of the paper to any extent?

MR. SCHACHT. Filling up is probably due to the dulling of the abrasive particles, causing heating which promotes loading with both the varnish and sawdust. If we take a piece of paper whose surface has been gummed up either through sanding varnish or sanding a resinous wood like pine, we find that if we clean the sheet and put it back on the drum we get very little additional service. I recall seeing a field test on pine where six hours was the average length of service before the cutting drum of a six-drum sander became completely covered with sawdust and pitch so that it looked as though it had been covered with oatmeal. We took the cover off and washed it. We got hardly two more hours of service, which was not sufficient to pay for taking off and cleaning the paper. In running the cleaned cover, the speed had to be dropped. An examination showed that the sharp edges of the abrasive had been removed, and when the cutting edge was gone there was more heating, and with the increased heating came the excessive loading.

MR. MCKEAGE. Is there a marked increase of heat?

MR. SCHACHT. When the little edge of the abrasive goes, increased heating is experienced, attended by rapid loading and finally a marked increase in temperature.

MR. MCKEAGE. If you use open coated paper with 50 or 60 per cent the amount of abrasive on the belt, I suppose you get the same amount of cutting by increasing the speed, but do you get a cleaner paper?

MR. SCHACHT. Increasing the openness of the coating will usually reduce the cutting capacity of the belt, even though the speed is increased on account of the reduction in the number of effective cutting points, provided that the sanding is not being done on a resinous material. On resinous or gummy material, the open coat will stay cleaner and give more production with the increased speed.

THOMAS D. PERRY.[7] In the salvaging of the dust through the blowpipe in the various factories, we accumulate a good deal of sandpaper dust and sand, which, according to the general opinion, causes difficulty on our grates and in many cases causes the grates to clog. Has any progress been made in separating sandpaper dust from other dust in woodworking manufacture?

MR. SCHACHT. We do not have that problem. All we do is to collect abrasive dust in a dust collector. I do not know of any way of segregating in the woodworking trade the dust from the abrasive material removed, and I do not know any way of reclaiming it.

MR. PERRY. The theory was advanced that there were certain formations coming from sandpaper dust that caused trouble on our grates.

MR. SCHACHT. I would question whether you get that proportion of sand to wood. I do not imagine in a day's running of a plant that you would get enough abrasive material to form anything on a grate.

MR. MCKEAGE. Which is the best buy, paper or cloth?

MR. SCHACHT. If it can be used, paper is the best. It is considerably cheaper and the coating is just as good on paper as it is on cloth. Therefore it will cut equally as well and often in tests it exceeds the production from cloth. Cloth is usually used where the limiting factor is flexibility. The cause of ultimate failure in most belts is not the rupture or failure of the backing, but instead the wearing out of the cutting surface. Paper is sometimes likely to be brittle if overdried in storage and then may crack and tear across if it is pinched, whereas cloth will stretch and accommodate itself. On a 6-in. belt there is quite a resistance to any strain. Paper is the best if one can use it, but where the conditions of service prevent its use, then combination or cloth should be selected.

P. H. BILHUBER.[8] When a board is sanded, snake marks occur. What are the results and what can be done to eliminate them?

[5] Long Island City, N. Y.

[6] Minnesota Mining & Mfg. Co., St. Paul, Minn.

[7] New Albany Veneering Company, New Albany, Ind.

[8] Steinway Piano Company, Long Island City, N. Y.

MR. SCHACHT. This is a problem which is always coming up. We find in some cases that no matter what paper is used, the snake marks will persist. Here we draw the inference that the fault lies in the machine and may be a condition of the drums, the felt covers, etc. By carefully removing the pitch and sawdust from the loaded spots on a snake-marked cover, we can attempt to find irregularities in the coated surface. It is practically always impossible for us to find one grain or series of grains which cause the snake mark, and calipers will show no differences in elevation greater than those present on other parts of the cover. In running hundreds of thousands of yards of paper with only infrequent snake-mark trouble, it is impossible definitely to place the blame on the sandpaper. I do not know what the abrasive manufacturer can do to eliminate snake marks beyond producing evenly distributed and level coatings. Are not snake marks generally noticeable on drum covers which have run their normal course of life?

MR. BILHUBER. In my opinion, it is not so much a case of the doctor as it is the nurse. There are certain reasons for the snaking on drum sanders. It is present in every drum sander that uses the oscillating drums. It is just the degree in which these snake marks occur which render them objectionable or not.

MR. MCKEAGE. One reason for the snake marks is that you have dried gears and they chatter, so that it affects the side oscillation, and causes marks like duck tracks, and this is also due to your felt getting too hard. The main thing is to get slower oscillation or else there is so much play in the mechanism that the proper effects cannot be secured.

MR. BILHUBER. The only kind of a machine to use in a plant is a good machine, and any that is older than six or seven years should be sold or given away.

MILTON KUBELL.[9] On my treatment on sanders, after 1200 or 1300 ft., I start getting snake marks. How would you account for that?

H. G. SEAMAN.[10] We find that as soon as the paper begins to wear down, the snake marks show up. The cause for this we found was that the rolls were not stationary while in operation.

ARTHUR SMITH, JR.[11] What were the main considerations which led the sandpaper makers to drop the double-faced finishing paper.

MR. SCHACHT. First, the demand for double-faced finishing paper dropped to a very low point. Secondly, the differences between single -and double-faced finishing papers are really very small. Double-faced paper consists of two plies that are easily separated in the middle, each ply being 30–32-lb. paper. Single-faced paper is made on 40-lb. paper, so the difference in paper weight is small. As far as the user is concerned, in buying two sheets of single-faced paper at approximately the same price as one sheet of double-faced paper, he gets a big bargain, because it is obvious that in making double-faced paper, we coat one side, run it through the whole drying process and then return it to be coated on the other side, which produces one sharp and one dull side. You probably would secure 100 per cent service from one side and hardly 75 per cent from the other. Double-faced paper is still available but it is not made as a standard item.

MR. SMITH. Our finishing foreman tells us that there is an advantage is using the double-faced paper double; that it permits the mechanic to get a better grip on the paper in holding it in his hand. Do you think that is so?

MR. SCHACHT. I am afraid that he would be wasting paper if he did it.

[9] National Casket Company, Long Island City, N. Y.

[10] Behr-Manning Company, Troy, N. Y.

[11] G. Smith Woodworking Company, Philadelphia, Pa.

Improvements in Steels for Wood-Cutting Saws and Knives

HENRY B. ALLEN,[1] PHILADELPHIA, PA.

The important factors in a review of saw steels are the blade which carries the teeth and the teeth themselves. There is little in sight to promise further marked improvements in the solid-tooth type of saw. The next logical step is to use inserted teeth made from materials known to be well adapted to the function of cutting wood. The same forward step was taken years ago in metal cutting.



THE two important factors in a review of steels used for saws are the blade which carries the teeth and the teeth themselves. The steel for the blade, after heat treatment, should combine high elastic limit, good ductility, resistance to fatigue cracking, resistance to impact, and uniformity. There is little in sight to promise further marked improvements in the solid-tooth type of saw. The next logical step is to use inserted teeth made from materials known to be well adapted to the function of cutting wood. This same forward step was taken years ago in metal cutting. Future changes in design, so as to put to use metallurgical discoveries, will require the best thought of saw user, machine builder, and saw maker. Research and untiring effort alone lead the way to accomplishment.

The saw is a venerable tool. It dates back to prehistoric time. The first saw consisted of serrations along the thin edge of a piece of stone or bone. As bronze and, later, iron were discovered, the same scheme was followed, only with the greater refinements made possible by the malleability of the metals.

When steel came into civilization, still further improvements were possible. Here was a material which could not only be forged into shape, but subsequently could be made hard and springy.

It is interesting to note that at this stage of development Western practice very probably branched off from that of the East. The bronze or iron saw, because of its lack of springiness, could be used only to full advantage when pulled through the material being cut. The Oriental still operates his hand saws in this fashion. The West, however, finding that a tempered blade possessed enough spring to prevent buckling, decided to do his cutting on the push stroke.

Civilization has always been slow to change. This conservatism has applied with full force in the field of the mechanic arts. Great changes, however, are taking place in our day; the teachings of science are more widely accepted and applied than ever before. If a curve were plotted between time and the application of progressive ideas in the field of mechanics, we would see a long, slowly rising line, with here and there an up or down trend, until a comparatively few years ago, when the curve started up the present rapidly increasing slope.

The saw maker and user have, in the past, been no exceptions to the general law of human conservatism. The manufacture of saws has been an art acquired by years of first-hand experience. Specialized knowledge and equipment have been required. The manufacture has therefore been retained in the hands of comparatively few. Probably for these reasons there is a remarkable scarcity of published information about the steels used.

Simplicity is a consideration of prime importance in any device. The saw has been a good exponent of this principle. Even today the greater number are made from a single piece of steel, the blade and teeth integral and of the same temper. The advantage of this design is mainly simplicity of manufacture, hence low cost. The disadvantage is that the steel forming the blade must also serve for the teeth. The blade, as we shall see later, requires properties which do not coincide with those best suited to a cutting edge.

The obvious way to eliminate these shortcomings of the solid-tooth saw is to provide separate teeth which can be mechanically attached to the blade. Thus each part of the saw can be made out of a material best suited to its respective function. Inserted-tooth saws have been used to some extent for many years, although not primarily for the reason just stated. In the past they have found their place mainly because of the ease of keeping them in running order and also that they hold to their original diameter. More attention has been lately paid, however, to their great usefulness in providing the means of applying a more efficient cutting medium for the teeth. The discovery of new steels and materials which display valuable properties for cutting wood, but which cannot be used for the blade, must result in the further development of the inserted-tooth saw.

There are therefore two general types of saws, the solid tooth and the inserted tooth, to be included in a discussion of the steels used. Inasmuch as the problems of today are concerned mostly with power-operated tools, the paper will be confined to that type.

There are two equally important factors which must be considered in a review of steels used for saws; namely, the blade which carries the teeth, and the teeth themselves. Both are of equal importance to the success of the saw. The subject of saw steels will therefore be taken up under the two headings. Knives also can be treated in the same way, although the problems offered by the knife back or blade, as distinct from the cutting edge, do not offer the same difficulty.

The Blade

Most tools can be made with as heavy a shank or body as is necessary for rigidly supporting the cutting edge. The saw, however, is an exception. By its very nature it is a thin tool. The policy of conservation in the wood industries is increasing the tendency toward greater thinness. While the saw blade has strength and rigidity in a direction truly normal to its thickness, it is weak and flexible under side strain. The rigidity of a saw has a direct bearing on the accuracy and smoothness of the cut produced and on the rate of cutting, which are considerations

---

[1] Chief Metallurgist, Henry Disston & Sons.

Presented at the Fifth National Wood Industries Meeting, New York, N. Y., Oct. 16 and 17, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

of first importance. An increase in the blade thickness is the direct way to produce greater stiffness, but this method is prohibited by the equally important consideration of wood wasted. A material having a higher modulus of elasticity would give greater stiffness to bending, but there is no relief in sight. We have to be content with the 29,000,000 elastic modulus of steel.

The speed of a revolving circular saw and the static tension applied to a band saw furnish a stiffening effect which compensates in part for the flexibility of the disk or band.

A circular saw revolving at wood-cutting speeds involves the complicated mechanics of centrifugal stresses and also vibrations. The stresses set up by centrifugal force are low as compared with the strength of steel. A 16-in. saw revolving at 3600 r.p.m. has a maximum stress from centrifugal force of about 6000 lb. per sq. in., and this occurs at the center hole. But while this stress is not high, it does produce some stretch in the blade, which is not uniform from center to rim. These strains in turn, if uncompensated, result not only in added lack of stiffness, but even in an actual fluttering of the rim.

To counteract the stretching from centrifugal force, the circular saw blade is given a prestraining operation called tensioning. This may be done by hammering, rolling, or otherwise, and results in a saw which will be "stiff" when up to speed. The stretching of the saw rim in work due to frictional heat is compensated for in the same manner. While the stresses set up by centrifugal forces are low, those induced by the prestraining treatment are high.

The steel for the blade, after heat treatment, should combine: (1) High elastic limit, (2) good ductility, (3) resistance to fatigue cracking, (4) resistance to impact, and (5) uniformity. And because most saws in use today are of the solid-tooth variety, there is an additional qualification; namely, (6) satisfactory edge holding.

It is self-evident that the steel and its fabrication into saws must not be too costly. Another consideration that is becoming increasingly important is that the steel weld readily.

(1) *High Elastic Limit.* The saw blade should behave as nearly as possible as a perfect spring. Operating strains, due to centrifugal force, frictional heating, or other normal causes, should produce as little permanent distortion as possible. If a blade is strained beyond the elastic limit, it will be stretched or distorted, and the saw will have lost its tension or have become uneven and lumpy. In any event, it will not run true, and so will produce a poor cut and not stand up under heavy feed.

Plain carbon-tool steel, formerly used for all saws, could not be heat treated to the high elastic limits found desirable for the more modern production rates without an over-sacrifice of other essential properties. For instance, a high elastic limit would be accompanied by brittleness. The usual carbon content of the steels used was 0.85 per cent, although it might range as low as 0.70 per cent up to 1.10 per cent. The carbon content used depended more on the edge-holding requirements of the teeth, although high carbon was often selected for a thick plate so as to get better hardening.

Nickel was the first alloy used to any extent in saw steel. The object was to produce a stronger blade without sacrificing toughness. The nickel content ranged between 1.25 to 3.5 per cent, with carbon between 0.60 and 1.00 per cent, depending on the use of the saw. The plain nickel steel constituted a real advance in saw steels, and it is still used to some extent. One objection to this steel is its tendency toward a striated, crystalline structure which may result in spalling teeth. Another difficulty encountered in the use of this steel is sluggishness in tempering, some of the hardened steel structure remaining, with consequent brittleness. These objections are more often met with when the nickel and carbon contents are on the high side. However, when on the low side the amount of benefit derived is open to question. The compositions shown in Table 1, (*a*), (*b*), and (*e*), are types still used to some extent.

The properties of nickel steels were so interesting that the next developments of alloy saw steels were mostly made with the view of eliminating its weak points by the use of one or more additional alloys. Following other alloy-steel developments, chromium was tried and found to give very high elastic limits, but at some sacrifice in ductility. The steel showed a tendency to be "brash," as the old sawmaker calls it when the steel will not stand considerable cold swaging without checking. Vanadium was added with the object of overcoming this "brash" condition, but without more than a partial cure. It does, however, produce a marked increase in toughness. Nickel-chrome steels are used today for many saws where the ductility requirements are not severe. Typical compositions are shown in Table 1, (*c*), (*d*), (*f*), and (*j*). Chromium is an inexpensive alloy and in some compositions can be obtained from the scrap metal charged into the steel melting furnace. Chromium, too, in small amounts, enhances the hardening power of nickel steels and so aids the saw manufacturer in hardening the thicker saws.

Another attempt to obtain high strength, together with high values for the other essential properties, has resulted in nickel-molybdenum steels, both elements being present in small amounts. Table 1 (*g*) shows a typical composition used very largely and with good results. The elastic limit can be made high without sacrificing ductility and toughness. The types shown under (*j*) and (*k*) are very good steels where no cutting edge is involved.

(2) *Ductility.* The elastic properties of steel, together with its ductility, make possible the tensioning or prestraining treatment essential to a band or circular saw. Mechanical tensioning is done by hammering or cold-rolling selected areas of the saw. The cold working elastically expands the steel adjacent to the spots where the force is applied, the latter spots being permanently stretched. A steel low in ductility can be given but little deformation by the cold working without dangerously approaching the breaking point.

A phase of ductility, the ability to be cold-worked without rupture, is an essential property of solid-tooth saws whose teeth are swaged for clearance.

The property of ductility is dependent, to a large degree, upon the inherent quality of the steel as determined by the method of manufacture and the quality of raw materials used. No alloy will compensate for inferior steel-melting practice.

As noted previously, chromium does not assist in developing ductility in saw steel. Nickel does help, and the combination with molybdenum, Table 1, (*g*), gives particularly good ductility with high elastic limit. Alloy steels have eliminated most of the body cracks which were fairly common in carbon-steel saws.

(3) *Resistance to Fatigue Cracking.* The high speeds and feeds at which production saws are now run have introduced new problems. Among these is the matter of gullet cracking, due to repeated stress below the elastic limit of the steel. Cracks of this type can usually be identified by the lines of progressive growth on the adjacent surfaces of the fracture. Sharp angles or corners should never be allowed where there is repeated stress, but the saw user often allows this condition to exist. Gullet cracks sometimes occur which cannot be traced to any such self-evident cause. Those who have made extensive studies of the phenomenon of fatigue cracking are generally agreed that the strength of steel to resist this type of failure bears a direct relationship to the ultimate strength; also that the freedom of the steel from dirt and other inclusions has a direct bearing on its resistance. If such is the case with saws, the present-day alloy steels, when clean and well made, should possess remark-

able resistance. This is not always true, and there is reason to believe that fatigue cracks in band and circular saws may be caused by complicated stresses which are aggravated rather than relieved by high physical properties. Good ductility in the steel is a valuable asset, however.

Special measures can be used to largely prevent fatigue cracks from forming. Properly designed and spaced rim slots is one means often used. Another method frequently practiced, in band-saw mills, is to cold-work the bottoms of the gullets, thus stretching the metal at those points. Keeping the saw sharp and preserving well-rounded gullets and even tension are self-evident and worth-while precautions.

(4) *Resistance to Impact.* A saw in service is subjected to repeated blows. If the teeth are sharp and the saw is in good shape, the normal impacts of the teeth in the wood are not excessive. But other shocks incidental or accidental to service may be heavy. One form of accident often met with is striking a piece of steel in the cut. The old-type carbon-steel saw would be apt to crack under such treatment, or at least be put badly out of shape. A good alloy-steel saw will either cut through the steel or will have teeth sheared off, the blade itself being left in fair condition. Side shocks which would distort a low-elastic-limit carbon saw will have little effect on the alloy blade.

(5) *Uniformity.* A saw which has not uniform physical properties will never be a satisfactory tool. If the blade has hard or soft spots, it can only be made flat with difficulty, it cannot be tensioned evenly, and it will not retain its trueness or tension in service. Lack of uniformity can be due to variations in steel composition or to faulty heat treatment. The first-mentioned cause is most apt to occur from failure to remove any decarbonized skin that may have been on the steel sheet. Faulty heat treatment is the usual reason for ununiformity and may result from poor heat-treating practice or by reason that the steel composition is unsuited to good hardening.

On account of the physical dimensions, saws must be hardened in a comparatively slow quenching medium such as oil. Carbon steels will not harden thoroughly in oil unless the piece is thin or unless it has been overheated, which is of course bad practice. Carbon steels, therefore, cannot be considered as suitable for saws to meet present high standards unless they are of light gage.

The judicious use of small amounts of alloy will give the steel suitable hardening properties. For this reason many saws that are to be used only for work of the lighter sort are made from low-priced alloy steels. Chromium is an alloy often used in this way, such as the composition shown in Table 1, (*h*). Where a slightly tougher steel is wanted composition (*i*) is a good choice.

TABLE 1 TYPICAL CIRCULAR SAW-BLADE AND BAND-SAW STEEL COMPOSITIONS

	Carbon	Nickel	Chrome	Vanadium	Molybdenum
(*a*)	0.70 to 0.90	2.00	..	..	..
(*b*)	0.70 to 0.90	1.25	..	..	..
(*c*)	0.70 to 0.90	1.25	0.25	..	..
(*d*)	0.70 to 0.90	1.25	0.35	0.20	..
(*e*)	0.70 to 0.90	0.70	..	..	..
(*f*)	0.70 to 0.90	0.70	0.25	..	..
(*g*)	0.70 to 0.90	0.70	..	..	0.15
(*h*)	0.70 to 0.90	..	0.40	..	..
(*i*)	0.70 to 0.90	..	0.40	0.15	..
(*j*)	0.55 to 0.65	1.25	0.35	0.20	..
(*k*)	0.45 to 0.55	..	0.80	..	0.20

Little need be said about the properties of the steel back in machine knives. Sufficient toughness is of course necessary, together with stiffness. Some types of knives, such as the hog and chipper, are subject to heavy impacts and pressures, and therefore the steel must possess more toughness than is needed for other kinds. The planer or finishing knife, on the other hand, will have sufficient strength if made out of a less tough material such as high-speed steel. Some typical compositions of solid knives are given in Table 2.

TABLE 2 TYPICAL KNIFE STEELS

Carbon	Nickel	Chrome	Tungsten	Vanadium
0.90	..	..	...	..
1.10	..	..	...	..
0.90	1.25	0.35	...	0.20
1.00	..	0.80	...	0.20
1.10	..	..	2.50	..
0.65	..	4.00	18.00	1.00
1.25	..	4.00	9.00	..

The Cutting Edge

The cutting action of a saw and of most knives is similar to that of other cutting tools. The wood is parted by a wedging action, and the power required is therefore a function of the cohesive strength of the material. When the wood is gummy, an additional factor of increased friction is introduced. No variety of wood is very strong as compared with metals, and therefore wood is said to be easy to cut.

There is, however, another and more vital factor than strength that enters into the cutting of wood, namely, abrasion. Comparatively little power is required to sever the fibers of any wood while the tool is sharp. The condition may be short-lived, however. The abrasive action starts at once to polish away the keen edge until, sooner or later, dullness results, enough to cause poor cutting and excessive power consumption. The saw maker and user have not sufficiently appreciated the relation between sharpness and smoothness of cut. A saw tooth is ordinarily considered sharp if it appears so to the naked eye, as it is left by a file or medium-grit abrasive wheel.

The abrasiveness of different woods varies very widely, even though their other properties may not be greatly different. An example of this difference is given by the test results shown in Table 3. The test was made with a 9-in. circular saw, electrically driven and with a positive feed, the wood being rip-cut.

TABLE 3

Kind of wood	Tensile strength, lb. per sq. in.	Power, in kw. Start	Power, in kw. Finish	Amount cut, ft.	Sharpness at finish
Maple 1 in. dry	19,100	0.70	0.78	426	Sharp
Bapepa 3/4 in. [a]	17,500	0.70	1.50[b]	27	Very dull
White pine 3/4 in.	10,000	0.55	0.45	766	Very sharp

[a] A South American wood, resembling mahogany.
[b] Over 1.5. The kilowatt-meter needle was off the scale and the motor stalling.

Abrasion is a complicated phenomenon about which little is really known. The type of abrasion met with in wood cutting can be pictured as a buffing action, the abrasive being cellulose and mineral salts, including silica. To make a particular steel more resistant to abrasion, the first thought would be to make the tool harder. But here enters one of the curious features about abrasion, at least under conditions met with in wood cutting. When the saw speed is very low, as in hand operations, greater saw hardness may result in increasing by several times the life of the edge. When the saw speed is high, however, as with a power tool, there is little difference in life between the harder and the softer tool. To obtain a marked improvement in edge-holding with fast-running tools it is necessary to change the steel composition.

The difference in tool life with change in tool speed and tool hardness, but with the same composition steel, is shown in Table 4. Although the material cut was a wood substitute, the comparison will hold for wood. The saws were made of 0.90 carbon-tool steel.

TABLE 4

Saw hardness, Rockwell (C)	Material sawed to dullness: Saw speed 150 f.p.m.	Material sawed to dullness: Saw speed 6000 f.p.m.
48 (filable)	In 8 ft.	In 5 ft.
62 (file hard)	In 36 ft.	In 6 ft.

It will be noted that at low speeds the increased life brought

about by the greater hardness was 350 per cent, whereas at high speed the increase was only 20 per cent.

Another example of the effect of tool hardness on the edge-holding properties is shown in Table 5. The tool was a knife used in an automatic handle-turning lathe and made from 0.60 carbon-tool steel. The wood cut was maple, finishing $1^1/_4$ in. in diameter. The speed was 3200 r.p.m.

TABLE 5

Hardness Rockwell (C)	Number of handles per grind
61–62	366
53–55	330
48–50	306
44–46	300

It is interesting to compare the results shown in Table 5 with those of Table 4. The increase of 12 to 14 points in Rockwell hardness between the 48 and the 62 hardness tools resulted in both cases in an increased tool life of 20 per cent. It is seen from Table 5 that the extreme difference in tool hardness between the hardest, which was tempered at 350 deg. fahr. and file hard, and the softest, which was tempered at 900 deg. fahr. and easily filed, resulted in a difference in tool life of only 22 per cent.

The effect of changing the composition of the steel is shown in Table 6. The tools were operating on the same job as that illustrated in Table 5, except that the wood cut was cherry. Their compositions are shown in Table 7.

TABLE 6

Steel	Hardness Rockwell (C)	Number of pieces per grind	Increased life over (*a*), per cent
(*a*)	61–62	276	
(*b*)	61–62	504	82
(*c*)	61–62	552	100
(*d*)	61–62	480	73
(*e*)	61–62	480	73
(*f*)	62–63	1020	270

It will be seen from Table 6 that the increase of carbon content, the hardness being the same, in the plain carbon steels (*a*), (*b*), and (*c*) resulted in a consistent increase in tool life, up to 100 per cent. Also that the medium-nickel–low-chromium-vanadium steel (*d*) gave about the same results as did the plain carbon steel (*b*) of like carbon content. The fact that the low-nickel–low-molybdenum steel (*e*) with comparatively low carbon, 0.76 per cent, had the same life as the higher carbon steel (*d*) is interesting. It would indicate that a steel of greater toughness, resulting from the lower carbon, can be had with no sacrifice in cutting life. The high-speed steel tool (*f*) had an unusually low life for this kind of steel, most saw and knife applications showing about a 10 to 1 ratio in its favor as compared with carbon-tool steel.

A significant point indicated by the foregoing tests is the far greater effect produced by a difference in carbon, as shown in Table 6, than by a difference in hardness, as reported in Table 5. The difference in hardness between high and low temper gave only a 22 per cent variation in tool life, whereas difference in carbon between high and low gave 100 per cent.

TABLE 7

Steel	Composition							
	C	Mn	Si	Ni	Cr	V	Wo	Mo
(*a*)	0.60	0.53	..	..	..	..	...	..
(*b*)	0.99	0.36	..	..	..	..	...	..
(*c*)	1.13	0.34	..	..	..	..	...	..
(*d*)	0.93	0.33	0.21	1.35	0.36	0.21	...	..
(*e*)	0.76	0.34	0.22	0.68	..	..	...	0.14
(*f*)	0.66	0.31	0.20	..	3.98	1.05	18.15	..

It must be borne in mind that the conditions under which the saw or knife is operating may have a marked influence on the comparative performance of different steels. In certain applications a carbon or low-alloy tool steel will give no service. Where the temperature developed in cutting is really high, only a material (as high-speed steel, for example) which remains hard at these temperatures will do the work. For some classes of work a steel having a tungsten content of from 2 to 8 per cent will give better service than without the addition, although not so good results as will high-speed steel. It can be taken as axiomatic that the better a steel resists softening at elevated temperatures and also the type of abrasive wear met with in wood cutting, the better service it will give.

We can make use of the improved properties of the steels in either of two ways: to obtain longer tool life under the same conditions of operation, or to maintain the same tool life but at a substantially higher rate of production. In either case, to effect a major change in tool performance, our present-day steels all fall into either one of two classes, high-speed steels and all other tool steels.

For wood as well as for metal cutting, high-speed steel is the best type known today. It is enough superior to warrant wider application to wood sawing than is now the case. The wood-working industry should profit more from the example set by the metal-working industry and establish, by fundamental research, more data about cutting its product. If the results of such studies were at hand, we would not only be making more use of the best steels we now possess, but we also would be better prepared for new developments in tool materials.

The recently developed cemented tungsten-carbide tool material will effect profound changes in wood cutting. It possesses toughness, together with hardness and resistance to abrasion, such that cutting speeds and quality of cut heretofore impossible will be obtained. A typical example of the results being obtained with this material operating on a machine designed for high-speed steel tools is given in Table 8. The job was planning, tonguing, and grooving fireproofed oak flooring. The 10-in. cutter head was operating at a speed of 3600 r.p.m. with a feed of 80 f.p.m.

TABLE 8

Knife material	Production per grind, feet	Increase, per cent
High-speed steel	400	
"Carboloy"	20,000	4900

Other applications of cemented tungsten-carbide knives and also saws are giving results similar to those indicated. The possibilities of such new materials cannot be ignored and can hardly be overestimated by the wood-cutting industry.

Conclusions

To one who has been in close touch with the recent developments of saw steels there is little in sight to promise further marked improvements in the solid-tooth type of saw. The two conflicting requirements for the one steel for the blade and for the cutting edge would seem to take us for further advances beyond the nature of this metal.

Up to 15 years ago, or thereabouts, the saw maker for a hundred years had spent an unbelievable amount of effort and ingenuity in effecting the best compromise possible with carbon steels. The next step was the development of an alloy-steel saw, with the object of producing a stronger blade and also to effect all possible improvement in edge holding. A much better blade resulted, but even with the aid of modern laboratories and metallurgical knowledge, there was little improvement in edge-holding properties, far from keeping pace with the potentialities of the new blade.

The next logical step is therefore to depart from the old-type saw and use inserted teeth made from materials known to be well adapted for the function of cutting wood. In the field of metal cutting this same step forward was made years ago.

Future changes in saw design, so as to put to use metallurgical discoveries, will require the best thought of saw user, ma-

chine builder, and saw maker. Research and untiring effort form the only road today to accomplishment, but the benefits in store for the woodworking industry are worth the struggle.

ACKNOWLEDGMENT

The author expresses his appreciation and thanks to Mr. S. Horace Disston, Vice-President, Henry Disston & Sons, Inc., for suggestions which have been most helpful in the preparation of this paper.

Discussion

J. P. POTTER.[2] Most of the experience that the writer has had in the woodworking field has been with saws and cutters. He has seen it pass from the old carbon steel to the high-tungsten steels and come on through to the cemented tungsten-carbide steel. A great many woodworking tools are made today without due regard to the safety of the swinging of steel and knives and saws, especially of the high frequency machine. When you go up to the neighborhood of 7200 to 9000, the centrifugal strains become very great. So far as is known, there has not been anything put on the market that offers absolute safety. The internal head or the solid-back head which was invented some 20 years ago comes the nearest to it, but it will not support a cutter projecting more than three-quarters or one inch.

Speaking primarily about saws, the prepared paper seems to be a very elaborate proposition. It needs very little addition.

The things noticed in practice regarding saws are the extreme misunderstanding on the part of those using the steel in knowing what strengths are required from it.

The matter of preparing steel for the shop is a subject which needs a lot of discussion. The stock patterns can be prepared in manufacturing plants far distant and remote from the point of manufacture and can be shipped in due time for ordinary purposes.

In plans for individuals, oftentimes one little house requires enough steel to write off the cost of the manufacture. The use of cemented tungsten-carbide steel is also costly. The writer has found that on stock runs the steel costs relatively little in proportion to the amount of wood run. It will range somewhere from 2 to 4 cents per 1000 ft. On an ordinary mold of 3 in. width, it will run with about 5 or 6 in. of steel with a 6-knife equipment all around. Those 60 in. of steel will run about 4,000,000 lineal feet of molding before they are completely worn out. That has been proved in soft woods along the Pacific Coast.

With the higher priced steel, it becomes a question. By the per cent of this paper, the steel will run so much longer than the ordinary carbon-tool steel that perhaps there would be no limit to the wear of the cuts. That brings up another problem of using this steel properly supported in very thin sections.

[2] American Woodworking Company, New York, N. Y.

AUTHOR'S CLOSURE

The points brought out by Mr. Potter are of real practical interest. The present-day use of saws and knives at higher speeds is no exception to the modern trend, and, as Mr. Potter indicates, the necessary safety precautions must be taken, not only in the design of the tool, but also in its use. A point in this connection may well be discussed briefly. An inserted-tooth saw or cutter head is usually refitted many times by the user. He must see to it that the blade or head is not only properly refitted, but also that it is maintained in good mechanical condition and balance. With higher speeds, the machine also must be kept in better shape than formerly was necessary.

In calculating the effect of centrifugal forces in rapidly revolving saws and knives, the subject should be considered from two angles. A revolving disk will have maximum stresses from centrifugal forces set up, not at the rim, but at the center hole. The stresses at the rim will be comparatively small. For example, a 16-in. blade with a 1-in. center hole revolving at 3600 r.p.m. will have stresses set up amounting to about 6000 lb. per sq. in. at the center hole and 1500 lb. per sq. in. at the rim. A 10-in. disk running at 7200 r.p.m. will result in about 9000 and 2000 lb. per sq. in. at the center and rim, respectively. These stresses are well within the factor of safety for a good steel and in themselves far from dangerous. Excessive rim strains set up by poor tooth or cutter-edge clamping methods vibrations, and accidental strains in operation are far more dangerous than those from centrifugal forces. The added danger from centrifugal forces as speeds increase does not come about as much through the greater possibility of a well-made and properly operated tool being broken by these forces as by reason of the much greater damage done if breakage occurs. The energy of a piece flying off the rim of a saw or cutter increases as the square of the r.p.m. of the tool. Thus doubling the speed of the rim increases the energy by four times, and protective measures must be bettered correspondingly.

Mr. Potter properly raises the question of economy in using expensive steels or tool materials for working ordinary woods. The relative tool costs must always be considered. An extreme case where a cheap steel is clearly good enough is where short runs of special molding shapes are made from soft wood with no particular requirement for surface finish. It would be poor economy to make an expensive tool only to discard it after a few hours' run. Special material like cemented tungsten carbide for woodworking is finding its first applications to production jobs where the wood is particularly abrasive and where machine down time or other factors are important. Users are, however, beginning to see its possibilities for more of their regular production work on common woods where accuracy or smooth finish is desired. Its use will grow as experience indicates its proper applications.

WDI-53-6

Need of Forest-Products Engineers in the Woodworking Industries

By NELSON C. BROWN[1] and RAYMOND J. HOYLE,[2] SYRACUSE, N. Y.

For many years it was thought that a forester had to do only with the growing and handling of the forest crop. Now the forest schools recognize that the utilization of wood products, including the conditioning, remanufacturing and merchandising of various forms, is just as important a part of forestry. To meet this situation, the curricula of some of the forest schools are changing. The woodworking industries have problems that need solution. Trained men who know about the technological features of wood are needed. In some way the forest schools have failed to train men to meet the demand in these wood-using industries. It is also felt that the industries have failed to cooperate with the schools in order to have the courses fit in with their needs. Closer relationship, however, is growing up between the industries and the forest schools.



Nelson C. Brown



Raymond J. Hoyle

As a general thing the woodworking industries are not showing as much progress as many of the other important industries. Their machines, methods, conditions of raw material, cost systems, and merchandising as a rule are years, if not decades, behind those of many well-known industries. The mortality among these woodworking industries is very heavy and will continue so until men who are especially trained for this field of work are sought by these industries. At this point it is fair to say that there are some industries in the woodworking field that are up-to-date, and their example should be followed.

Wood Industries Lack Trained Men

Why is it that men have not been trained for the woodworking industries? The answer is, that the schools where these men might be prepared have not realized that there was a field for such training, and the industries that need these men have apparently believed for the most part that years of experience in the industry was the most desirable kind of preparation. The reason why wood-using industries have not shown more progress can be laid at the doors of the industry as well as of the schools of higher education.

Educational Facilities Lacking for Most Engineers

Men trained in schools of engineering and schools of forestry are found in this industry, but the number is comparatively small, and their training has not been well suited to the work involved. Engineering schools have given practically no thought to the subject of wood—its growth, structure, manufacture, distribution, and remanufacture, conditioning, and use. Some forestry schools, on the contrary, offer considerable training in these various fields of wood. Wood, unlike many other materials, is not a man-made or controlled product, and consequently no two pieces are exactly alike, even though cut from the same position within the same tree. In fact, two such pieces are often dissimilar in many respects. When it also is considered that there are over one-hundred commercial woods in this country, one can realize why it is essential for a man to have considerable training in the field of wood if he is going to solve the many problems in connection with its manufacture and use.

Broadening Scope of Forestry

The wood-using industries as a rule have not sought the graduates of forestry schools, for they were of the opinion that these men were trained to grow and manage timberlands. The forestry schools years ago were concerned primarily in training men for handling the forests, and a few schools still cling to this idea, but some schools are today training men for other phases of forestry. Forestry is a young and growing profession and frequently sees new branches of the work and new fields of specialization. Forestry is such a very large field that one cannot expect to be trained for more than one of its branches. As the engineer must be trained in the chemical, mechanical, electrical, civil, or other separate branches so must the forester be trained in one of the several distinct branches of forestry.

It is unfortunate, in a way, that the public has come to think that forestry schools are training men only for the one field, viz., timber growing and managing, because when a forestry graduate applies to a wood-using industry for a job, the student is generally regarded as one who is prepared only for woods work. If the

---

[1] Acting Dean, New York State College of Forestry. Mr. Brown received his B.A. and M.A. from Yale. His background of experience includes U. S. Forest Service, 1908–1911; Professor in charge of Forest Utilization, New York State College of Forestry, 1912–1917 and 1921–1930; U. S. Trade Commissioner in Europe, 1917–1918; Adviser to Czecho-Slovak Republic on forest management; partner and manager of the American Woods Export Association, and member of the New York State Reforestation Commission. He is the author of "Forest Products, Their Manufacture and Use," 1919; "The Lumber Market in Italy, Spain, and Portugal," 1919–1920; "The American Lumber Industry," 1923; co-author of "Elements of Forestry," 1914. Mr. Brown has contributed many articles on various phases of the lumber industry.

[2] Professor of Forest Utilization, New York State College of Forestry. Mr. Hoyle received his B.S. from the New York State College of Forestry at Syracuse in 1917 and M.S. in Economics from Syracuse University in 1930. He has been a member of the New York State College of Forestry faculty since 1918. Mr. Hoyle is author of the following college publications: "Wood Using Industries of New York," 1921 and 1928; "The Manufacture and Use of Small Dimension Stock," 1927. During vacation periods he has been connected with the U. S. Forest Service, Hammond Lumber Co. of California, U. S. Forest Products Laboratory, National Committee on Wood Utilization, and the West Coast Lumbermen's Association. Mr. Hoyle has contributed various articles to trade journals on the subjects of marketing, uses of wood, and improved practices in the lumber and woodworking industries.

Presented at the Fifth National Wood Industries Meeting, New York, N. Y., Oct. 16 and 17, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

forestry schools that train men for the wood-using industries gave these men an engineering degree and called them "wood engineers" or some other name that would indicate to a prospective employer something of the nature of their training, it would be much easier for them to get jobs and to be properly placed in the industry.

Utilization of Forest Products

There are so many phases of the utilization side of forestry alone that the forest schools have found difficulty in choosing the various courses that would adequately prepare men for these several fields in four years. Logging, lumber manufacture, merchandising, remanufacture, conditioning of wood, and pulp and paper manufacture are all distinct fields for specialization. The wood-using industries (the field about which this paper primarily is concerned) is no doubt the largest of these divisions and should afford the greatest single outlet for graduates specializing in the consumption phase of forestry. If a student definitely knows that he wants to become, for example, a logging engineer or a sawmill manager, his courses should be chosen accordingly. Since most students do not know what special field they want to enter, emphasis should naturally be laid on the largest field. As yet the several fields of work for the most part are not clearly defined nor is there much assurance that the industry will afford a job in these several fields. Until the industry calls for men of a definite training, the schools will have to prepare men along rather broad general lines.

Experience at Syracuse

The New York State College of Forestry at Syracuse has recognized a need for men especially trained for the utilization side of forestry. Since its inception this school has offered courses designed to prepare men for this field. Several times in the past 15 years the school has added new courses or changed old ones to meet the changing demands of the lumber industry and to meet new developments and tendencies. At the present time those students who feel that they want to enter some phase of wood utilization spend the first two years in studying English, modern languages, mathematics, botany, chemistry, physics, zoology, entomology, drafting, dendrology, plane surveying, and timber estimating. (These same courses are also taken by nearly all students preparing for any phase of forestry, since they are fundamental and of a more general nature.) The third and fourth years are spent in specializing. The courses taken in the third year include one semester of elementary accounting, one year of economics, one semester of technical and report writing and business correspondence, one semester of history of forestry, one year of silviculture including the foundation work and forest propagation, one semester of logging and one of lumber manufacture, one semester of forest products other than lumber, one semester of wood structure and identification, and one semester of railroad transportation. Courses taken the fourth year include one semester of forest pathology, timber physics, business law, public speaking, portable milling and woodlot logging, lumber salesmanship, business methods, new developments and problems of the lumber industry, the economics of production, distribution and consumption of lumber, and one year each of wood preservation and dry kiln engineering. If a student is quite certain of the field of work he hopes to enter, he is allowed to substitute four courses in engineering, business economics, forestry, or science for certain specified courses in the fourth year.

Educational Courses Require Balance

If these courses have been noted carefully, it will be seen that the student has not only received good fundamental training in many basic subjects, but also has been given a somewhat specialized training for the problems and work of the lumber industry. He will understand the growth and behavior of trees and the economic and technical problems of manufacture and use. He will know the technical, physical, and mechanical properties of woods and why these woods can or cannot be used for some purposes in certain markets. The student has an understanding of the conditions, problems, and costs involved in converting a forest into lumber or other forest products, and the problems of merchandising, remanufacturing, and using this lumber in distant markets. An experimental dry kiln affords opportunity of practical kiln operation where the student learns in a one-year course of the actual practice as well as the theory and problems involved. Another year's course of study and actual operation of the experimental wood-preservation plant gives the student specific training for a definite field of work. If desired, additional courses may be taken in chemistry or pathology to supplement or round out his training in the preservation field. In fact, the student, because of the opportunity of electing several different courses, can take work that will give extra training for any one of the several fields of work. Courses elected in business and economics give a background and insight into these two phases of the industry, while courses in engineering are of benefit particularly to one who enters the wood-using industry field.

Practical Application

At the end of the second year, three months are spent in a camp in the woods where the classroom theory is applied and practiced. This practical application and the long hours of work are of great value. At the end of the third year, the student is given five months' vacation and encouraged to get a job in a mill, wood-using industry, wood-preservation plant, a retail yard, or in some phase of the lumber industry where he will get practical experience. During the senior year an especially arranged trip is taken to some place in the South where pine and hardwood logging operations are studied in detail and under supervision, and where other phases of the lumber industry are also incidentally visited and studied.

Industrial Appreciation of Specialized Executive Training

At this point one may say that this is all very good training, but that he does not see how these men can be of particular value in one's own organization, which may be a furniture plant or an auto-body plant. The best answer to such a question is that some of these industries are taking these men and are finding them advantageous to the organization. These men are progressing and are securing the better positions and becoming the executives. The industry as well as the student must have some vision and must realize that four years of training in a rather general field have not prepared a man to make a success of any job that he may be given. It must be remembered that while he has had opportunity of operating an experimental kiln or wood-treating plant or has had some practical experience during vacation periods, he is nevertheless only a student who has a good background for further training. The large progressive industries of this country, and many smaller ones too, often take the college men into the organization for a special course of training which will point or adapt them to their particular work. Judgment should be used in working the man into the organization.

Industrial Demand Stimulates Educational Output

The more rapidly the woodworking industry calls for men of special training, just that much more rapidly will the forestry schools be able to give courses that will be more applicable to this industry. It would seem that preparation for this work will

soon include such courses as kinematics, machine design and layout, mechanics, power, shop practice, factory management, economics of manufacturing, industrial engineering and investigations, time and motion studies, a detailed study of the uses of wood, glues, plywoods, wood finishes, and industrial psychology. These fourteen additional courses would have to be given in place of some of those already outlined. Those least applicable to the problems of the woodworking industry would have to be eliminated. Some of those eliminated might be the botany courses, zoology, entomology, surveying, mensuration, forest history, planting, lumber salesmanship, and one course in timber preservation.

Course of Study Recommended

With this change in courses a student will have had basic training courses, such as English, mathematics, languages, chemistry, physics, economics, and speech. He will have had such business courses as accounting, merchandising, transportation, factory management, industrial psychology, and economics of manufacturing. The courses pertaining to wood would be growth of forests, forest economics and distribution, wood-technology courses, timber physics, forest pathology, logging and lumber manufacture, forest products other than lumber, problems of the lumber and woodworking industries, seasoning and preservation of wood, uses of woods and grades, glues, plywoods, and finishes. The engineering courses would be mechanics, kinematics, machine design and layout, power, shop practice, industrial engineering and investigations, and time and motion studies.

This variety of courses when summarized shows ten general courses, six business courses, fifteen courses pertaining to wood, and nine to engineering. This gives a total of forty courses, all of which would be of three hours per semester, except English, mathematics, modern language, and wood technology. These four subjects would be of three hours each for two semesters. If one desires more mathematics, of course there could be some elimination of modern languages, or other similar changes could be arranged. It should be noted also that forty courses for four years is an average of 15 hours' credit per semester, which is a rather light schedule. It is therefore possible to add one course each semester or eight courses in four years. These eight courses could all be in one of the four groups of subjects or more or less evenly spread among all groups, depending on how much importance was placed on certain subjects by different individuals.

The authors feel that the proportion shown in the foregoing is about right, but that is a matter of opinion. Having been engaged in training men for this field of work for many years and knowing the work and the problems that these men have to meet in this line of work and knowing further of the desires of managers in hiring young men for this work, they feel fairly certain of the needs in these four groups of study.

Specialized Training

Wood, machines and equipment, purchasing, merchandising, and costs are the big problems of a woodworking factory. One man cannot be an expert in all of these fields, but he can know much about the most important ones and can have a general idea of the least important. Merchandising and cost-accounting men are being well prepared by many schools, but what school prepares a man who understands wood from the forest to the finished product? When the price of white oak of a certain grade reaches a certain point, is it advisable to buy another grade of this wood or can another wood be substituted, and what grade in this substitute wood should be used? If a door that has been hung six months pulls apart at the joints or cracks across the panels, what is the matter? Why did the door next to it show no such tendencies? These are typical of hundreds of such problems that for the most part are unsolved or only partly solved because the leaders in this industry have not been trained in the subject of wood. While the problems of a woodworking plant are of several natures, as already indicated, the chief problem and the problem that is least understood is generally that of its raw material, wood.

Cooperation Imperative Between Industry and Schools

The industry and the forestry schools can well afford to become better acquainted. The industry knows too little of the work that the schools are doing, and it is offering few or no suggestions of how better to prepare men for its use. The faculties of the schools have had too little practical contact with the industry and seem to hesitate to keep that contact frequently renewed and alert to the problems and conditions. The progress of the woodworking industry depends upon the training of the young men who are entering this industry.

WDI-53-7

Developments in the Stabilization of Painting Practice for Wood

By F. L. BROWNE,[1] MADISON, WIS.

Painting practice is changing from a craft to a part of engineering. The Forest Products Laboratory has tested the painting characteristics of softwoods. The tests included edge-grain and flat-grain boards, light woods consisting mostly of spring-wood and heavy woods with wide bands of summer-wood, white lead and prepared paint, various combinations of paint, linseed oil, thinner, and drier for the priming coat, and aluminum paint as a first coat. Conditions of dryness and of humidity entered into the tests. The basic factors responsible for the chief uncertainties of paint behavior were studied, and the way is pointed for a possible stabilization of painting practice for wood through further research.



PAINTING practice is in course of transition from a craft to a part of engineering. (1)[2] The breakdown of the apprenticeship system, the multiplication of new materials available for painting, the increasing variety of the surfaces to be painted, the accelerating tempo of building and manufacturing operations, and the rising standards of acceptable service have overwhelmed the old-fashioned painter. However, the engineer is not yet fully prepared for his new responsibilities because the background of technical data on painting practice is fragmentary and largely unreliable, and means of measuring the serviceableness of paint coatings is lacking. A specification for a durable exterior paint finish cannot be written with the same assurance of success that is possible in designing a structure to carry a given maximum load. Quality in a paint job is judged primarily by the costliness of the materials and the labor of applying them rather than by dependable yardsticks of performance.

Under the circumstances it is not surprising that painting practice is far from stabilized. Many kinds of paint are sold for similar uses, and opinions about their relative merits differ bewilderingly. Details of applying paint vary widely. New woodwork is sometimes painted with two coats, sometimes with three. Admixtures of oil, thinner, and drier are variously recommended. There is uncertainty about the limitations of weather, condition of the surface, and drying time between coats within which painting may be done safely. When paint jobs turn out unsatisfactorily, the cause of failure is frequently difficult to determine. All too frequently coatings give way soon after they have been applied, even though conscientious efforts were made to do first-class work. When reasonable length of service is attained, it is difficult to determine when a coating ceases to function properly and requires renewal. Opinions then differ about the way to prepare the surface for repainting and how to repaint it.

A major source of confusion about painting practice for exterior woodwork arises from the marked variation in durability of coatings on different kinds of wood. The fact has long been recognized, but statements about the kinds of wood that give good or bad service and what should be done about it have been strikingly contradictory. Some have thought that certain woods are inherently lacking in paintability and should not be used in places where painting is important. Others have held that each kind of wood requires an appropriate adjustment of the proportions of pigment, oil, and thinner in the priming coat, but the formulas for each kind of wood have never been stated definitely, that being left to the painter's judgment. Still others have maintained that for some woods the turpentine in the primer should be replaced by volatile liquids of greater solvent power. Many special priming paints have been proposed for use on woods upon which paints fail soonest.

Forest Products Laboratory Tests

To provide reliable information about such moot questions the Forest Products Laboratory in 1924 began an experimental study of the painting characteristics of softwoods. (2) Test fences were established at 11 places in the United States representing a great range in climates. On the fences, panels of 18 species of softwoods were exposed and painted. In general, at every station there were four panels of each species, two made with flat-grain boards and two with edge-grain boards. Of these one panel of flat-grain and one of edge-grain boards were painted with three coats of white-lead-paste paint and the other two panels with three coats of lead and zinc prepared paint.

In the 1924 tests no attempt was made to adjust the proportions of pigment, oil, and thinner in the priming coat according to the kind of wood painted. All woods were painted in exactly the same way. The mixtures of white-lead-paste paint were:

	First coat	Second coat	Third coat
Paste white lead (92 per cent pigment), lb	100	100	100
Raw linseed oil, gal	4	1½	3½
Turpentine, gal	2	1½	1
Paint drier, pt	1	1	1

The lead and zinc prepared paint consisted of:

Pigment, 64 per cent by weight, composed of:	Per cent by weight
Basic carbonate white lead	60
Zinc oxide	30
Asbestine	10

[1] Senior Chemist, Forest Products Laboratory, maintained at Madison, Wis., by the Forest Service, U. S. Department of Agriculture, in cooperation with the University of Wisconsin. Dr. Browne joined the staff of the Forest Products Laboratory in 1918, when he engaged in developing water-resistant glues, particularly for use in aircraft. When investigations of wood finishing were begun in 1922, he was placed in direct charge of the experiments. He is a graduate of the course in chemistry at Cornell University and obtained the doctorate at the University of Wisconsin in colloid chemistry. He is a member of the American Chemical Society and of the Society of Sigma XI, a fellow of the American Association for the Advancement of Science, and an assistant editor of *Chemical Abstracts*.

[2] The numbers in parentheses throughout the paper refer to the List of References at the end.

Presented at the Fifth National Wood Industries Meeting, New York, N. Y., Oct. 16 and 17, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Vehicle, 36 per cent by weight, composed of:	Per cent by weight
Raw linseed oil	90
Turpentine	5
Liquid drier	5

For the priming coat 1¼ pints of raw linseed oil and 2 pints of turpentine were added to a gallon of this paint. For the second coat 2½ pints of turpentine were added to a gallon. For the third coat the paint was applied without any addition.

All test stations were visited at least once each year by the same inspector. During the first year or so the appearance of the paint coatings changed materially. (3) The gloss faded, the white surfaces became dirty, then the coatings began to disintegrate at the surface in the form of a fine powder and some of the dirt became dislodged, and finally the coatings began to check in fine, presumably superficial fissures. There were interesting differences in appearance between coatings of the two kinds of paint and between the different exposure stations, but the kind of wood beneath the coating had very little effect upon its appearance. However, as the tests continued, other changes set in that varied markedly with the kind of wood. These changes were of two sorts; namely, in the integrity of the coating, and in that of the wood itself. Sometimes deep slits developed in the coating at the edge of which the coating curled up and broke off in little pieces, laying the wood beneath bare. Sometimes paint checks finally became deep, interlacing cracks, and little areas of coating outlined by the cracks fell off bodily, leaving the wood bare. Such changes were recorded as deficiencies in the integrity of the coating. But sometimes checks opened in the wood where otherwise the coating seemed to be intact. Such changes were recorded as inadequate protection by the coating against wood weathering.

Results of the 1924 Tests

The most important facts revealed by the 1924 tests were the following: (4)

Summer-Wood and Spring-Wood. The variable behavior of paint coatings on the softwoods is due primarily to difference in behavior on summer-wood and spring-wood. Failure of aged coatings in integrity usually begins over summer-wood long before it sets in over the neighboring spring-wood, and therefore failure in coating integrity proceeds most rapidly on boards that contain the greatest amount of summer-wood.

Summer-wood is the harder, darker-colored part of the annual growth ring in the wood. Structurally it differs from spring-wood in that the wood cells are flattened and have thicker walls and smaller cavities than the wood cells in spring-wood. Summer-wood is roughly three times as heavy as spring-wood. Experience in impregnating wood with liquids proves that the heavy summer-wood is more easily and deeply penetrated by liquids than the light spring-wood. Experiment proves that the liquids in oil paint follow the same general rule and penetrate more deeply into summer-wood than they do into spring-wood, although there has long been a contrary impression among painters. The pigments of oil paint are too large to penetrate beyond the cavities of those wood cells that were cut open in forming the surface of the board. Like other liquids, resins exude more easily from summer-wood than they do from spring-wood, but there is no evidence that summer-wood actually contains more resin than spring-wood.

Aged coatings apparently do not adhere to wood substances as glue does, but cling to the wood by a mechanical embedding of the paint in the open cell cavities at the surface, much as plaster clings to lath. Summer-wood, with its small cavities, furnishes inadequate anchorage so that small pieces of aged coating easily become detached and fall off. However, if the band of summer-wood is very narrow, the little pieces of coating may retain sufficient grip in the spring-wood on either side to remain in place, arching over the summer-wood.

Some Woods Require More Effective Protection Than Others. A second factor causing variable behavior of coatings on softwoods is the fact that some woods require more adequate paint protection to prevent wood-checking, which is the first sign of wood weathering. (5) Test panels were often observed in which the wood had begun to check although the coating otherwise seemed to remain intact. The woods on which this happened were the kinds that quickly develop conspicuous checks and cracks when exposed to the weather without paint protection. Frequently the wood checks became centers from which failure of the coating in integrity progressed.

Edge-Grain and Flat-Grain Boards. Coatings remain intact longer on edge-grain boards than they do on flat-grain boards of otherwise similar wood. The difference in behavior is accounted for by the disposition of summer-wood at the surfaces of the two kinds of boards. Since the surface of an edge-grain board lies roughly in a plane along a radius of the tree, the bands of summer-wood are reduced to the narrowest width possible. The surface of a flat-grain board, on the other hand, is in a plane approximately tangent to some of the growth rings of the tree, so that the bands of summer-wood at the surface of the board may flare out into large areas.

Edge-grain boards weather-check less easily than flat-grain boards, doubtless because they swell and shrink less in width, and therefore coatings continue to furnish sufficient protection longer on the edge-grain boards.

Bark Side and Pith Side of Flat-Grain Boards. One side of a flat-grain board is always nearer the bark and the other nearer the pith of the tree from which it was cut. The bark side makes a better surface for painting than the pith side. On the pith side of some boards the summer-wood of one annual ring may separate from the spring-wood of the next ring and curl upward in a sharp, easily splintered edge. Paint coatings are often unable to prevent such grain shelling, and consequently the coating may be badly disrupted.

Density and Ring Width. Since summer-wood is much heavier than spring-wood, the density of a piece of softwood is a fairly good measure of the proportion of summer-wood in it and therefore of its painting characteristics. Paint fails sooner on heavy boards, because they contain more summer-wood than light ones. However, another factor must also be considered—namely, the width of the annual growth rings (usually recorded as the number of rings per radial inch of the log). When the growth rings are narrow (many rings per inch), there may be a fairly high proportion of summer-wood, and yet the bands may be narrow. Such wood holds paint better than wood of equal or slightly lower density in which the growth rings, and consequently the bands of summer-wood, are wide.

The lumber of many species varies widely in density and in ring width, and the painting characteristics vary correspondingly. Any classification of species with respect to painting characteristics is therefore a rough approximation at best and involves much overlapping; that is, light, narrow-ringed boards of a given species may be superior, from the point of view of painting, to heavy or wide-ringed boards of species assigned higher positions in the classification. In general, edge-grain boards of any species are as good or better for painting than flat-grain boards of the species assigned to the next higher class. Subject to these reservations, the softwoods were classified for painting characteristics on the basis of the 1924 tests as set forth in Table 1, which lists also the relative amounts of the species cut into lumber in 1927.

Lumber Grade and Knots. The principal defects that form the basis for grading lumber are disadvantageous from the point of view of painting. Coatings over knots often discolor, crack, and fall off, and the wood of the knots checks conspicuously. Pitch often exudes through coatings over pitch pockets, pitch streaks, or pitchy wood. Low-grade lumber is therefore less desirable for painting than high-grade lumber of the same species unless the low-grade lumber is remanufactured in such a way that the defects are cut out. Even then the advantage may still lie with the high-grade lumber, because as a rule the high-grade lumber is cut from the outer portions of the log where the growth rings are usually narrower than they are nearer the center of the log.

TABLE 1 CLASSIFICATION OF WOODS FOR PAINTING CHARACTERISTICS AND THE ANNUAL PRODUCTION OF THESE WOODS AS LUMBER

Classification, Using U. S. Forest Service Common and Botanical Names	Percentage of all softwood lumber cut in 1927
Group 1—Woods on which coatings served longest, both in integrity and in protection:	
Cedar, Alaska; *Chamaecyparis nootkatensis*	1
Cedar, Port Orford; *Chamaecyparis lawsoniana*	
Cedar, Western red; *Thuja plicata*	
Cypress, Southern; *Taxodium distichum*	2
Redwood; *Sequoia sempervirens*	2
Group 2—Woods on which coatings failed in protection sooner than on woods of Group 1:	
Pine, Northern white; *Pinus strobus*	5
Pine, Western white; *Pinus monticola*	
Pine, sugar; *Pinus lambertiana*	1
Group 3—Woods on which coatings failed in both integrity and protection sooner than on woods of Group 1:	
Fir, white; *Abies concolor*	1
Hemlock, Eastern; *Tsuga canadensis*	7
Hemlock, Western; *Tsuga heterophylla*	
Pine, Western yellow; *Pinus ponderosa*	10
Spruce, Eastern; *Picea spp.*	2
Spruce, Sitka; *Picea sitchensis*	
Group 4—Woods on which coatings failed soonest, especially in integrity:	
Douglas fir; *Pseudotsuga taxifolia*	30
Larch, Western; *Larix occidentalis*	1
Pine, Southern yellow; *Pinus spp.*	38

Heartwood and Sapwood. Few differences in the behavior of coatings on heartwood and sapwood have been observed so far. In the few cases that have occurred, the advantage has sometimes been with the heartwood and at other times with the sapwood.

Extractive Substances Characteristic of Species. Although the primary cause of variation in the behavior of coatings on different species is found in the amount and distribution of summer-wood, a factor of secondary importance is the presence in certain woods of characteristic substances that can be separated from the woods by extraction with suitable solvents. Thus cypress and redwood each contain a substance peculiar to the species that affects the behavior of paint. When paint is applied to either of these woods while they are wet with water, the coatings will not harden until the moisture content has been reduced to a reasonable amount. On the other hand, coatings cling to the summer-wood of redwood and cypress somewhat longer than they do to the summer-wood of most other species. It seems likely that the extractives in redwood and cypress have a favorable effect on the durability of coatings.

All pines contain rosin or a similar oil-soluble acidic substance. Free organic acids in linseed oil are known to be harmful to paints containing zinc oxide, but not so harmful to straight white-lead paint. It was therefore to be expected that the lead and zinc paint would not prove quite so durable on the pines as it did on other woods of similar density and ring width, but that no such differences between pines and other woods should be observed in the case of the straight white-lead paint. The results of the tests confirmed this expectation.

Results in Different Climates. The same classification of the woods was found to apply in all climates, but the extent of variation differed materially. In climates where there are long periods of extreme dryness, failure of coatings by sloughing bodily from summer-wood takes place most rapidly and most characteristically. Such conditions accentuate the discrepancies in behavior on different woods. Where the humidity rarely falls very low, coatings cling longer to the summer-wood, the paint itself has more chance to disintegrate or wear away, and the difference between species is somewhat less marked.

Of the two paints used in the tests, one proved distinctly superior to the other on all woods in arid climates, but just as definitely inferior on all woods in the warm, humid climate of Florida. In the intermediate climates one paint often proved the more durable on the pines, while the other lasted longer on woods other than the pines.

The 1925 Primer Tests

A second series of tests was started in 1925 to determine how far the durability of coatings on woods containing much summer-wood can be improved by altering the proportions of pigment, oil, and thinner in the priming coat, by replacing turpentine with special solvents, or by using special priming paints. The test panels were divided into two halves, of which one was painted by the standard procedure followed previously in the 1924 tests, while the other half was painted by one of the special procedures to be studied. Thus comparison of new and old procedure could always be made on the same boards. The woods studied in 1925 were Southern yellow pine, Douglas fir, Western larch, Western yellow pine, Eastern hemlock, Western hemlock, and Southern cypress.

Results of the 1925 Tests

The following results were obtained in the 1925 tests: (6) *Proportions of Oil and Thinner in the Primer.* In spite of the old and widely accepted tradition that the durability of paint coatings on woods containing much summer-wood can be improved by mixing the priming coat with more than the usual amount of turpentine (and correspondingly less oil and pigment), the tests proved that the priming-coat proportions have very little effect upon the serviceableness of the coatings. If the comparisons had not been made on the same boards, no differences at all would have been detected. As it was, the durability of the coatings with turpentine-rich primers was judged slightly inferior to that with the standard primer; clearly the comparisons between species made in the 1924 tests would not have been altered in any way by changing the proportions of the priming-coat ingredients.

The result of these experiments is reasonable. Thinning with turpentine is usually recommended in the belief that deeper penetration of the paint oils will secure firmer adhesion of the coating. However, penetration to begin with is deeper in summer-wood, from which coatings come loose first, than it is in spring-wood, to which coatings cling longer. The theory is therefore discredited at the outset.

Special Solvents as Paint Thinners. Replacement of turpentine with volatile liquids that are better solvents for the oils and resins in some woods has been recommended as a means of improving paint service. Three series of experiments have been made by the Forest Products Laboratory to test the theory. The volatile liquids included gum spirits turpentine, steam-distilled turpentine, destructively distilled turpentine, turpentine oxidized by bubbling air through it, petroleum distillates ranging in volatility from gasoline to kerosene, petroleum distillates from four types of crude oil, benzol, and coal tar naphtha.

Tests were made on Southern yellow pine, cypress, and Eastern hemlock.

The effects of varying the volatile liquid were in all cases so slight that they can be of interest only from the point of view of theory. None of the liquids gave better results than turpentine on any wood. There were indications that oxidized turpentine added slightly to the durability of the coatings. The beneficial effect of turpentine must therefore be attributed to the nonvolatile residue left when turpentine evaporates in the presence of oxygen or to a desirable alteration in the drying of linseed oil in the presence of the turpentine product.

Special Priming Paints. Five special priming paints were studied in the 1925 series of tests. Of these two had very little effect on the behavior of the coatings and two proved definitely harmful to the durability. The fifth primer, aluminum paint, materially retarded the loosening and detachment of coatings from summer-wood and thereby prolonged the life of coatings

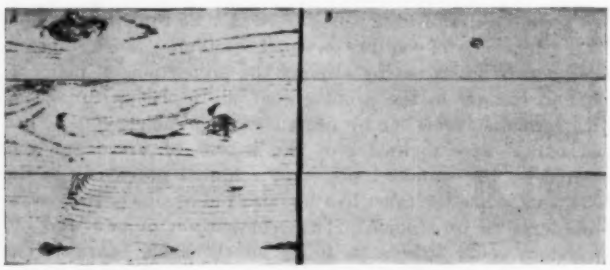


FIG. 1 CHARACTERISTIC FAILURE OF A PAINT COATING BY FALLING OFF THE SUMMER-WOOD AND THE BENEFICIAL EFFECT OF ALUMINUM PRIMER IN RETARDING SUCH BEHAVIOR

(The left-hand part of the panel, marked *A*, was painted according to the usual procedure of applying three coats of prepared paint, of which the first contained a little extra linseed oil and turpentine. The right-hand part of the panel, marked *B*, received first a priming coat of aluminum paint, followed by exactly the same second and third coats of paint that were applied on *A*.)

on woods containing much summer-wood. Fig. 1 illustrates the improvement in paint service brought about by priming with aluminum paint.

Aluminum priming paint (7) is made by adding polished aluminum bronze powder to a suitable liquid just before it is to be applied. From $1^1/_2$ to 2 lb. of the powder is stirred into the liquid; grinding in a paint mill is neither necessary nor desirable. The liquid should be either a kettle-bodied linseed oil thinned to suitable viscosity with paint thinner and drier or a long oil spar varnish (at least 80 gal. of oil per 100 lb. of resin). Ordinary spar varnish may be mixed with boiled oil, but the cost is likely to be greater. Aluminum priming paint must be thoroughly dry before the top coats of paint are applied over it.

Other priming paints or priming procedures have been suggested in recent years. Some of them are being tested by the Forest Products Laboratory or by paint manufacturers. (8)

ABNORMALLY EARLY PAINT FAILURES ON HOUSES

Much uncertainty about painting practice has arisen in the past because paints that give excellent service when exposed on test fences sometimes fail in an unreasonably short time, perhaps within a year, when applied on houses. Within the last few years observations by the Forest Products Laboratory (9) and by the West Coast Lumbermen's Association (10) have shown that such difficulties are due neither to faulty paint nor to defective lumber, but are caused by improper methods of drying the building moisture in new buildings, errors in designing the structure, faulty construction, or unusual conditions of occupation. The essential factor is water gaining access to the back of the painted boards either through leaking joints of one kind or another or by condensation from moist air coming in contact with the backs of the boards during cold weather. Many details that frequently prove sources of trouble have been described. The only sure cure for such ills lies in preventing access of water to the backs of the painted boards. No paints now on the market are proof against moisture failures of this kind, and no woods are immune to them.

CONCLUSION

Studies in recent years have contributed largely to the stabilization of painting practice for wood by removing much of the mystery and vague speculation that has beclouded the subject and also by setting forth clearly the basic factors responsible for the chief uncertainties of paint behavior.

The durability of coatings on different softwoods varies widely because aged coatings do not adhere satisfactorily to summer-wood and the amount and distribution of summer-wood in softwoods ranges between wide limits. To a limited extent paint service may be improved by selecting, for purposes where painting is important, woods that are light and have only narrow bands of summer-wood. However, painting is usually only one of many considerations in choosing a wood, and other requirements often dictate the selection of woods with wide bands of summer-wood. Furthermore, Table 1 indicates that by far the greater part of the available lumber supply is of kinds that contain prominent summer-wood.

The woods with wide summer-wood bands can be improved for painting purposes by cutting them as far as possible so that edge-grain surfaces are exposed for painting. The flat-grain boards should be surfaced so that the bark side rather than the pith side is exposed.

Further improvement in paint service on woods containing much summer-wood is not to be sought in special proportioning of the oil and thinner in the priming coat or in choosing unusual volatile liquids. The painter may remain free to employ these variables to facilitate application. Thus he may vary the proportion of thinner to control the consistency and spreading rate of the paint.

The durability of paint coatings on woods with wide bands of summer-wood can be increased by priming them with a properly made aluminum paint, given ample time to dry.

Abnormally early failures of coatings are not due to the paint nor to the wood, but rather to unhealthy moisture conditions which must be cured before acceptable service can be expected.

However, after full advantage has been taken of the foregoing means of improvement, painting practice for wood still falls short of the standards that engineers expect of the materials with which they work. Failure of coatings by bodily detachment from the wood, leaving the surface bare, is inherently unsatisfactory because it leaves a part of the wood temporarily unprotected, presents a difficult surface for repainting, and affords an insecure foundation for the new coating. The ideal coating would wear away evenly and remain continuous and firmly adherent until the need for repainting becomes clearly apparent in lack of opacity. A real stabilization of painting practice for wood cannot be reached until this ideal is achieved through research.

LIST OF REFERENCES

(1) Browne, F. L., "Wood Finishing, a Glance Ahead," *Mechanical Engineering*, 48, 1286 (1926).

(2) Browne, F. L., "Wood Finishing Studies of the Forest Products Laboratory," Proceedings, Scientific Section, Amer. Paint and Varnish Mfrs. Assn., Circular 184, p. 278 (1923). Browne, F. L., "The Painting Characteristics of Different Kinds of Wood," *ibid.*, Circular 219, p. 175 (1924).

(3) Browne, F. L., "The Painting Characteristics of Woods, II, Results After Two Years' Exposure," *ibid.*, circular 290, p. 202 (1926). Browne, F. L., "Procedure Used by the Forest Products Laboratory for Evaluating Paint Service on Wood," Proceedings, Amer. Society for Testing Materials (1930).

(4) Browne, F. L., "Properties of Wood That Determine the Service Given by Exterior Paint Coatings," Official Digest, Federation of Paint and Varnish Production Clubs, no. 95, p. 106 (1930); *Amer. Paint Journal*, April 7, 1930, p. 22; *Paint, Oil and Chemical Review*, Mar. 20, 1930, p. 9.

(5) Browne, F. L., "Paint and the First Stages in the Weathering of Wood," Proceedings, Scientific Section, Amer. Paint and Varnish Mfrs. Assn., Circular 238, p. 289 (1925).

(6) Browne, F. L., "Effect of Priming-Coat Reduction and Special Primers Upon Paint Service on Different Woods," *Industrial and Engineering Chemistry*, August, 1930.

(7) Edwards, J. D., and Wray, R. I., "Protecting Wood With Aluminum Paint," *Industrial and Engineering Chemistry*, 19, 975 (1927).

(8) Gardner, H. A., Proceedings, Scientific Section, Amer. Paint and Varnish Mfrs. Assn., Circular 231, p. 234 (1925); 281, p. 111 (1926); 314, p. 432 (1927).

(9) Browne, F. L., "Some Causes of Blistering and Peeling of Paint on House Siding," Proceedings, Scientific Section, Amer. Paint and Varnish Manufacturers Association, Circular 317, p. 480 (1927).

(10) Hartwig, O. R., "Causes of Paint Failure on Sidewalls," *ibid.*, Circular 355, p. 742 (1929).

Grading Timber for Strength

By LEYDEN N. ERICKSEN,[1] WASHINGTON, D. C.

The paper deals with the development of grading rules for structural timbers, the principles involved, effect of structural features on strength, and the application of the principles in commercial practice. Structural grading rules are based on the provisions of the American Lumber Standards, developed by the combined efforts of lumber producers, distributors, specifiers, and consumers, under the auspices of the U. S. Departments of Commerce and Agriculture. Structural material is divided into classes according to size and use: horizontal load-bearing members such as joists, planks, beams, and stringers, and vertical compression members, as posts and columns.

The strength of wood members is influenced by the density of the wood, moisture, seasoning, and the size, location, and number of strength-influencing factors present, such as knots, shakes, checks, and slope of grain. Heartwood, sapwood, and wane are more important from other points of view than for their influence on strength. Principles of structural-material grading have been applied in a practical manner by lumber-manufacturers' associations in drawing up commercial grading rules.

Safe working stresses for various grades and different conditions of exposure have been established in accordance with strength of clear wood, effect of strength-influencing factors permitted, the effect of seasoning, time of loading, and the characteristics of each species. Basic principles can also be applied to general construction material carried in retail lumber yards, thus insuring maximum strength and efficient utilization.



PUBLISHED grading rules for structural material in use at present were established by lumber manufacturers' associations upon the basis of the general provisions and structural grade examples given in the American lumber standards. Standards as a whole are recommended basic provisions covering the manufacture, grading, and marketing of softwood lumber products, including yard lumber, factory and shop lumber, and structural material. All organized groups interested in lumber, including producers, distributors, specifiers, and consumers, have had a part in their development. They were developed under the auspices of the U. S. Departments of Commerce and Agriculture, and are published in Simplified Practice Recommendation R 16-29—Lumber, otherwise known as the "Fourth Edition of the American Lumber Standards."

Section IV of the Standards applies to structural material. It is intended for use as a general guide in the preparation by lumber manufacturers of grading rules for structural material in their particular woods. Because of variations in the inherent characteristics of different species, associations must make such adaptations in these provisions as may be necessary for their species.

Classes of Structural Material

Before discussing the grading of timber for strength and the factors which must be considered it is necessary first to consider the classes into which such material is divided from the standpoint of size and use. Three main classes are so recognized: (1) Joists and planks, (2) beams and stringers, and (3) posts and timbers.

Joists and planks include material from 2 to 4 in. thick and 4 in. and up in width. Among its principal uses are floor joists, rafters, factory flooring, planks, bridge planks, and platform joists.

Beams and stringers include material 5 in. and thicker and 8 in. and over in width. They are used for heavy horizontal load-bearing members, such as bridge, trestle, and wharf members, beams in heavy mill construction, and similar uses.

Posts and timbers include material 6 in. by 6 in. and larger, and square or nearly square in cross-section. Material 5 in. square is sometimes included. It includes members used vertically as columns and such horizontal members as caps and sills.

Factors Affecting Strength

The purpose of grading timber for strength is to segregate pieces into classes having definite load-bearing values. This load may be such that it subjects the member to a bending stress, to compression either parallel with or perpendicular to the grain, to tension, or to a shear stress. All of these are considered in the grading of structural material.

The strength of wood members and their ability to resist stresses are governed by several factors. First to consider is the strength of the clear wood of the species in question, then the character of growth and wood structure in the individual piece, and finally the structural features which may be present, such as knots, pitch pockets, shakes, and checks.

Species and Growth

The strength of clear wood of all species varies with its density, which is a measure of the amount of wood substance present. If it were compressed to the point where all air spaces were excluded, the wood of all species would have the same density. Because some woods have thicker cell walls and smaller cell cavities than other woods they contain more actual wood substance per unit of volume and are correspondingly heavier and stronger. Likewise, there is a considerable range in density within a single species with a corresponding range in strength properties. All properties are not influenced in the same degree by variations in density; bending strength and stiffness increasing approximately as the first power, hardness and strength in compression across the grain as the square, and shock resistance by a slightly higher power of the density.

---

[1] Forest Products Engineer, National Lumber Manufacturers Association. Mr. Ericksen was graduated, with B.S. in Forestry, from the University of Minnesota. He was employed for five years at the U. S. Forest Products Laboratory in investigations of industrial uses for wood and the grading of hardwood lumber. He was for one year with the Western Electric Company, engaged in development work in connection with wood requirements. For the last three years he has been employed by the National Lumber Manufacturers Association in work dealing with lumber standardization and grading and lumber specifications.

Presented at the Fifth National Wood Industries Meeting, New York, N. Y., Oct. 16 and 17, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Density

In some softwoods the density may be estimated closely by the proportion of summerwood present, the darker harder portion of each ring of annual growth. Advantage is taken of this in the grading of structural material in Douglas fir and Southern pine. For example, in order to meet the structural grade provisions for these woods, pieces selected for density must contain at least one-third summerwood and have a rate of growth of at least six annual rings per inch, as measured in the prescribed manner on either end. This is illustrated in Fig. 1.



Fig. 1 Dense Six-Inch by Eight-Inch Beam or Stringer
(This piece averages 6 annual rings per inch and 51 per cent summerwood, the dark portion of the annual ring. It is a good example of "dense" material.)



Fig. 2 Close Grain Six-Inch by Six-Inch Post or Column
(This post averages 15 annual rings per inch and contains 30 per cent summerwood. It is therefore classified as "close grain" material, not having the 33⅓ per cent summerwood necessary to be classed as dense.)

Material which has less than six annual rings per inch is considered as meeting the density requirement if it contains one-half or more summerwood. Southern pine or Douglas fir meeting the summerwood and ring requirements is termed "dense."

Close Grain

In some species material having medium to narrow annual rings, as illustrated in Fig. 2, also is recognized as being superior in strength to that which has wide annual rings. This is known as "close grain" material, and although it is not as great an assurance as density, selection for rate of growth provides material suitable for many purposes. Close grain, i. e., not less than six nor more than 20 annual rings per inch, with defects limited as in the dense grades, is provided in several structural grades of Douglas fir. Similarly, the structural grades of California redwood provide for material having not less than 10 rings per inch with the maximum number limited to 25 and 35, depending upon the grade. In other softwoods material having a medium rate of growth has good strength properties depending on other factors such as knots, shakes, and slope of grain.



Fig. 3 Six-Inch by Eight-Inch Post or Column
(This piece averages 5 annual rings per inch and contains 25 per cent summerwood. It therefore is not classed either as close grain or dense material.)

Effect of Moisture on Strength

The moisture present in wood has an important bearing upon its strength properties. Wood in the standing tree or when freshly cut contains moisture in two forms, hygroscopic moisture which is contained in the cell walls, and free moisture which is contained in the cell cavities. As green wood dries it first loses the free moisture until it reaches a moisture content of from 25 to 30 per cent, at which point the free moisture is gone and the wood begins to give up its hygroscopic moisture. Changes in moisture content above this point, called the "fiber saturation

point," have little or no effect upon strength properties. As wood loses moisture below the fiber-saturation point its strength properties increase. As wood dries, however, it shrinks, and in material of structural sizes shrinkage is usually accompanied by a certain amount of checking and splitting, as a result of which some of the increased strength due to drying is lost.

In addition, timbers in use may be subjected to a range of moisture conditions, from dry interiors to the continually wet conditions encountered in some exterior uses. All of these factors are considered in establishing working stresses for different conditions of use.

Knots

Knots are among the important strength-influencing factors which are considered in structural grading rules. Their influence depends upon their size, number, and location in a piece. The quality of a knot does not influence its effect upon the strength of a wood member, and no distinction, therefore, is made in structural grades between intergrown knots and encased knots or knot-holes.

Location of Knots

Since knots in certain parts of horizontal load-bearing members (beams and stringers) have a greater weakening effect than in other parts they are more severely restricted in such locations. They have the most influence when located near the top or bottom edges in the center portion of the length. In these areas knots are required to be smaller than toward the ends or on the center line of the wide (vertical) faces. No knots are permitted anywhere in the piece, however, more than twice the size of knot permitted at the point of maximum stress.

The structural grading rules of one lumber manufacturers' association limit the knots near the ends of pieces in the same manner as in the center portion, thus assuring ample strength.

The restrictions on the sizes of knots at edges of wide faces of joists are more severe than would be required for use flat as planks. Under practically all conditions of use, however, knots along the edges of planks are more objectionable than those along the center lines. The same knot limitations are therefore applied to material to be used either as joists or as planks.

In timbers used as columns knots of the same size have the same influence regardless of their location on the faces or in the length. Knots are therefore limited in the same manner in all parts of such timbers.

Measurement of Knot Sizes

The influence of knots also varies according to the area of their projection on the cross-section of the piece. They are therefore measured in a manner which most closely approximates this influence.

On wide faces of joists and planks the mean or average diameter of a knot is taken as its size, as shown at *B* and *C* in Fig. 4. In such thin and relatively wide material, whether used on edge or flat, this is a safe measure of the influence of knots on strength and has the additional advantage of being directly applicable to yard grades of lumber. This method of measurement can be applied to spike knots as well as to round and oval knots, and excludes those which would have too great an effect on strength.

The influence of a knot on the top or bottom faces of beams, stringers, joists, and planks depends largely on the extent to which the surface fibers are cut. This is measured by the projection of the knot on a line at a right angle to the edge, and the size of such a knot is accordingly taken as its width between lines parallel to the edges of the piece, as shown at *A* in Fig. 4.

The depth of penetration of a knot on the wide or vertical face of a beam or stringer is of great importance while the influence of the number of surface fibers cut, and the amount of grain distortion, is considerably less important than on the horizontal (top or bottom) faces. The smallest diameter of such a knot is, therefore, taken as the basis for measurement, as shown at *C* in Fig. 4.

In structural members used as posts or columns two factors must be considered in connection with knots: (1) The effect upon the strength of the member in compression parallel to the grain, probably best measured by the small diameter; and (2) the influence upon the bending strength when the column begins to fail, probably best measured by the projection of the knot on a line at right angles to the edge. The method of measurement which most nearly represents average conditions is the average of the maximum and minimum diameters, which accordingly is used in measuring knot sizes in posts and columns, as illustrated at *B* and *C* in Fig. 4. In short columns the area of the cross-section occupied by the knot is of primary importance, the factor of bending strength increasing in importance as the column gets longer.

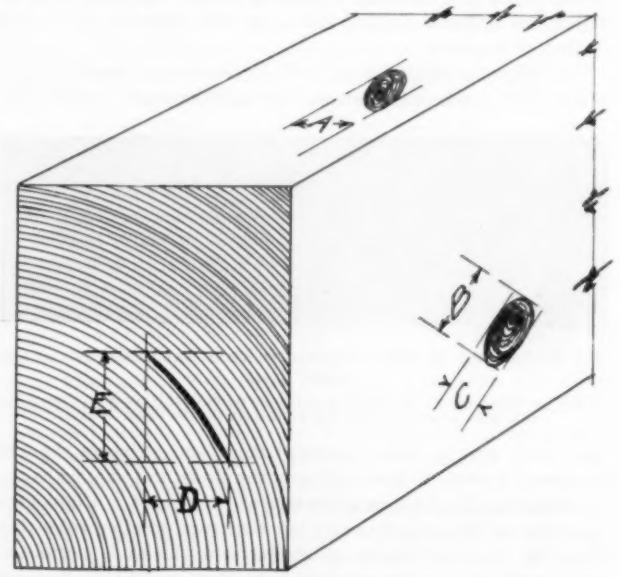


Fig. 4 Section of Structural Member Showing Method of Measuring Size of Defects

(Knots on narrow faces of beams, stringers, joists, and planks are measured between lines parallel to edges, as at *A*. Knots on wide, or vertical, faces of beams and stringers are measured on their smallest diameter, as at *C*. Knots on wide faces of joists and planks and on all faces of posts and timbers are measured on their average diameter, average of *B* and *C*. Shakes are measured on the end of the piece, in joists, planks, beams, and stringers between lines parallel to the wide faces, as at *D*; and in posts and columns in the direction in which the shake extends the greater distance, as at *E*. The illustration shows a side-cut timber, pith not present. Boxed-heart timbers contain the structural center of the tree, i. e., the pith.)

Knot Sizes

The size of knots is permitted to increase with the size of the piece. In narrow faces of beams and stringers up to 6 in. wide and in wide (vertical) faces of beams, stringers, and joists and faces of posts up to 12 in. wide, the increase is directly proportional to the width of the faces. Because of the greater distortion of grain around larger knots, with a correspondingly greater influence upon strength, the relation between sizes of permissible knots and the size of the piece in members larger than those mentioned is, however, limited to a smaller ratio, the increase being proportional to the square root of the wider faces to these sizes.

In addition to limiting the size of single knots, structural grading rules limit the number and aggregate size of knots within

certain areas of the length. In horizontal members the aggregate diameter of all knots in the center portion cannot exceed a certain proportion of the width of the face, depending on the particular grade under consideration. The aggregate diameter of knots in posts and columns is limited so that the size of all knots within any 6 in. of length does not exceed twice the diameter of the largest knots permitted in the particular grade under consideration.

Shakes and Checks

A shake is a lengthwise separation of the wood, which occurs usually between and parallel to the rings of annual growth. A check is a lengthwise separation of the wood, which occurs usually across the rings of annual growth. Shakes appear on the end sections of a piece of structural material as narrow openings, curved tangentially to follow the growth rings, while checks appear as narrow openings extending across the rings in a radial direction from the pith. Both may also appear on one or more of the four surfaces.

The effect of shakes and checks in horizontal members is to reduce their resistance to shear. Shake is measured on the ends



Fig. 5 Section of Joist Placed so That the Knot Is on the Upper Edge

(It has less influence on the strength of the member in this position than if the joist were placed so that the knot were on the lower edge.)

of a piece, and in joists, planks, beams, and stringers the size is measured between lines enclosing the shake and parallel to the wide faces of the piece, as illustrated at *D* in Fig. 4. Checks are measured in the same manner, but structural grading rules provide that no combination of shakes and checks is permitted which would reduce the strength more than would the allowable shakes.

Ordinarily shakes and checks have but little effect upon the strength of posts or columns. In such material the size of a shake is measured in the manner described, but across the end in the direction in which it extends the greater distance, as at *E* in Fig. 4.

Slope of Grain

Deviation in the direction of the grain of timbers away from a line parallel to the edges may be due to several causes. The fibers in the standing tree may incline in one direction or the other. Or, as a result of the natural taper in the tree, the grain of the wood in timbers having parallel edges may deviate. This deviation is called "slope of grain" and is expressed in terms of the amount of deviation that takes place over a certain length, as 1 in 20 or 1 in 15, meaning a deviation of 1 in. in a length of 20 in. or 1 in. in a length of 15 in. Until the slope of grain exceeds 1 in 40 there is not much reduction in strength. The effect of slope of grain greater than this depends upon the manner in which the piece is to be used and upon its location with respect to the top and bottom and in the length of the piece. It is of greatest importance on the top and bottom faces and toward the center of the length. It has less influence upon the strength of columns than upon horizontal members. Extreme cross-grain in columns, however, may cause checking as the piece dries. Slope for grain is measured over a distance sufficient to avoid the influence of short, local deviations.

Other Features Considered in Grading

In addition to the strength-influencing factors mentioned timbers have other characteristics which are considered in grading them for certain purposes. Among them are the proportion of sapwood and heartwood present, wane, and in some woods the presence of pitch pockets.

Heartwood and Sapwood

Contrary to the impression of many there is no inherent difference in strength between heartwood and sapwood, other things being equal. In timbers graded for strength, therefore, no restrictions need be made on the amount of sapwood present. Where decay resistance is necessary sapwood should be limited if the timber is to be used untreated. Sapwood absorbs wood preservatives more readily than heartwood, however, and if timbers are to be treated a large amount is desirable.

Wane

Wane is the absence of wood on the edge or corner of a timber resulting in the lack of a square corner. It is usually caused by the surface of a tree intersecting the edge of a timber. Wane ordinarily is not a serious factor from the point of view of its effect upon strength. It is considered rather because of its influence upon appearance, bearing areas, and nailing. Wane is measured according to its width as compared with the width of the faces on which it appears. While it could theoretically be permitted in all structural grades, the higher grades exclude it entirely.

Pitch Pockets

Pitch pockets are openings between the annual rings, usually containing pitch and sometimes bark. They do not have sufficient influence on strength to be classed as defects, but if a large number are present the piece should be examined closely to make sure that shakes are not present.

Commercial Application of Structural Provisions

The principles outlined have been applied in a practical manner by lumber manufacturers' associations in drawing up commercial grading rules for structural material. The permissible knots, shakes, slope of grain, and other factors are tabulated for each grade so that the exact description of the material permitted in individual grades can be judged without difficulty.

Material is graded at the mill by expert inspectors, who from long experience are able to segregate it accurately into the proper quality classes. In order to indicate to purchasers the grade of structural members many mills grade-mark structural material, stamping on each piece its exact grade. In this way the engineer or architect is assured that the material purchased and used conforms to specifications and will have the required strength properties.

Working Stresses for Structural Material

Definite working stresses have been established as a guide to the proper use for the various grades of structural material. Values are given for material exposed to three conditions of moisture: Continuously dry, occasionally wet but quickly dried, and more or less continuously damp or wet.

The values for material to be used in continuously dry conditions apply in sheltered locations and interiors not subjected to high humidities. The occasionally wet but quickly dried condition applies to such uses as bridges, trestles, grandstands, and open sheds.

The values for material to be used under more or less continually damp or wet conditions contemplate such uses as docks and wharves where the timbers are continually subjected to moisture.

The engineer preparing specifications should judge as to which of these classes best represents each project.

The subject of determination of working stresses is a subject complete in itself and can only be touched on here. The salient points considered in their derivation are outlined. Values are provided for extreme fiber in bending, compression parallel to grain, compression perpendicular to grain, horizontal shear, and modulus of elasticity.

Working stresses for material of structural sizes and quality are based on the strength of green clear wood as determined by the U. S. Forest Products Laboratory, Madison, Wis., in its comprehensive studies of the mechanical properties of woods and of the influence of various factors upon strength. In deriving them consideration has been given to the various strength-influencing factors outlined, the effect of seasoning and exposure, the effect of knots, their size, number, and location, the effect of shakes and checks, and the influence of cross or spiral grain.

In addition, allowance has been made for long-time loading, and the nature of the defects which may be characteristic of different species. Certain woods are more likely to contain shakes than others, some have their knots grouped more closely, some tend to check more in seasoning. All of these factors which study and experience have shown to be important have been considered.

An enormous amount of detailed and intensive research has been devoted to the study of the strength properties of wood and the effect of various factors on strength. The results of this work and the experience acquired through long study of wood have been used in setting up these stresses.

The influence of the strength-influencing factors varies for the different properties for which values are given. For example, the size and location of knots influence the stresses for extreme fiber in bending and in compression parallel to grain, but have little effect upon horizontal shear and compression perpendicular to grain. Shakes and checks influence horizontal shear principally and in extreme cases compression parallel to grain. Slope of grain influences the values for extreme fiber in bending and compression parallel to grain. The extent of exposure with the resulting range in moisture content has an effect upon the stress for extreme fiber in bending and compression parallel and perpendicular to the grain. The size of the piece affects the stresses for extreme fiber in bending, but not the other properties.

Density and close grain, in the woods in which they are considered in grading, permit an increase in the stresses for all properties except modulus of elasticity. Values for modulus of elasticity within a species are the same regardless of the grade, size of the piece, or exposure in use.

Working stresses are established on the basis of the lowest quality of lumber admitted in each grade. Each grade, however, contains material ranging in quality from that described up to that described for the next higher grade. Practically all of the material, therefore, could take higher stresses than those recommended. The published working stresses consequently are ultra-conservative and safe, and can be used with assurance.

Up to the present time grading rules and working stresses for structural material have been established and published by lumber manufacturers' associations for three species, Douglas fir, Southern yellow pine, and California redwood. Selection of other species, both softwoods and hardwoods, for structural purposes can be made according to the structural grade examples given in the American Lumber Standards.

These principles of grading structural material for construction uses can also be applied to stock used for such purposes as ladders, airplanes, masts, flagpoles, railroad-car sills, and crossarms. For example, slope of grain is important in lumber to be used in airplanes, where the best possible material is necessary. Likewise, in ladder stock, where failure may cause loss of life, both slope-of-grain and rate-of-growth limitations insure maximum strength.

These principles can be applied to much of the general construction lumber carried in retail lumber yards throughout the country, or by examination of the lumber on the job, material of good strength properties can be selected for certain structural parts of buildings where strength or stiffness is particularly desirable. For example, floor joists and other horizontal members can be so placed that the larger knots are on the upper edge, as in Fig. 5 rather than on the lower edge where they would have a greater effect upon the strength. It has been established that the load-bearing capacity of horizontal members of structural grade is increased up to $^1/_8$ or more when the better edge is placed down. Present working stresses are based on the assumption that each member may be used with the poorer edge down, and are consequently lower than they could be safely if care was taken to place the better edge down. This can readily be done on the job, however, thus assuring maximum strength. In some woods a considerable number of pieces in retail yards will be found to meet the density and close-grain requirements of the structural grades and thus can be depended upon to have good strength properties. By applying these principles advantage can be taken of the maximum strength of the lumber and the most efficient utilization will be effected.

Dust Explosions, With Special Reference to Woodworking Industries

By HYLTON R. BROWN,[1] WASHINGTON, D. C.

Explosions in flour mills have been common, but now the hazard is recognized in grain elevators, starch factories, sugar refineries, woodworking plants, textile mills, and other factories in which materials in the form of dust are manufactured. The woodworking industry is considered susceptible to this danger, especially since there is created a great quantity of dust from saws, planers, and sanders, and the recent manufacture of wood flour. More than thirty explosions in woodworking plants have been investigated. These indicate that wood dust is more easily ignited than some of the grain dusts. Precautionary measures are suggested, and regulations have been tentatively adopted for the prevention of dust explosions in wood-flour manufacturing establishments.



IN RECENT years the investigation of dust explosions and the development of methods of preventing dust ignitions have been extended by the Chemical Engineering Division of the Bureau of Chemistry and Soils in the U. S. Department of Agriculture to include the various industrial plants in which the hazard is known to exist. The first dust explosions recorded occurred in flour mills, and for many years such explosions were the only ones reported. Now the hazard is recognized in grain elevators, starch factories, sugar refineries, woodworking plants, textile mills, and factories in which rubber dust, sulphur dust, metallic dusts, powdered milk, chocolate, and cocoa, or other combustible materials in the form of dust are manufactured or handled.

Changes recently made in the industrial field, both in the manufacturing processes and in the utilization of by-products, have served to create a dust-explosion hazard in many plants not previously considered susceptible. The woodworking industry is an example of this type. Larger scale operations creating greater quantities of dust from saws, planers, and sanders have increased the possible hazard, and the demand for wood flour has induced some operators to reduce their scrap to powdered form. Even where the scrap is used for fuel, hogs reduce the material to small size so that it can be transported mechanically, and in so doing frequently create sufficient fine dust to form an explosive mixture.

More than thirty explosions in woodworking industries have been investigated, including explosions of wood dust, wood flour, wood pulp, and sawdust. A number of lives were lost, and many workmen were injured in these explosions which the investigations showed were caused principally by fires, overheated bearings, and electrical sources of ignition. A great many of the wood-dust explosions occurred in shavings vaults, especially ones in which the fine dust was blown into the vault with the shavings. A vault of some kind is generally provided for the collection of the refuse from the various machines in a woodworking plant, and this is the most likely spot where an explosion might be expected in the average plant.

Laboratory Tests

In the laboratory, tests have shown that wood dust is more easily ignited and produces higher pressures than some of the grain dusts which experience has taught are capable of producing tremendous pressures and completely destroying concrete and steel structures when the proper mixture of dust and air is ignited.

These same laboratory tests, although not strictly comparable because they were made on samples of wood dust as received in the laboratory without regard to particle, size, or moisture content, show a considerable variation in the results obtained with different kinds of wood dust, and also indicate that the upper and lower limits of concentration may vary greatly for different dusts.

For instance, in a comparison of uniformly fine redwood and white pine wood flour it is shown that with a concentration of 0.1 oz. per cu. ft. the redwood produced a pressure of 2 lb. per sq. in., while the white pine produced a pressure of 12 lb. per sq. in. However, when the concentration was increased to 0.4 oz. per cu. ft., the difference in pressure was not so great proportionally, being 19 and 34 lb., respectively.

In another series of tests with samples collected at Southern woodworking plants, somewhat similar variations were noted. With a concentration of 0.1 oz. per cu. ft., cypress dust produced a pressure of 15 lb. per sq. in.; a mixture of poplar and cedar dust at the same concentration produced 23 lb. pressure; and a mixture of tupelo and sweet gum under the same conditions gave a pressure of 22 lb. per sq. in. In the tests of oak, cedar, and mixed hardwood dusts, concentrations of 0.2 oz. per cu. ft. were used, and the pressures produced were 25 lb. for oak, 26 lb. for cedar, and 25 lb. for the mixed hardwood. These figures are cited not so much to draw a comparison between different wood dusts as to show the wide variation in the results. If any comparison can be drawn it is that the softwoods as a rule produce higher pressures than the hardwoods when the concentration is low, but as the concentration increases the pressures increase more rapidly for hardwood than for softwood. Of

---

[1] Engineer, Chemical Engineering Division, Bureau of Chemistry and Soils, U. S. Department of Agriculture. Mr. Brown was born in Williamsburg, Pa., on May 11, 1895. He studied flour-mill engineering and was graduated from the Pennsylvania State College in 1916. An advanced technical degree was conferred by the same school in January, 1923. Since 1917 he has served the U. S. Government in the development of methods of preventing dust explosions in industrial plants. During the War Emergency Campaign he was in charge of the Eastern District and served in the same capacity during the dust-explosion prevention campaign of the U. S. Grain Corporation. He has served as a representative of the U. S. Department of Agriculture on various committees and organizations, such as the American Standards Association, National Fire Protection Association, and the National Fire Waste Council of the U. S. Chamber of Commerce. Mr. Brown developed the method of using flue gas for the prevention of dust explosions in feed-grinding systems and is the author of a number of articles on dust explosions, their characteristics, and methods of prevention.

Contributed by the Safety Committee and presented at the Annual Meeting, New York, N. Y., Dec. 1 to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

the most importance, however, particularly to the operator of a woodworking plant, is the fact that all of the wood dusts tested have been found to be quite explosive, which indicates very conclusively the necessity for the adoption of explosion-prevention measures. Before considering some of the preventive measures recommended it may be well to review briefly some of the explosions investigated in which certain outstanding facts indicate quite definitely the methods to be followed in reducing or eliminating the dust-explosion hazard in woodworking plants.

Explosions Investigated

A short time ago a serious dust explosion occurred in a woodworking plant in the Middle West where the shavings and dust



Fig. 1 Concrete and Steel Will Not Withstand the Pressures Produced in Dust Explosions

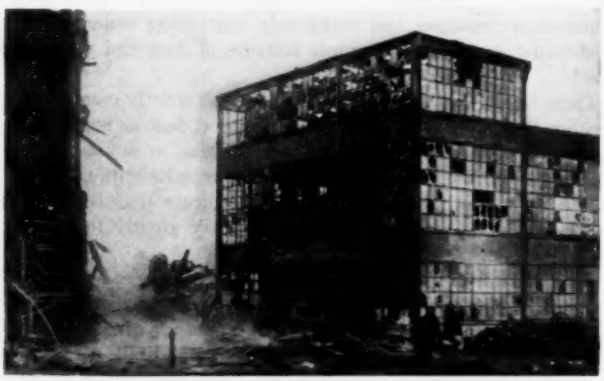


Fig. 2 A Building That Survived a Dust Explosion
(Large window areas and light wall and roof construction frequently serve as vents to release the pressure of a dust explosion and to prevent serious damage.)

from the finishing machines were deposited in a brick vault extending from the boiler room through the various floors to the roof of the plant. This vault 22 ft. by 37 ft. and 35 ft. high with 8-in. walls extended 6 ft. above the building and was covered with a flat wooden roof. Two metal cyclone dust collectors were supported by a steel framework directly over the vault and discharged the material they received from the machines through the wooden roof into the vault. Material was removed from the vault through three doors at the bottom located about 8 ft. from the furnace doors. Fire is believed to have entered the vault through one of these doors, although sparks or a smoldering fire may have been carried with the material blown into the vault from the plant. When the fire was discovered two men went to the roof of the plant and attempted to open a 2 ft. by 3 ft. door in one end of the vault. An explosion occurred instantly, and the men were buried under bricks and debris. Flames shot out of the vault openings in the boiler room and enveloped three employees there. The force of the explosion was sufficient to lift the vault roof and the heavy steel framework which supported the two cyclone collectors, and the entire mass dropped into the vault.

Previous explosions are believed to have been caused by backdrafts from the firebox of boilers, and there is also a case on record where a spark from an emery wheel near a woodfinishing machine ignited the dust, and the fire was drawn into the dust-collecting systems. In another case a bolt dropped into the cutters of a grinding machine and produced a shower of sparks which ignited the finely divided dust in the air.

One of the explosions reported occurred under peculiar circumstances. Oil and sawdust on a bearing caught fire and ignited sawdust on the floor below. In an effort to extinguish the blaze the assistant engineer dashed a pail of water on the floor with such force that some of the light dust was thrown into suspension and exploded.

A somewhat similar explosion but on a much larger scale occurred at the woodworking plant where the scrap, shavings, and

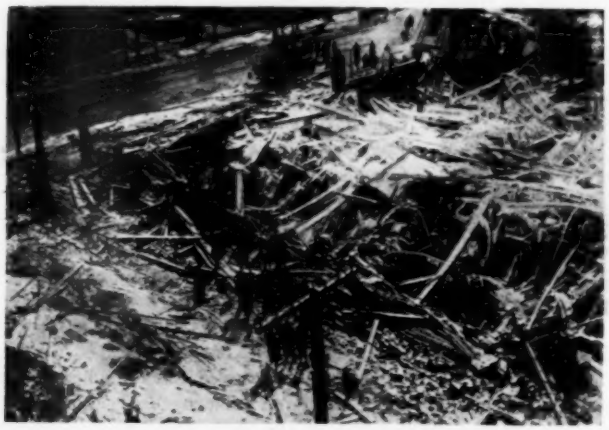


Fig. 3 Result of a Dust Explosion in a Woodworking Plant

sawdust were passed through a hog and blown to a storage tank outside of the boiler house from which it could be drawn as needed for boiler fuel. A fire at the hog flashed through the pipe line and ignited the material in the storage tank. Firemen attempted to draw off the contents and wet it down with a hose as it fell from the tank to the ground. The dust cloud formed in this way was ignited when the burning scrap fell from the tank, and the resulting explosion enveloped and seriously burned several of the firemen. Soon after this explosion firemen attempted to draw off the contents of interior bins following a fire in a Western woodworking plant, and six of them were killed when the dust cloud thus formed ignited and exploded.

Precautionary Measures

The information obtained in the laboratory tests of wood dust and during the inspection of woodworking plants indicates very clearly some of the precautionary measures which must be adopted in order to obtain protection against dust explosions. It is of course impossible to have a dust explosion without a sufficient quantity of dust in suspension in the air and a spark or flame to ignite the dust. As in all other dusty industries, therefore, cleanliness is of the utmost importance. If the plant is kept clean, there will be no dust to form a cloud through which an explosion can propagate, even if a minor

dust ignition should occur in or around one of the machines. Careful attention to the elimination of all possible sources of ignition, such as open flames, electrical apparatus likely to produce sparks or arcs, cutting or welding torches, and exposed element electrical-heating devices, will also serve to reduce the dust-explosion hazard. It has been found possible to produce a dust explosion by breaking in a dust cloud a lighted incandescent electric lamp. Evidently the filament remains heated after the glass is broken for a sufficiently long interval to ignite the dust particles with which it comes in contact. It is essential, therefore, to provide double globes and guards for all lamps of this kind used in a dusty atmosphere.

Steps should be taken to prevent the accumulation of static electricity on machines or equipment, because some dusts can be ignited by static sparks. Metallic sparks also are likely to cause trouble, and magnetic separators should be installed wherever necessary to prevent foreign material entering grinders or other machines.

Special care should be taken to prevent hot bearings or friction of belts and moving machinery which may cause a fire or dust ignition. Fire-resistive construction should be used wherever possible, and automatic or self-venting windows often prove valuable in releasing, before serious damage can be done, the pressure produced in a dust explosion.

Safety Codes

The precautionary measures which have been mentioned are the ones more generally applicable to the average plant. Complete safety codes for the prevention of dust explosions in grain elevators, flour and feed mills, sugar and cocoa plants, starch factories, and pulverized-fuel systems have been prepared by the National Fire Protection Association through its Dust Explosion Hazards Committee and have been approved by the American Standards Association. A special study of dust control in grain elevators has been made by the Chemical Engineering Division of the Bureau of Chemistry and Soils, and the results, together with certain suggested changes in the design and installation of dust-collecting systems, have been published in U. S. Department of Agriculture Bulletin 1373.

Regulations for the prevention of dust explosions in wood-flour manufacturing establishments have been tentatively adopted by the National Fire Protection Association, and attention is being given to safety requirements for other types of woodworking plants. It is suggested that owners or operators of woodworking plants obtain a copy of either the tentative code or one of the completed codes and consider carefully the safety requirements. Undoubtedly many of the recommendations in the completed codes can be followed or adapted to use in plants where wood dust is present and will serve to reduce considerably the dust explosion hazard. The Chemical Engineering Division of the Bureau of Chemistry and Soils in the U. S. Department of Agriculture will furnish on request any information on dust-explosion prevention which is available for distribution.

Discussion

H. G. Wiberg.[2] The data made available by the author draws attention to a hazard in woodworking plants not commonly thought of or known. The information should be utilized to prevent explosions similar to those cited from occurring in other plants. However it is thought that additional information which may in fact be already available, but which is not given in the paper, will be necessary.

The results of tests with respect to the pressure resulting from the explosion of different kinds of woods and at various concentrations are mentioned. It is wondered whether any data indicating the necessary concentration of wood dust to create an explosive mixture are available. Have any tests of that sort been made? Does this vary with the kind of wood and the fineness of the dust?

In discussing the explosions investigated, no mention is made of the products being manufactured. Were the plants sawmills, veneer plants, furniture mills, cabinet works, piano factories, planing mills, or sash and door plants? Fine dust is found only in certain types of woodworking plants, and we can therefore exclude a great many plants from this discussion.

For instance, the writer has been in a great number of planing mills, sawmills, sash and door plants, carpentry and cabinet shops, wood-turning plants, veneer product and cooperage shops where there were open flames, sparks of various kinds, and bearings and shaftings coated with sawdust and dripping with oil. They had been that way for long periods. He personally has not heard of a dust explosion in this type of plant.

Are not the explosive conditions confined to plants where work involving a large amount of fine sawing and sanding is found, such as furniture, radio-cabinet, piano, and panel factories? And perhaps the size of the plant has some bearing.

It seems that the efficiency of blower systems, where large quantities of fine wood dust are made in the process of manufacture, has a distinct bearing on the problem. If the suction is adequate and the heads closely surround the cutter or abrasive tool, there will be little free dust around the machines. This does not mean, of course, that the precautionary measures cited by the author should be disregarded, but that efficient dust removal will minimize, if not obviate, the hazard in the workroom.

The paper points to the conclusion that the great hazard is at the storage vault, tank, or cyclone. The writer does not note any specific recommendations for prevention at that source. Perhaps there are none. Has that phase been investigated?

A word with respect to the health hazard in woodworking plants. The writer cannot recall any claims for compensation because of work in an atmosphere which we might say is full of dust. There are few such places, anyway. As a rule, where there is a quantity of really fine wood dust coming from machines, it is kept down by dust removal systems. On the other hand, the writer has been in many places where the air was full of coarse wood particles and has not come across any cases of disease as a result. Apparently, the ordinary nasal dust-removal equipment takes care of that type and prevents injury.

The writer is reminded in this connection of a paper at a session of the woodworking section of the National Safety Council on infection from wood. The medical director of a large plant in Wisconsin, where there is a great deal of woodworking, made a positive statement that infection is not caused by any property of any kind of wood. Where infection follows puncture of the skin by splinters or any other form of wood, it is caused by some foreign material either being carried by the wood or present on the skin at the point of entry.

E. H. Schurch.[3] At the George W. Smith Woodworking Company's plant in Philadelphia, we have installed an Allington-Curtis screw-feed shaving stoker to handle hog refuse, shavings, and sawdust from the various machines. This consists of a small collector mounted above a storage bin on the roof of the boiler house. In the bottom of this storage bin are four screws which convey the fuel to two openings at one end of the bin. At this point, the fuel drops into the feeder screws, which are

[2] Lumber Mutual Casualty Insurance Company, New York, N. Y.

[3] Plant Engineer, Geo. W. Smith Woodworking Co., Philadelphia, Pa.

about 5 or 6 ft. in length, terminating in an almost vertical cylindrical chute which conveys the fuel into the furnace.

These screws are about 14 in. in diameter, and have a clearance of about 1/4 in. in the cylinder, which is supposed to prevent any fire from getting back into the storage bin. At the upper end of this vertical chute, there is an opening almost equal to the area of the chute, the idea of this being to admit a down draft into the furnace and prevent a flareback. At our request, this opening was closed with a loose hinged lid.

Since closing this opening we have had three fires, two of which were confined to the feed screw, but the other got up into the storage bin, which was hot enough to buckle some light iron sheets.

The question in mind is why, with all these screws agitating the fuel and no doubt keeping a lot of dust in suspension, was there not an explosion when the fire got into the bin. Mr. Brown's answer is that we did not get just the right condition or mixture to cause an explosion. In other words, we were just lucky that time.

This dust-explosion talk has thrown such a scare into us that we have installed steam jets in the openings from the bin to the feed screws and also in the bin itself. We wish to be just a little safer next time, in addition to being just lucky.

The writer also asked Mr. Brown if there was any danger from sparks in the case of a large collector which was dusting freely, located close to the railroad. His answer is that there was a possibility of an explosion, but he did not consider it very serious.

Author's Closure

Mr. Wiberg's question concerning the concentration of dust necessary to create an explosive mixture is one which several investigators have attempted to answer, and the results obtained in their experiments vary widely. The best information available indicates that to produce a wood-dust explosion it is necessary to have at least 10 oz. of dust per 1000 cu. ft. of air. This minimum concentration or lower limit cannot be taken as the absolute limit for such dusts wherever they are created because of the possible variation in size and composition under different manufacturing conditions, but it may be used as a guide in developing or providing dust-control equipment. One of the weak points in any work done with dusts lies in the inability to create and maintain a uniformly concentrated dust-air mixture. The heat capacity of the ignition source also has a decided effect on the limit.

In view of the laboratory tests referred to, it is believed that the lower limit of concentration for softwoods will be lower than that for hardwoods, but the figure given, 10 oz. per 1000 cu. ft., should serve for all practical purposes as the lower limit for wood dust.

Among the more than 70 wood-dust explosion reports in our files will be found listed nearly all types of woodworking plants, although it is generally specified that the dust producing the original ignition was fine. The greater number of explosions have occurred in sash and door plants, cabinet shops, wood-flour plants, or other establishments where fine dust is generally present, but sometimes ignitions occur in sawmills or planing mills when fine dust allowed to collect on overhead ledges is thrown into suspension and ignited.

Once a flame reaches the vault or shaving bin an explosion usually occurs unless the blowers have been shut down and the vault is comparatively free of suspended dust. Steam jets and water sprays are sometimes used to reduce the hazard at such places, but the method generally followed is to endeavor to provide protection rather than means of prevention. In bins properly vented with loose covers or swinging doors in the ratio of 1 sq. ft. for each 80 cu. ft. of volume, destructive pressures cannot be produced by an ordinary dust ignition.

The dust-ignition problem mentioned by Mr. Schurch is one generally encountered wherever an attempt is made to feed log refuse, shavings, and sawdust to the firebox of a boiler. Some success has been attained in preventing ignitions due to back drafts by installing steam jets or sprays. A system of chokes or dampers designed to prevent the passage of a flame will be found much more satisfactory, it is believed. Chokes may be produced in screw conveyors by removing a section of flights usually for a distance equal to about three times the diameter of the screw and substituting pins for the flights. This arrangement forms a seal in the conveyor box by allowing material to accumulate. The pins prevent packing, and the oncoming material forces that ahead of it through the choked section. Seals of this kind will prevent flashes or back drafts from the boiler reaching the storage bin, but care should be taken in shutting down to see that the material in the choke is not burning and the damper in the feed line to the boiler is closed.

Revolving dampers or rotary feeders are proving satisfactory in many instances as a means of preventing flames from the boiler reaching the storage vault. They consist of a paddle wheel or star wheel revolving horizontally, inside a close fitting cylinder with openings at the top and bottom. Material entering at the top falls between two of the paddles or blades and is carried around as the wheel revolves until it reaches the lower opening, through which it drops. The object with all such devices is to prevent the forming of a flue or direct opening from the boiler firebox to the vault.

WDI-53-10

Study in Present-Day Wood Finishing

By H. A. DERR,[1] MARIETTA, OHIO

A specific case of wood finishing is described in detail, a type of plywood floor covering that is completely finished by the manufacturer, the operations being carried out while the flooring is continuously conveyed. The finish (a water-resisting coating applied to the under side of the flooring and a filler, two coats of lacquer, and a coat of wax applied to the face of the flooring) is completed in approximately two hours.



THE general matter of the technology of wood finishes and their application has been ably discussed in the sessions of this division, notably by Silverstein[2,3] and by Kennedy[4] in papers that have covered in considerable detail the characteristics of various wood finishes, particular attention being given to types of materials and methods of application then in current use.

It is the purpose of this paper to present a description of a specific case of present-day wood finishing, illustrating what may be accomplished through the coordination of the woodworker and the wood finisher and utilizing materials and facilities available.

The case cited is a type of plywood flooring completely finished by the manufacturer, the operations being carried out while the flooring is continuously conveyed and with the finish (a water-resisting coating applied to the under side of the flooring and a filler, two coats of lacquer, and a coat of wax applied to the face of the flooring) completed in approximately two hours.

This product and its process of manufacture are considered unique, particularly in that the development is the result of intensive research, with the project carried from inception to large-scale production in less than one year. In this development, certain important fundamental conditions contributed largely, as follows:

(1) An unusual opportunity was presented for cooperation between woodworker, equipment manufacturers, and finishing-material supplier, complete coordination of all effort being effected by an engineer familiar with plywood manufacture.

(2) No existing organization, equipment, or traditional technique for finishing the product was required to be incorporated or accommodated.

(3) The development of the finish was put in the hands of one manufacturer of finishing materials, thereby providing the most favorable conditions for the proper interrelation of all finishing materials.

(4) The quality of the woodwork delivered to the fitnishin department is of the highest order, thereby minimizing he surgfacing operations during finishing. The excellence of the woodworking was favored by expert assistance in applying the principles of flooring manufacture to plywood construction.

(5) Adequate financial resources and other facilities were available to carry out the research and development.

From the standpoint of finishing, the problem presented was as follows:

(*A*) To determine the most suitable and practical type of finish for prefinished plywood flooring.

(*B*) To develop such finishing materials as would accomplish the desired type of finish in the shortest time and with minimum operations.

(*C*) To determine the most favorable methods of application and to adapt the finishing materials to such application, making proper adjustments to conform with conditions imposed by continuous movement of the flooring during finishing and in large-scale operations.

A—Type of Finish

The plywood to be finished was for the most part faced with sawn white oak in strips, with considerable strip flooring faced with rotary-cut white oak or sawn red oak, as well as parquetry of plain or quartered white oak and to a minor degree parquetry involving walnut, mahogany, and maple, singly or in combination, and also a small proportion of maple strip flooring.

Practically every type of finish available was given consideration. With the oak flooring as the product of primary importance, finishing materials most suitable for that wood were selected first. Naturally the matter of color required early attention. A light antique oak color was tentatively selected. Its general popularity being later confirmed by an expansive survey of representative architects, home owners, etc., it was decided to standardize on it for all regular work on oak. Color combinations for the other woods were later worked out, using modifications of the finishing materials selected for the oak.

It was then determined that the desired color effect could be produced on the oak by a combined stain and filler. The type of finishing coat, however, involved careful consideration and investigation. The flooring manufacturer naturally desired as highly finished a floor as was compatible with the practical requirements. The aim was to standardize upon one finish, of the type which would appeal to the majority of buyers. This condition tended to eliminate the thin, "hungry looking" type of treatment which purposely leaves little or no protective film of finishing material over the surface, however desirable this might be from an artistic standpoint, particularly where antique effects are desired.

It therefore became apparent that a full-bodied, smooth coating with subdued gloss was desirable. To produce such an effect became the next step. Various varnishes and lacquers and shel-

---

[1] Factory Manager and Technical Director, Marietta Paint and Color Company. Mr. Derr has been in his present position for eight years. Formerly he was for seven years with the Sherwin-Williams Company, Cleveland, Ohio, in the capacities of superintendent of technical service department, factory superintendent, and chemist. His extensive previous experience includes chemist with Kelloggs & Miller, linseed oil manufacturers, Amsterdam, N. Y.; and du Pont Company, Experimental Station, Wilmington, Del., and Carney's Point (N. J.) plant; and as instructor in chemistry at Washington and Lee University, Lexington, Va. He received the degree of B.S. in Chemistry from Washington and Lee University in 1910.

[2] S. M. Silverstein, Providence Meeting of the A.S.M.E., May 3 to 6, 1926.

[3] S. M. Silverstein, Annual Meeting of the A.S.M.E., Dec. 6 to 9, 1926.

[4] Paul S. Kennedy, National Meeting of the Wood Industries Division, A.S.M.E., Oct. 17 and 18, 1927.

Presented at the Sixth Annual Wood Industries Meeting, Winston-Salem, N. C., Oct. 15 and 16, 1931, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

lac were tested to determine what type and combination would develop the most durable and practical coatings.

The service to which a floor is subjected is of the most severe type, involving the tread of feet, scuffing from the movement of furniture, abrasion of hard particles carried in, exposure to water in small quantities, and occasional spillage of various liquids.

The floor finish must be hard, tough, and resistant to abrasion and water, but it should not be unyielding. Theoretically, the floor coating should be neither harder nor softer than the flooring, but should as nearly as possible become a part of the floor itself, so that when the floor is dented, the finish should not shatter or crack visibly nor should the finish leave the wood at the dented area, but rather it should follow the depression in the wood without loosening of the bond between the finish and wood.

It was determined by tests for wearing properties, exposure to the weather, and immersion in water that oak flooring coated with certain lacquers or varnishes offered the best general service, particularly if there also was applied a coat of wax. Sections of oak flooring finished in various ways were laid in passages or stairways. Stair treads offer a particularly rapid means of determining the relative wearing properties of flooring treatments. The earlier wearing tests were of this type. A device recently developed, best described as a twin-step testing machine, reproduces the average human step, successively applying pressure of 25 lb. per sq. in. through a leather pad which is turned 5 deg. at each step. Two samples can be compared side by side.

Although certain varnishes would make possible a high-quality finish at relatively low material cost, there was the considerable handicap of a comparatively long drying schedule, even with the fastest type of synthetic-resin varnishes. As it was planned to use a continuously moving conveyor for the flooring during finishing, it would be more economical to use a somewhat higher cost finishing material, particularly when lacquer offered other distinct advantages.

The type of finish finally selected resolved itself into the following: For the face of the oak flooring, a combination filler, two coats of lacquer, and one coat of wax; for the back or underneath side of the flooring, one coat of a water-resistant, rapid-drying synthetic-resin varnish carrying aluminum bronze, so as to develop proper leafing and coverage and giving a high degree of protection.

The decision as to the finishing materials for the face of the floor has been confirmed by an investigation, results of which were published in June, 1931, as circular No. 384 of the Scientific Section, Educational Bureau, American Paint and Varnish Manufacturers Association, entitled "Finishes for Oak Floors." The conclusions are in part as follows:

(1) It would appear that wax-like materials applied directly to oak flooring give entirely unsatisfactory service, in that they become dark very rapidly, allowing the dirt to become ground directly into the grain of the wood in such a way that it is very difficult to remove such dirt.

(2) It would appear that of all the finishing tests the ones which gave the best general results, resistance to grinding of dirt into the wood, the lightest in color and the best general appearance, were those in which the wood was first filled with silica wood filler and then given several coats of varnish, shellac, or lacquer.

(3) We believe that it is advisable that shellac, lacquer, or varnish finishes on floors be waxed before the floors are put into service and that wax coatings should be renewed frequently. The wax acts as a lubricant, protecting the finish from scratching. It is probable that a floor thus finished would remain in a presentable condition indefinitely if properly cared for.

B—Finishing Materials

With the general type of finish clearly defined, development work on the finishes took the form of producing materials which would, without impairing the high quality of the finish, make possible the minimum of time for the finishing. The several finishing materials adopted and the minimum time allowance for each coating are as follows: Combination filler, 30 minutes; first-coat lacquer, 20 minutes to permit sanding; second-coat lacquer, 20 minutes; wax, 6 minutes; aluminum backing, 10 minutes. The backing was applied prior to finishing operations on the face of the flooring, giving longer opportunity for drying. The actual drying schedule established in production is somewhat slower than the foregoing allowances. All of the drying operations are assisted by temperatures ranging up to 134 deg. fahr. (See Table 1.)

It is probably desirable at this point to give attention to each of the finishing materials involved.

Combination Filler. By this is meant a material which fills the pores of the wood and at the same time introduces the desired color into the finish. In many cases a considerable depth of color can be introduced into the wood, but in this instance the staining effect required is relatively slight. The development of a combination filler to meet the unusual requirements for speed of drying was difficult. In the building of a wood finish the foundation is important. This foundation in the finishing of oak, which presents large, open pores, is the filler. After it has been properly packed into position, it should harden without shrinking or swelling, bringing the surface of the wood to a uniform level and providing a smooth, non-porous surface that will permit of building up other coatings until the full finish is produced.

TABLE 1 FINISHING SCHEDULE FOR PREFINISHED PLYWOOD FLOORING (SEE FIG. 1)

(Conveyor Moving Continuously at Rate of 14¼ Ft. per Min.)

1	Loading on conveyor in woodworking department.
Aluminum backing (total time 11½ min.):	
2	Sprayed on back of flooring.
3	Force-dried at 130 deg. fahr. for 1½ min.
A	Air-dried at room temperature for 2½ min.
B	Force-dried at 134 deg. fahr. (relative humidity, 46 per cent) for 6 min.
4	Flooring turned back-side down.
A	Air-dried at room temperature for 1½ min.
Filler (44½ min.):	
5	Brushed on face of flooring.
6	Flatted by air-blast from fan (room temperature) for 2 min.
7	Padded-in by 26-in. Strand rubbing machine.
8	Rubbed-off by hand after ½ min.
A	Air-dried at room temperature for 1 min.
B	Force-dried at 134 deg. fahr. (relative humidity, 46 per cent) for 39½ min.
9	Cooled by air-blast from fan (room temperature) for 1½ min.
First-Coat Lacquer (34½ min.):	
10	Sprayed on.
A	Air-dried at room temperature for 5 min.
C	Force-dried at 104 deg. fahr. for 6 min.
D	Force-dried at room temperature in chamber providing rapid air circulation for 11½ min.
A	Air-dried at room temperature for 1½ min.
11	Sanded.
A	Air-dried at room temperature for 3 min.
D	Force-dried at room temperature for 6 min.
A	Air-dried at room temperature for 1½ min.
Second-Coat Lacquer (25½ min.):	
12	Sprayed on.
A	Air-dried at room temperature for 4½ min.
C	Force-dried at 104 deg. fahr. for 6 min.
D	Force-dried at room temperature for 12 min.
A	Air-dried at room temperature for 3 min.
Wax (8 min.):	
13	Padded-on.
14	Rubbed-off after 1 min.
15	Polished after 1 min.
A	Air-dried at room temperature for 6 min.
16	Removed from conveyor for inspection and packing.

Total time for complete finishing, 2 hr. 4 min.
Total time for complete cycle of conveyor, 2 hr. 12½ min.

By ordinary standards a good filler should be hard enough in from 24 to 48 hours to permit the application of lacquer finishing coats, assuming normal room temperature. For many years the 24-hour drying schedule had been in effect when varnish-type coatings were to be applied over the filler, and with the advent of lacquer finishing coats, it was for a while felt necessary to permit a drying period of 48 hours to offset the softening action

to be expected from lacquer coatings involving liquids of high solvent power.

It having been tentatively determined that a force-drying condition with a temperature ranging from 130 to 140 deg. fahr. would be feasible for the work proposed, experimental work which had been under way on this general type of filler for some three years was directed at this specific problem, and a filler was finally developed which could be coated over with lacquer materials within 30 minutes after completion of the filling operation when assisted by a force-drying condition of rapid air circulation at a temperature from 130 to 140 deg. fahr. and with a relative humidity of approximately 45 per cent. This same filler when air-dried at normal room temperature takes lacquer coatings properly within $1^1/_2$ hours. To provide a liberal factor of safety, the process finally set up provided for a force-drying period of $39^1/_2$ minutes, or a total of less than 45 minutes for the time required from the first operation in the application of the filler until the lacquer coat was applied over the filled work.

Prior to the development of this filler, it is believed that the most rapid-drying practical filler available was that used by a large radio-cabinet manufacturer, in whose process the wood filler is air-dried for $2^1/_2$ hours.

To make possible this drastic reduction in the time schedule no sacrifice of practical working properties or quality was involved, as indeed this filler has proved to be remarkably adaptable, particularly as to what may be done with it in the manipulation of the initial set-up immediately following the application of the filler.

The filler, supplied in paste form, is thinned with benzine in the proportion of 7 lb. of filler to 1 gal. of benzine. In this thin consistency it is brushed upon the surface of the wood. Evaporation of the volatile material immediately begins, and when this has developed to where the surface of the filled work "flats down," the filler is "padded in" by rubbing across the grain. In the process finally established only two minutes elapses between the application of the filler and the "padding in" process, during which time the work is under an air blast from a fan. It was found, however, that the same filler, if properly adjusted, could be passed through a dry room at 130 deg. fahr. for six minutes before "padding in" and still work properly.

In a wood filler there are four different types of materials usually involved. The "backbone" of the filler consists of inert pigment of transparent type; then there is the binder, which causes the hardening action; other materials are used to introduce the desired color, and volatile liquids are used for thinning the paste to proper working consistency.

While a filler is ordinarily rather simple in its chemical composition, the filler required in this case is quite complex. In the same four types of materials, 23 different materials are included; 8 come as raw materials to the filler manufacturer, while 15 are more or less highly processed before fit for use. The bulk of inert transparent pigment used is finely powdered silica, but two other inert pigments are used in combination with silica. The binder used in this filler differs radically from that usually employed in wood fillers. Ordinarily, raw linseed oil with a proportion of drier is utilized. However, this case, in order to make possible hard drying in the minimum time while retaining proper working qualities, required the development of a highly processed oil combination in which a certain balance of materials is carefully adjusted.

The coloring materials used are for the most part natural earth pigments, to avoid the tendency of the lacquer solvents toward "bleeding" of the color. The dyes also are of a special type.

The volatile portion of the filler comprises petroleum and coal-tar distillates balanced to give the desired speed of initial set-up and final hard drying.

The results obtained with this filler are an indication that research has accomplished what was predicted by Silverstein,[3] that "a filler will be developed with a drying time which will more nearly approach that of lacquer instead of the 48 hours now required."

Lacquer Facing Coater. Lacquer is not customarily used as a floor coating, primarily because of the difficulty in application by brushing and, secondarily, because the average lacquer is not suitable for floor treatment.

Prefinishing of the flooring overcomes the first difficulty, and the great diversity of materials which scientific research has recently made available to the lacquer manufacturer makes possible lacquers that are entirely practical for use on floors, giving great durability and introducing into the floor finish at the same time qualities which are peculiar to lacquer.

A lacquer for use on wood floors must provide for strong adhesion, and at the same time be sufficiently elastic or plastic to "stay with" the wood during volume changes due to varying humidity and also when the floor is dented in service.

Resistance of the flooring finish to weather exposure is also of importance, as during occasional exposure at open windows or doors. Practically any lacquer is highly resistant to the action of water on its finished surface, but only certain types of lacquers resist the action of moisture from beneath the film, i.e., moisture which finds its way into the grain of the wood under the finish, with resultant swelling of the wood and consequent strain upon the bond between the lacquer and the wood.

Such a condition, in an extreme degree, is represented by the immersion of wood coated on one face with lacquer, but with end grain or back of the wood, or both, unsealed. Therefore tests of this type with the finished flooring immersed in water for periods of 24 hours and longer were freely used in the development of the lacquer. Also, various types of weathering tests were utilized, in some of which the flooring under test was completely sealed with finishing coats, while in others only the face was sealed or finished.

In general, it was found that those lacquers that stood up the best in immersion tests also gave greatest durability under outside exposure to the weather. The weathering tests, however, are particularly important because of the action of the ultra-violet rays of sunlight.

The methods employed for testing the value of the lacquer with respect to wearing properties have already been described, and these wearing tests, immersion tests, and weathering tests constitute the most significant tests available, it being found that lacquers which satisfactorily answered these tests would, in general, meet other and special tests, such as exposure to sub-freezing temperatures, to soap solutions, brine, alcohol, gasoline, ink, and various commonly used medicinal preparations having staining or corrosive properties.

In the formulation of the lacquer it was necessary, also, to have the material properly adjusted for application to the filled work while the flooring retained a temperature of 125 to 130 deg. fahr., involving unusually slow set-up on the part of the lacquer in order to obtain a smoothly flowed finish.

Furthermore, the lacquer facing coater must permit sanding within a very few minutes after application of the first coat, and waxing in a similarly brief time after the application of the second coat, followed by packing in bundles, face to face and back to back, a little more than an hour after the application of the first coat of lacquer facing coater.

It was for a while considered necessary to use different lacquers for the first coat and second coats, but later, for several reasons of a practical nature having to do with the operation of the plant, it was decided that inasmuch as one lacquer could be developed which would meet all of the requirements for both first coat and

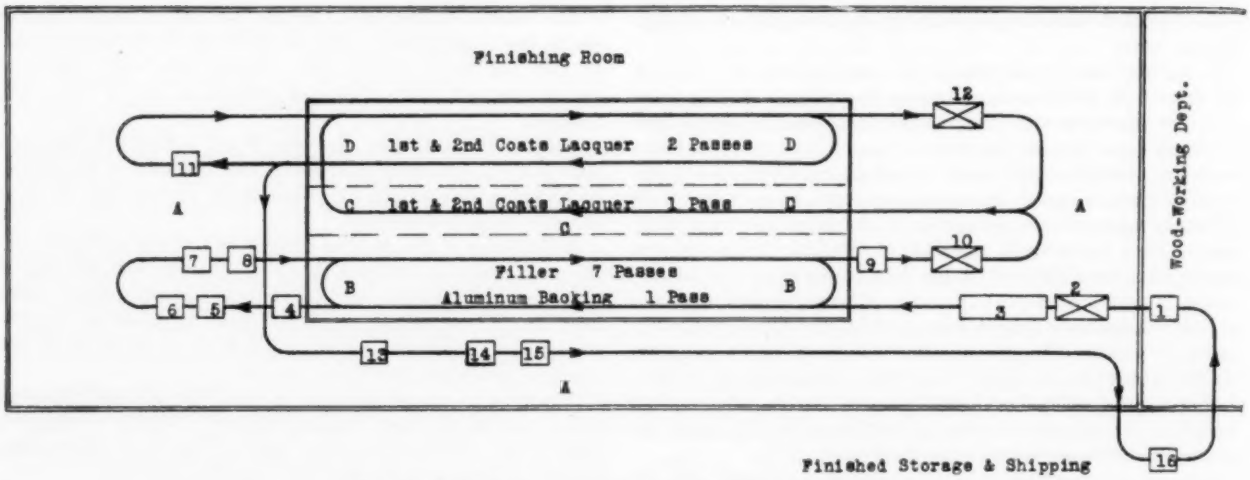


FIG. 1 CHART OF OPERATIONS ALONG CONVEYOR SYSTEM
(See Table 1 for key.)

second coat work, it was desirable to utilize only one lacquer.

The composition of a lacquer coating which complied with all these varied and more or less conflicting conditions is interesting from a technical standpoint, but it may be said that the development of such a lacquer has been made possible only through the utilization of a synthetic resin of a very unusual type in combination with a group of the most modern solvents, some of which have been available to the industry only during the past year or two. Nine different materials are involved in its make-up, all of which come as raw materials to the lacquer manufacturer, while each of these materials in itself represents the product of a more or less complex chemical process resulting from and controlled by applied organic chemistry of a high order.

Wax. The general popularity of a wax finish on hardwood floors is well-founded, because of the comparative ease with which this finish may be maintained or renewed, the upkeep of the floor being thereby simplified. At the same time, a beautiful, subdued sheen is developed upon the floor by repeated waxing, and this effect is reproduced on the prefinished flooring by the use of a coat of wax as the final finishing application. Another useful service which the wax coating performs is the protection of the finished faces of the flooring during storage and shipping, the wax acting as a lubricant, preventing slight marring as the result of the tight bundling of the strips in steel-strapped cartons and serving also to protect the face of the flooring during the laying operation.

The wax combination used, while polishing readily, is not slippery. The principal component of this wax is the time-tested beeswax, in combination with other waxes and suitable solvents.

Aluminum Backing. The use of an aluminum backing coat for the bottom side of the flooring tends toward an increase in cost over a clear or unpigmented backing coater, but the features of increased efficiency in moisture resistance and increased attractiveness of appearance justify its employment. Because of its lower cost a varnish-type vehicle, rather than lacquer-type, was selected.

The leafing action of aluminum bronze during the setting-up of the film makes possible the special efficiency in moisture-proofing attributed to aluminum coatings. This same characteristic tends to retard the drying of the vehicle. The development of a varnish-type vehicle for this purpose, which would comply with the extremely rapid drying schedule imposed and offer a very high degree of moisture resistance, represented a most interesting problem from the standpoint of its technology. The back or underneath side of the flooring is coated as the very first operation in finishing. Within ten minutes the aluminum coating must be set sufficiently hard to permit handling, as the flooring is at that stage of the process turned over on the conveyor. While the aluminum backing coater continues to dry and harden during the finishing of the face of the flooring, it is only two hours and four minutes from the time the aluminum backing coater is applied until it may be packed in bundles for shipment.

The vehicle portion of the aluminum backing coater is a striking example of what can be accomplished by the new technique in varnish making, and its composition is an illustration of the so-called synthetic varnish. Representing a synthetic resin of highly complex origin (phenolic-condensation product of the bakelite type) properly cooked with china-wood, or tung, oil, with the addition of suitable metallic driers and volatile thinners, there is presented a varnish which gives not only the desired speed of drying, toughness, and water resistance, but a full-bodied film meeting all of the practical requirements of a one-coat sealing and protecting finish on the under side of the flooring.

This entire finish represents a further interesting illustration of the interdependence involved in modern industrial civilization. The four different items of finishing materials supplied by the manufacturer of finishing materials to the woodworker in this instance represent a total of 48 different materials which, coming as raw materials into the processes of the maker of finishing materials, are in many cases themselves the highly processed products of the chemical industries, which in turn draw upon the natural resources of widely separated parts of the world.

This may also serve to illustrate the importance of the interrelation of one finishing material to another and the advisability of obtaining when possible all finishing materials from one source to insure the proper correlation of the various finishing materials; which point is of greatest importance when for any reason a change becomes necessary in any one of the coatings.

C—METHODS OF APPLICATION

Let it be kept in mind that the application of all finishing materials was required to be done while the flooring is in constant motion, and with the flooring temporarily assembled to form panels 24 inches wide (the width of four strips) and 80 inches long, the panels supported by standard carriers coupled to the conveyor chain. This condition naturally influenced the selection of methods of application.

The aluminum backing coater and both the first and second

coats of the lacquer facing coater are applied by spraying, hand operated.

The filler is applied by brushing, "padded in" by a machine consisting of a rotating disk 26 inches in diameter, equipped with a series of felt pads as rubbing members, as shown in Fig. 2, and is rubbed off by hand.

Owing to the high degree of perfection of surfacing and smooth-

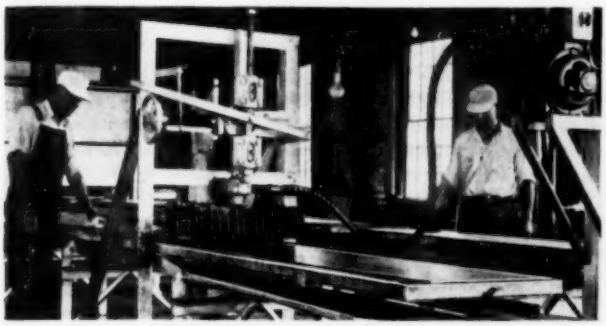


FIG. 2 APPLYING THE FILLER
(The man to the right is brushing on filler. The rotary machine in the center is ready to "pad in" the filler. The man to the left is rubbing off the filler.)

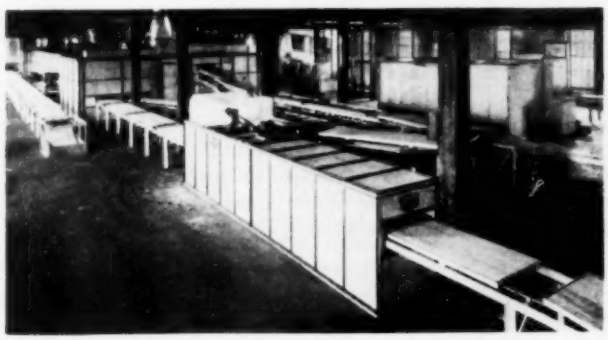


FIG. 3 UNFINISHED FLOORING COMING UP ON CONVEYOR FROM WOODWORKING DEPARTMENT
(Flooring is here bottom side up, and the aluminum spray station, not shown, is at the right, just before conveyor passes into back-coating drier, in the center.)



FIG. 4 OPPOSITE END OF DRIER FROM LACQUER STATIONS
(Showing where filler is applied and intermediate sanding and rubbing is done.)

ing operations on the flooring before its entry into the finishing room, only one slight hand-sanding operation, supplied by one man, is required during the finishing operations. (See Fig. 4.)

The wax is padded on, rubbed off, and polished at the present time by hand. These operations can, however, all readily be accomplished mechanically.

It was recognized that spraying application, particularly of the lacquer coats, involves the waste of considerable material in excess of that actually falling upon the work, as well as the use of a large proportion of volatile material which serves only to spread the solid content of the lacquer into place and is then lost by evaporation; and serious consideration was given to the possi-

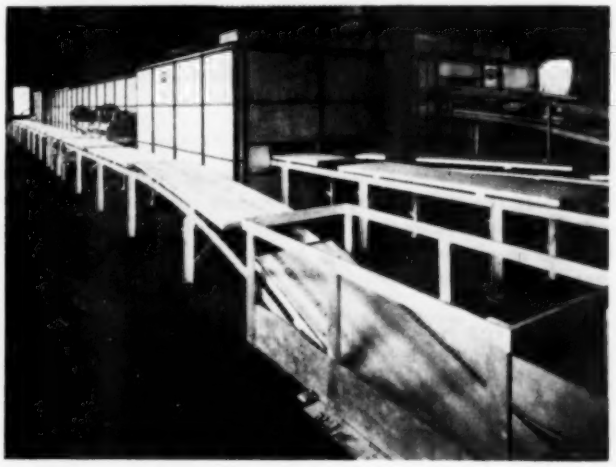


FIG. 5 FINISHED FLOORING ON CONVEYOR TO THE PACKAGING DEPARTMENT AND STOCK ROOM
(Waxing crew is located along the chain at left.)

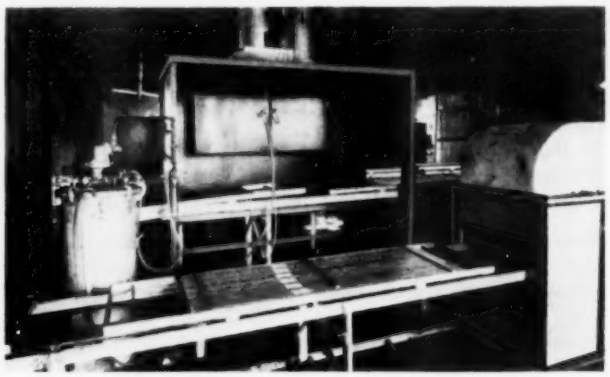


FIG. 6 SPRAY BOOTH FOR APPLYING FIRST LACQUER COATING
(Similar booths are used for second lacquer as well as for aluminum back coating. In the foreground, flooring is coming from back-coating drier, bottom side up.)



FIG. 7 CONVEYOR CHAIN INSIDE OF DRYING ROOM, WITH SIDES REMOVED
(The various decks wind around in a vertical spiral, giving an all-over travel exceeding 2000 lineal feet.)

bility of application by rolling, which process permits of the use of heavier bodied or higher non-volatile content material. Such application, however, was deemed not as practical as spraying for the purpose at hand.

In the final adjustment of finishing material to meet the actual conditions involved in the large-scale production process, surprisingly few and slight changes were required to be made in materials previously developed and tested in the laboratory. This is to be attributed largely to the full cooperation and complete exchange of information between the flooring manufacturer, the manufacturer of equipment, and the manufacturer of the wood-finishing materials.

Operation of Finishing System

Figs. 3, 4, and 5 show the general character of the equipment set-up. In the center of the large finishing room is built a drier system which includes three separate compartments, each with a separate air-circulating system and humidifier and automatic temperature control, heat being supplied by closed steam coils.

The conveyor is arranged to pass back and forth through the drier and extending a considerable distance beyond both ends of the drying chambers. All spraying operations are located at one end of the main room along the conveyor, while the filling and sanding operations are carried out at proper stations at the other end of the system. (See Fig. 1.)

One loop of the conveyor extends into an adjoining building in which the woodworking operations terminate, and here the conveyor is loaded with flooring ready to be finished. (At another point along this same loop the finished flooring is removed from the conveyor for inspection and packing.)

The flooring enters the finishing room back side up (see Fig. 3), and is temporarily assembled for finishing in panels. Moving continuously at the rate of $14^1/_4$ ft. per min., it is finished as described in detail in Table 1.

In this process the product being finished is not withdrawn at any time from the conveyor line, there being no provision made for the semi-processed product being accumulated in pools or side streams as in some other wood-finishing operations which make more or less use of chain production methods. As the result, there is no flooring in evidence in the finishing department except that which is actually in movement at all times, the amount of product being held at any time in semi-processed condition being governed simply by the capacity of the conveyor.

Conclusion

Distinct advances have been made in the art and science of wood finishing during very recent years.

The application of scientific research based on long experience in practical wood finishing and utilizing improved types of materials now available make possible the development of high-quality finishes on wood in a fraction of the time formerly required.

Complete cooperation between manufacturer of woodwork and manufacturer of wood-finishing materials, with proper coordination of knowledge and effort, has been demonstrated to bring a prompt solution to a difficult and unique problem in wood finishing.

Such coordinated effort, applied according to sound scientific principles and making use of modern mechanical equipment, has resulted in the development of schedules for the finishing of wood comparable in speed of operation and uniformity of product with those in effect for metal finishing.

Such finishing schedules do not necessarily involve any sacrifice whatever in the quality of the wood finish, as wood-finishing materials available today which can be adapted to such rapid schedules make possible finishes of higher quality than have heretofore been possible or economically practical.

"Science can, and will, help the woodworking industries, particularly in the application of cheaper and more durable materials. The extent to which science can be of assistance depends largely on the attitude and cooperation of the woodworker."[2]

WDI-53-11

Utilization of Wood Waste in the Chemical Industries

By BEN B. FOGLER,[1] CAMBRIDGE, MASS.

The principal uses, other than for fuel, for the wastes from the sawmill and woodworking industries are found in the following chemical industries: Destructive distillation, the manufacture of pulp, paper, and building boards, and those industries consuming large quantities of wood flour, principally linoleum, plastics, and explosives. The technical aspects of this waste utilization have been so satisfactorily worked out that a project of this sort may be evaluated almost entirely from the economic side. Competition from other raw materials, together with certain economic factors, requires that all of these operations be based on an ample, well-located, and relatively long-lived supply of waste wood. In many of these processes the waste wood must be separated either as to size or as to variety, although in some cases similar varieties may be processed together.



Bachrach

THIS paper is intended to present a comparative summary of the several outlets which the chemical industries afford for the salvage of waste from the woodworking industries. The discussion is limited to those processes in which the immediate conversion of the waste or its ultimate utilization fall within the so-called chemical industries. It is, however, true that, except for fuel and the limited market for the disposal of these wastes in the form in which they are produced, the chemical industries afford the only outlets which have been developed on a substantial scale for waste of this nature.

Consideration is similarly being restricted to those developments in which the technical aspects have been satisfactorily worked out, both as regards procedure and the details of commercial equipment, so that such economic factors as the supply of waste, the cost of conversion, and the possibility of profitably distributing the finished product are the determining factors in such a venture. In view of the importance of the economic aspects of this situation and the fact that these vary so widely from case to case, no attempt will be made to set up over-all cost data on any of these processes, although an effort will be made to discuss the relation of the more important factors in each process as a guide to the consideration of any individual case.

As determined by the limitations noted, the processes considered are those for the manufacture of the following materials:

Wood flour
Wood distillation products
Pulp and paper products
Building board.

From an economic viewpoint, there are a number of common elements in these conversions which, to avoid duplication, will be considered at this time and then referred to only briefly in the detailed discussion of each process. On account of existing competition all of these operations must be carried out on a comparatively large scale. It is essential, therefore, that a careful check be made of the supply of raw material and the cost of getting it to a central processing point. Consideration must also be given to the uniformity in the quality of the material as regards species of wood, as well as to the continuity of volume throughout the year.

Most of these processes involve a substantial investment in equipment, some of it having a comparatively long life; and the analysis of the supply of wood waste should establish its availability over a period to amortize properly the capital investment in the waste plant. This is particularly true in those operations where the finished product is in competition with an essentially similar product made from other raw materials or from new wood.

As regards the cost of conversion, the usual economic factors apply, although there is frequently an available supply of excess or low-cost power about such primary woodworking operations.

The problem of profitably distributing the finished product is one of the most important phases of such a venture. With the possible exception of building board made from the waste from sawmill or wood-dressing operations, the product of such a salvage project will be distributed through entirely different channels from those in which the basic product of the parent company is sold. The parent company entering upon the manufacture of such a by-product is therefore usually faced with the alternatives either of marketing its product through a sales agent or of setting up a separate sales organization for the distribution. The cost of such an organization, and also that of the supervisory personnel and other elements of fixed overhead in the by-product plant, are among the most important factors in determining the scale upon which an enterprise of this sort must be operated.

Another closely related phase of this general situation is the matter of choice as to whether the producer of the waste shall undertake its conversion to by-products or turn this work over to another group. The latter may be an organization subsidiary to, and jointly financed by, a group of several independent mills operating in a territory from which the waste may be economically collected and forwarded to a central point for processing.

Alternately, an entirely independent operator may purchase the mill waste as a raw material for his manufacturing operations. In this case the operator is in the position of setting up a secondary operation based on a primary one which he does

---

[1] Mechanical Engineer, Arthur D. Little, Inc. Mem. A.S.M.E. Mr. Fogler has had a long experience, having served three years in general mill engineering work; seven years in charge of production engineering, Ludlow Mfg. Associates, Ludlow, Mass.; two years in charge of mechanical research, Research Division, Chemical Warfare Service, Washington, D. C.; four years as mechanical engineer, Experimental Engineering Department, National Lamp Works, General Electric Company, Cleveland, Ohio; and five years with his present connection.

Presented at the Fifth National Wood Industries Meeting, New York, N. Y., Oct. 16 and 17, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

not control, and if he is wise, he will not confine himself to a single source of supply.

Careful consideration should be given to the location of the source of wood waste with reference to the markets in which the proposed salvage products are to be consumed. Neither wood waste nor the majority of products made directly from it in appreciable volume are of sufficient value per unit of weight to bear the cost of rail transportation to far-distant markets. This situation will effectively shut out many wood-waste salvage operations as long as there are adequate sources of similar waste more advantageously located with respect to markets in which the products of these operations are consumed.

Wood Flour

Wood flour is wood reduced to a fineness approximating that of cereal flours. It may be made of either hard or soft wood in any form, from the bolts of new wood from which some special flours are ground, to sawdust and shavings which serve as the raw material for the majority of the domestic flours. Much of the equipment in this industry follows cereal mill practice, both in the grinding and final sizing operations.

One of the major items of cost in the manufacture of wood flour is the power required in the grinding operation. An obvious economy in this direction is the use as raw material of wood waste in as finely subdivided form as is available in the requisite quantities. For this reason shavings and sawdust are preferred raw materials and are used in the manufacture of most of the flour produced in this country.

Although accurate detailed data are not available on the production and consumption of wood flour, a conservative estimate places the total quantity used last year at 85,000,000 lb., of which some 15,000,000 lb. are imports mainly from Germany and the Scandinavian countries, whose low production costs and cheap ocean transportation to our Atlantic seaboard permit their product to compete with domestic flour on a price basis in some of the important consuming centers in spite of a duty of 33½%.

Approximately half of the wood flour consumed in this country is used as a filler in the manufacture of linoleum. While ground cork was originally used for this purpose and is still preferred in some grades, such as battleship and the darker inlaid and jaspé patterns, on account of its superior resiliency, the lighter color of wood flour, permitting pigmentation through a wide range, makes it a more satisfactory filler for those patterns in which light colors and delicate shades are desired.

Explosives of the type of dynamite account for a third quarter of the total demand for wood flour. Here it is used, along with oxidizing agents, such as sodium and ammonium nitrate, to serve as an absorbent and to cut down the sensitivity of nitroglycerin and to produce an explosive which can safely be accepted for transportation by the common carriers. Kieselguhr, a diatomaceous earth, was originally used for this purpose, although it is entirely inert in its chemical contribution to the reaction at the time of detonation. It has been found, however, that cellulosic fillers of the nature of wood flour not only serve a similar purpose, but also furnish carbon and hydrogen to react with the excess oxygen in the balance of the mixture, thus producing a more complete explosive reaction. The users of explosives have learned to associate light colors with fresh high-quality dynamite, which, in turn, has created a demand for light-colored flour for this market, although there is no reason chemically why a darker material would not serve the purpose equally well.

The third major outlet for wood flour is in plastics, where it again serves as a filler or body. This market now consumes close to 10,000 tons a year of the highest grade flour that is made in large quantities. The resinoid plastics of the type of Bakelite and Durez often contain up to 60 per cent by weight of flour, which has proved the most satisfactory all-purpose filler for such plastics and is used in approximately 90 per cent of the molding powder made.

The balance of the demand is split up between several varieties of relatively minor outlets, among which might be mentioned the surfacing of wall paper, the manufacture of incense, and as a cleaning agent for the finer grades of furs.

Quoted prices on wood flour range from $23 per ton for the coarser linoleum grades to $50 per ton for the finer grades used in resinoid plastics. Some special grades sell as high as $80 per ton. The possibility of converting wood waste, which in many instances is of value only as fuel to wood flour salable at the prices noted, is one which has been extremely attractive to many lumbering and woodworking interests. It is significant, however, that the ratio of failures to successes in the manufacture of this commodity has been extremely high.

There are several reasons for this situation. The first is the rigid specifications which have been laid down by the consuming industries covering the fineness of grinding, color, and the resin content of wood flour. Many of these specifications are technically sound; others have been drawn more or less arbitrarily, to insure a continuation of a material of a grade which has been found satisfactory to the consumer without regard to whether or not flour of some other specifications might serve his purpose equally well or even better. The result of these specifications, however, has been to restrict the raw material from which flour acceptable to the trade can be made to a relatively few species of wood which, in the main, must be kept entirely separate for each particular grade of flour. Fully 75 per cent of the flour made in this country is ground from white pine. Poplar, fir, maple, and birch are used for special grades.

In the majority of the products containing wood flour as a filler, the binder and other ingredients cost so much more per pound than the wood flour that the incentive to establish more competitive sources of flour supply is correspondingly lessened. At the present time the large producers and consumers of wood flour are both relatively few, and apparently are working in reasonable cooperation. This, with the balancing effect of foreign competition, creates a market situation into which it is rather difficult for a new producer to break.

In selecting a location for the manufacture of wood flour, consideration must be given to the costs of raw material, power, and transportation to market. While shavings or sawdust are usually cheap at the planing mill, the cost of freighting to a central flour plant, even within a few miles, results in appreciably more expensive raw material than when the flour plant receives its entire supply by conveyors from the planing mills.

The major factor affecting the demand for wood flour is the rapid growth of the plastics industry. This use calls for a high-grade, light-colored flour, finely and uniformly ground and conforming to close specifications as to resin content.

An independent wood-flour operation should not be set up on a scale of less than 10 to 15 tons of finished product. A plant of 5000 tons annual capacity involves capital charges approximating $50,000, and the nature of the industry requires a substantial amount of working capital.

Destructive Distillation

Technically the recovery of values from wood waste by destructive distillation is one of the most fascinating of the chemical salvage industries. No one who has seen the plant of this type operated by the Ford Motor Company at Iron Mountain, Mich., or the unusually complete educational model of this installa-

tion which has been exhibited so widely can fail to be impressed with the thoroughness with which these conversion processes have been worked out and the step up in value between the waste from the sawmill and wood-body plant and the extensive line of wood-distillation products and their derivatives produced in the Ford plant.

The destructive distillation of wood which today is practiced almost exclusively on hardwood consists essentially of two steps, carbonization and distillation. In the first operation wood is heated in closed retorts from which charcoal and crude pyroligneous acid are obtained, together with non-condensible gases, the latter of value only as a source of heat in the process. The charcoal from the retorts is cooled, graded for size, and except where the high proportion of fines necessitates a briquetting operation, this portion of the product is ready for the market. In the still house the crude pyroligneous acid is initially separated into acetic acid, crude methanol, and wood tar. The refining and conversion of these intermediate products varies with the individual plant, and any adequate discussion of them is beyond the scope of this paper.

The economic soundness of any wood-distillation project depends upon simultaneous satisfactory markets for its three major products:

Methanol
Acetic acid or calcium acetate
Charcoal.

While there are firmly established and increasing markets for both methanol and acetic acid, this situation itself has contributed largely to the development of synthetic processes for the manufacture of both products. Several of these processes are now in commercial production on a large scale, and their output is seriously affecting the price of these materials. The production of synthetic methanol, starting in this country in 1927, has so increased that in 1930 it will represent over one-third the domestic production, while in the same period the price has fallen off almost 50 per cent.

The increased demand for acetic acid and its derivatives caused by the rapid growth of the lacquer and rayon industries has been sufficient to absorb the synthetic production of these materials at satisfactory prices, until early this summer, when quotations on both acetic acid and acetate of lime broke sharply, following the announcement of the new plant of the Commercial Solvents Corporation for producing acetic acid by the fermentation of corncobs. The liquor used to feed the bacteria effecting this fermentation is a by-product of other large-scale operations of this company. The reported capacity of the initial unit of this installation is 5000 tons of acid per year.

With the falling off in the market for charcoal for metallurgical use, the finding of satisfactory outlets for this material is rapidly becoming a problem of specialty selling, on a commodity that will not bear heavy transportation expense.

This general market situation has kept the wood-distillation industry in a very disturbed condition for a period of several years. Improved processes and equipment and favorable outlets for their production, frequently in part within affiliated industries, have kept some of the larger plants operating at an apparent profit. The most important factor in maintaining the wood-distillation industry in a position to operate in competition with the synthetic products just noted is the development of processes for the recovery of acetic acid of required commercial grades directly from the still liquor without the intermediate production of calcium acetate.

The majority of the wood-distillation plants are operated mainly on cordwood cut for this purpose plus any sawmill edgings that are available within an economic shipping range. Where this process is operated largely on the waste from woodworking plants, such as furniture and body plants, special retorts must be used such as the Badger-Stafford continuous retort for block waste and the Seamen rotary retort for sawdust and shavings. Both equipment and operating costs for plants with retorts of these types are higher than in the case of plants operating on cordwood to the extent of wiping out a large part of the advantage in the cost of raw materials accruing from the use of waste wood. A wood-distillation plant involves an investment of $5000 to $7500 per ton of bone-dry waste treated per day, depending on the type of waste and the extent to which the intermediate products are refined. Under present conditions a plant of 75 to 100 tons input capacity per day is the smallest that should be considered.

Pulp and Paper Products

The utilization of waste coniferous woods in the manufacture of pulp and paper products has made rapid strides within the past five years. This period has brought into production approximately 12 mills, producing in the aggregate some 1500 tons of pulp daily from raw materials consisting largely of the waste from the lumbering operations on the Pacific Coast. This represents approximately 10 per cent of the domestic consumption of pulp, and the availability of this material at highly competitive prices along the Atlantic seaboard has caused radical changes in the manufacturing practice of many Eastern mills. The Western mills are making sulphite pulp from Western hemlock, while Douglas fir and Western pine waste are being converted by the kraft process. Similar kraft conversions based on waste from Southern pine sawmills have been in operation in the South for a longer period.

The conversion of waste soft woods of almost any variety to standard pulp and paper products is purely a problem in economics for which the following brief specification may be drawn:

"An adequate supply of waste wood of varieties sufficiently similar to cook together uniformly to support a plant of economic size over its normal life, so located that the combined manufacturing and distributing costs will permit the product to be delivered to the consumer at a price equal to and preferably below that of the competing product."

Under the present highly competitive conditions in the industry the minimum unit operation in either sulphite or sulphate pulp mills should have a capacity of 100 tons per day consuming the equivalent of, roughly, 200 cords of wood.

The commercial status of the manufacture of paper products from hardwood waste of the varieties which are milled in large quantities is less clearly defined. The difficulty of using these woods arises from the comparatively short length of the ultimate fibers, when these are separated by chemical cooking and fiberizing operations, which produces a pulp of low strength in comparison with that obtained from the coniferous woods.

Under the demand created by the recent and continuing rapid growth in the use of fiber shipping containers made from both corrugated and solid stock, so-called "semi-chemical" cooking processes have been developed for hardwood which hold a real promise of a more profitable utilization of waste trimmings from these varieties. The best known of these processes is the Forest Products semi-chemical cook.

So far the only strictly commercial development of this process has been the manufacture of paper for container stock from the spent chestnut chips left from the production of tannin extract. Several plants of this type are in commercial operation, and the development has brought about a marked change in the economic structure of the chestnut extract business.

Processes of this type, of which there are others under develop-

ment, can be successfully applied to the conversion of mixed hardwood waste to container stock wherever an adequate supply of such waste is advantageously located as regards manufacturing costs and proximity to markets. Yields by these cooking processes are substantially higher than from the conventional chemical cooks. There are a number of features which indicate the possibility of profitably converting hardwood in units of approximately half the size of those which can be economically operated by either the sulphate or sulphite processes. Such a hardwood conversion unit should have a minimum capacity of 50 tons of product per day and requires approximately 50 cords of wood.

Building Board

Another important outlet for wood waste is in the manufacture of building board, the use of which has increased in a spectacular way during the past few years. This material is now produced in a variety of forms ranging from the insulating boards of the types of Celotex, Masonite, and Insulite to the dense, almost synthetic wood, of which Prestwood is the outstanding example. New wood as well as such wastes as sulphite screenings and sugarcane bagasse now furnish a considerable proportion of the raw material for the present boards.

While the manufacturing processes for products of this type are somewhat restricted by active patents, they offer with the proper combination of industrial interests an exellent opportunity for the utilization of wood waste of practically all the varieties worked on a large scale. Such wastes, in fact, are now being utilized in a substantial amount of the present production by cooking processes similar to those already outlined.

At present prices this field offers an unusually attractive margin above the cost of production; and manufacturing facilities have had this stimulus for expansion over and beyond that created by a rapidly growing market. This situation, together with the wide range and large quantities of available waste or low cost material which can be converted to a satisfactory building board, is bringing about an expansion of the industry which is likely to result in overproduction with a corresponding break in prices and curtailing of profit.

A new process still in the development stage offers the possibility, with relatively minor changes in equipment, of changing a considerable portion of the waste of wood-dressing plants from shavings to a fibrous material which in turn can be converted by developed processes to a rigid insulating board.

Building board as well as many of the container products which can be made from waste wood are in most instances competitive with products made from wood. Regardless of its relative merits as compared with the earlier products made from virgin wood, these newer reworked or synthetic materials are now firmly established. An appreciation of this situation is already evidenced by several of the far-sighted interests in the lumber and woodworking industries which are arranging their manufacturing setup with such salvage operations in view.

Conclusions

Commercial processes are available for the conversion of the majority of the varieties of wood waste from either sawmill or woodworking industries to products for which there is an established market. Many of these processes require the separation of the waste wood by varieties, and sawdust and shavings require separate treatment and are usually used in the manufacture of other products than those which can be made advantageously from slab edging and trimmings.

On the majority of these waste-wood products existing competition is of such a nature that new enterprises must operate on a substantial scale and frequently involve an investment in the by-product plant approximating that in the primary operation.

Distribution costs of the finished by-product are a very important factor in the utilization of waste wood and may frequently bar out a development offering both a satisfactory supply of waste wood and low conversion cost.

Supplies of waste wood from a small operation, unless it is one of several producing the same type of waste within a small area, cannot as a rule be salvaged by processes of the type discussed. An increasing number of special developments are making profitable use of these smaller quantities of waste, although in most instances local conditions and merchandizing skill have been primary factors in the success of these operations.

Although the economic position of the major wood-waste salvage operations may be definitely established with relation to synthetic or other competitive processes on the basis of existing technology, this balance is continually being upset by the results of research work within these industries. This element is of importance as indicating for any enterprise of this sort the need for a very thorough analysis of each individual case at the outset of the undertaking and the provision for a qualified technical staff as an integral part of the business. For the same reason a conservative policy should be followed in setting up the depreciation periods for plant and equipment.

Discussion

Leon V. Quigley.[2] One should find good incentive in the numerous uses to which even waste wood is adapted. Mr. Fogler has epitomized nicely the economic factors involved in the establishment of a plant for wood-waste salvage or conversion. His indication of difference and diversity of marketing problems involved in the sale of primary products and by-products is brief but significant.

To the molder of plastics, particularly those of the Bakelite phenol-resinoid type, the term wood flour implies a filling substance. The term filler is, however, unfortunate, inasmuch as wood flour is not added to phenol resinoid to contribute to the weight or bulk of finished product. Neither does it represent an attempt to conserve on the amount of resinoid binder used. In phenol-resinoid material wood flour has two principal functions: (1) It contributes to the facility of molding—modifies favorably the flow properties, etc.; (2) it improves the property of shock resistance, yielding a molded part mechanically strong and tough, instead of one relatively fragile, which would result if the resinoid were used without reenforcement. Truly, wood flour had much better be called a reenforcing material than a filler.

Physically, the wood flour acts as an absorbent for the resinoid. While the analogy is imperfect, one might liken the function to that of a blotter absorbing ink. Wood flour serves as a carrying agent—a mass of fibrous material in which the initially liquid resinoid is dispersed.

The choice of wood is interesting, since all woods are not suited to provide commercial wood flour. In view of the remarks of Mr. Fogler, it will be realized that wood flour is a much more refined product than sawdust. It is unfortunate that many contributors to the literature have confused the two terms. Wood flour is pulverulent in a somewhat strict sense of the word, rather than granular or flocculent like sawdust. From the viewpoint of the plastics industry, it will be appreciated that a more nearly homogeneous molded product results from the use of wood flour than would be possible using ordinary sawdust. Preference rests with such woods as pine and spruce. Choice of wood depends directly on composition, chemical and

[2] Leon V. Quigley, Technical Editor, Bakelite Corporation, New York, N. Y.

physical. From the physical standpoint, hard woods are rejected in favor of soft because lower in absorptive power.

Texture, or the degree of fineness, is a property of especial importance. The finished molded unit must frequently be of good appearance, and to this end it is necessary that the wood flour fibers be entirely covered by penetration, or film, of resinoid, plus pigment or dyes which it embodies. An appreciable problem is to so adjust the fineness of the fiber that the flour incorporates itself unnoticeably in the molded piece, retaining nevertheless a microscopically fibrous texture, which will contribute to the molded product a degree of strength and toughness, which is more characteristic of fibers than of powders.

Author's Closure

One of the difficulties in preparing this paper has been the limited space which the author was forced to allot to each of the major utilizations of waste wood, any one of which might profitably be made the subject of a paper of the same length. Mr. Quigley's discussion has provided a valuable and much appreciated amplification of that part devoted to wood flour.

The term "filler," while thoroughly understood within the several plastics industries, is, as Mr. Quigley points out, a somewhat unfortunate designation for a very important element in the composition of a wide range of industrial plastics, as well as in the specific field of phenolic resins. While it is true that some of the materials classed as "fillers" are used primarily as diluents to reduce the cost of the plastic compound as a whole, in the majority of instances the proper selection of fillers, both as to material and physical specifications, enables the plastic technologist to tremendously extend the range of physical properties and appearance of the product available with a single resinous binder.

In the wood-flour industry the results of the careful work of research groups in the phenolic resin field, represented by Mr. Quigley, are found in the rigid specifications already noted for the molding grades of flour.

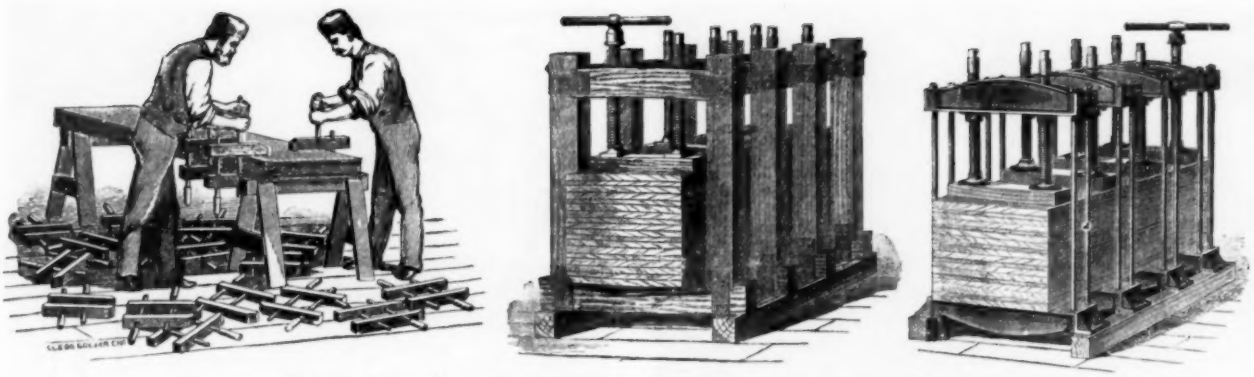


FIG. 1 HAND PRESSES, THE ANTECEDENTS OF THE HYDRAULIC PRESS
(Left, the age-old hand screw, applied to plywood; center, wooden-frame plywood presses, extensively used 30 years ago; right, iron or steel sectional presses, still used for hand work.)

Hydraulic Presses for Plywood

BY HARRY G. FRANCIS,[1] RUSHVILLE, IND.

The paper reviews the progress made in the clamping of plywood, especially in the last 35 years, when wooden hand screws and wooden-frame hand-screw presses were in general use. Improved hand clamps, improved hand-screw presses, and power-screw presses have their uses for certain classes of work and comparatively small output, and inverted-cylinder hydraulic presses are necessary for special methods and classes of work. Plywood is an artistic and utilitarian product with increasing appeal for both manufacturing and construction purposes. As the market for plywood products expands, new problems will necessitate further advances in press design. Plywood presses of the future may outdistance those of the present as much as the presses of the present demonstrate improvement over the old hand presses.

HISTORY relates that the earliest plywood press consisted of bags of hot sand, which both held the veneer flat against the core and accelerated the glue drying, and there is indisputable evidence of this process being used around 1500 B.C. Various other types of presses—lever, toggle-joint, and screw—came successively into use, but it was not until within the last 50 years that any standardized mechanical designs appeared. A few of the older designs are shown in the woodcuts reproduced in Fig. 1.

Though the hydraulic principle is one of the oldest and the most efficient methods of producing high pressure, its advantages in plywood gluing are not fully appreciated. This may be because so many inefficient hydraulic-press installations are still in service that plywood manufacturers are inclined to judge all hydraulic presses by these antiquated predecessors of the present-day high-production units.

HYDRAULIC PRESSURE IS MEASURABLE

Where pressures are to be closely regulated or a predetermined amount of pressure is to be applied, hydraulics must be used, being the only type of pressure that can be accurately measured and controlled. Until recent years, however, hydraulic presses as regularly furnished for the plywood factories were comparatively slow, although faster than hand-screw or power-screw presses. The pipe connections were not carefully made, methods of piston packing were not developed to the present-day efficiency, and the controls were not in the most convenient position.

MOTORIZED SCREW PRESSES

These apparent difficulties in the early types of hydraulic presses or the lack of intelligent operating methods led to further efforts to develop efficient mechanical presses. The most successful of these for plywood production is the screw type, as it can be made to operate at a proportionately faster speed, and when run by an electric motor, the pressure can be approximately gaged by an ammeter. This type of press is giving satisfaction where users do not require high or accurate pressures and the work is not of large area. When high pressures are required, however, the friction losses are excessive. A greater percentage of the power input is lost in friction, and as the friction losses are not in a constant proportion to the pressure exerted by the press, there is no way of knowing what part of the power input is being consumed by friction, and therefore no way of knowing what pressure is being applied to the work.

IMPROVEMENTS IN HYDRAULIC-PRESS DESIGN

In the meanwhile, development in the hydraulic field has been rapid, and the former objections to hydraulic presses have been eliminated. Methods of piston packing have been improved, pumping systems have been developed whereby any speed of operation can be had free from pulsation, and the controls are so simplified and protected that a child could operate them. The hydraulic press is built more compact than before; in fact, in many cases it is a self-contained unit requiring no more floor space than the press frame alone. A hydraulic press can be had to meet the most exacting requirements. There are a great many different styles and combinations, vertical and horizontal.

MODERN HYDRAULIC PRESS TYPES

The most popular of the modern hydraulic-press arrangements will be described. As different manufacturers use different trade

---

[1] President and Engineer, Charles E. Francis Company.

Presented at the Sixth Annual Wood-Industries Meeting, Winston-Salem, N. C., Oct. 15 and 16, 1931, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

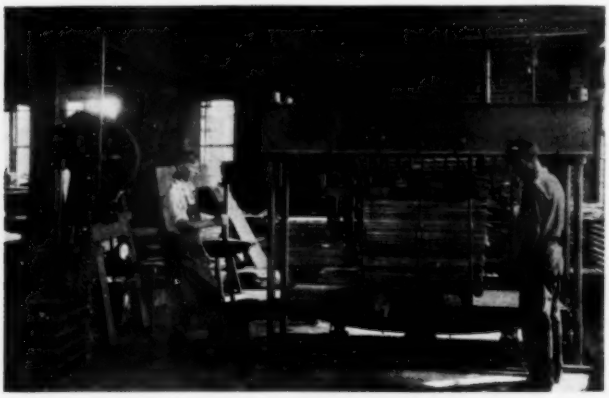


FIG. 2 HYDRAULIC PRESS WITH RECIPROCATING PUMP
(Pump shown at left, with cylinder below lower platen.)

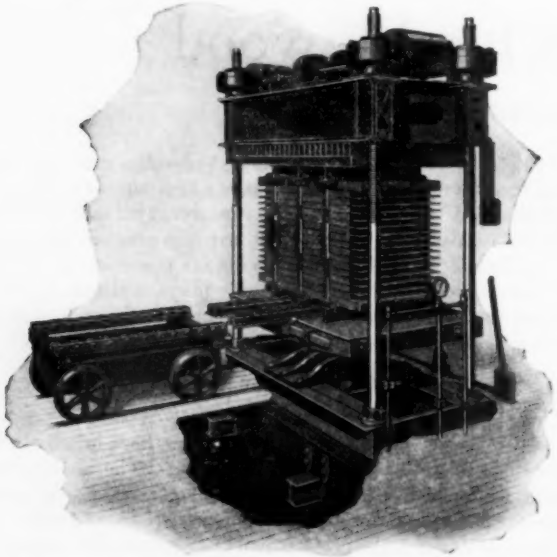


FIG. 3 HYDRAULIC PRESS WITH RECIPROCATING PUMP
(Hydraulic cylinder below, operating lower platen, and motorized geared nuts above, operating upper platen for quick opening and closing.)

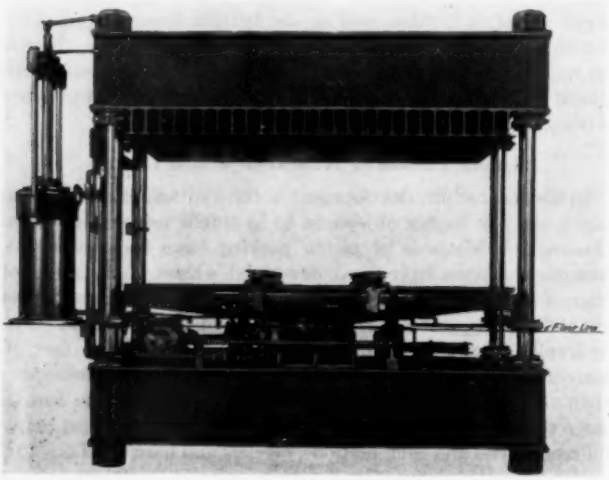


FIG. 5 HYDRAULIC PRESS WITH ROTARY PUMP AND INTENSIFIER
(Rotary pump mounted under lower platen, with intensifier for building up ultimate high pressures.)

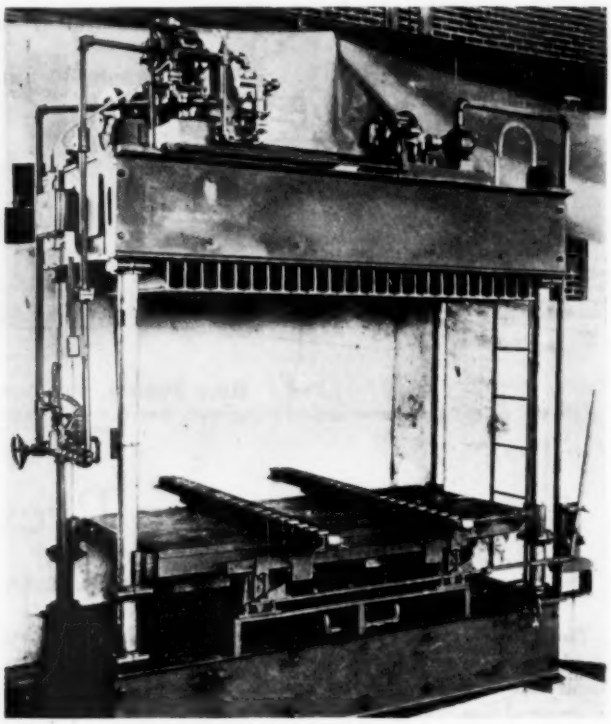


FIG. 4 HYDRAULIC PRESS WITH RECIPROCATING AND ROTARY PUMPS
(Rotary auxiliary pump mounted above for quick closing of platen, supplemented by reciprocating pump for ultimate high pressures.)

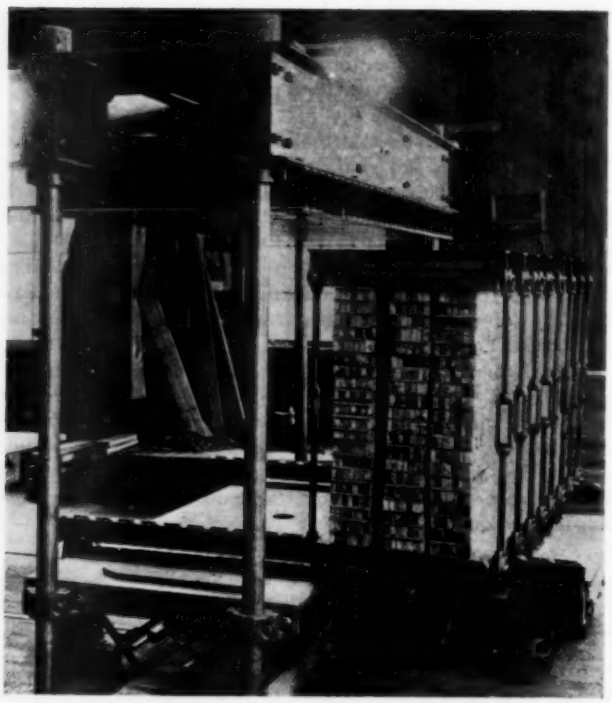


FIG. 6 CLAMPED BALE OR PACKAGE EMERGING FROM PRESS
(Ultimate pressure obtained in press is preserved by I-beams and turnbuckle rods until the glue takes its initial set.)

names and model numbers, the types will be described in details of construction and compared by key letters.

(*A*) *Press With Reciprocating Pump.* This is the most common form of hydraulic press (see Fig. 2) and has been built in this way for quite a number of years. This type is suitable where large production is not a problem and where large-dimension work is done to a limited extent and occasionally left in the press until the glue sets. It can be equipped with contact-making pressure gage, which can be set to start the pump motor when the pressure begins to drop and to stop it when the predetermined pressure is built up. This is an advantage over the older method, whereby the pump was left running continually against a relief valve, both in the saving in power consumption and in the wear on the pump.

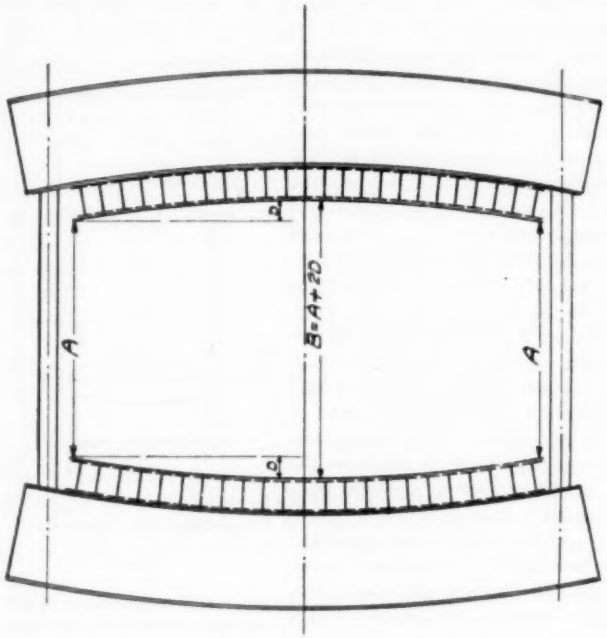


FIG. 7 STRAIN-ROD PRESSURES, WRONGLY APPLIED
(Side view, outside strain rods, platens supported at corners.)

(*B*) *Press With Reciprocating Pump and Motorized Rapid Traverse or Quick-Closing Head.* This type of press is similar to and has all of the advantages of the first-mentioned type, with the additional advantage of speed. The nuts on the top of the strain rods are in the form of gears, from which the top member of the press is suspended, as shown in Fig. 3. The internally threaded gears are motor-driven in unison through a connecting train of gears, making the top member quickly adjustable and enabling the operator to close the press on the work very rapidly, using the pump only for building up the final pressure.

This type of press combines all of the advantages of the power-screw type of press (not shown), with the superior pressure-applying qualities of the hydraulic press. It is suitable for large production, and usually has a closing speed of about 36 in. per min. When the plywood is laid up and the clamps are tightened, the pressure may be released through the motorized gear chain and the package removed immediately without waiting for the lower platen to return, as this platen (hydraulically operated) has not traveled very far in applying the final pressure.

This type of press is usually equipped so that the operation is semi-automatic; that is, when the plywood is placed in the press, the operator needs only to close the valve and press the "down" button of a three-button push-button station. The top member of the press will quickly run down until it comes in contact with the plywood. The pump will then automatically start to build up pressure, and the top will automatically stop. When ready to release the bale of plywood, the valve is opened and the "up" button is pressed. The pressure is thus released, and the top returns to normal open position and stops automatically. The normal open position is adjustable. Due to the short travel of the lower platen, no time is wasted in waiting for its return to unload, but there is still enough lower-platen travel to provide a platen-operated unloading device, the advantages of which will be described farther on. A greater variation in heights of packages can be handled without the use of fillers or without the use of extra-long hydraulic cylinder and ram, as the "daylight" (clear opening between platens) can be made to adjust from maximum to zero, and the adjustment for low packages can be made very rapidly.

Both the initial cost and the operating cost of this press are somewhat higher than with other types of production presses using hydraulic transmission throughout.

(*C*) *Press With Reciprocating High-Pressure Pump and Rotary Auxiliary Pump.* This combination is shown in Fig. 4 and embodies all of the advantages of type A, with additional improvements. It is also an efficient production press, as a rotary auxiliary pump can be had

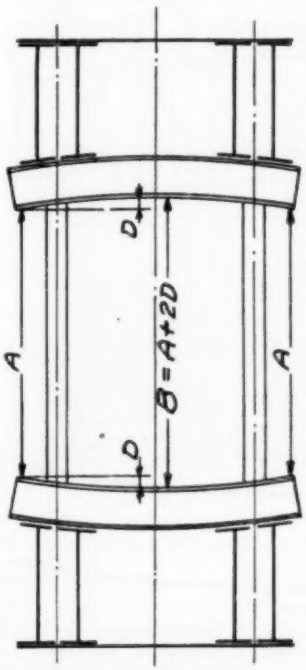


FIG. 8 STRAIN-ROD PRESSURES, WRONGLY APPLIED
(End view, outside strain rods, platens supported at corners.)

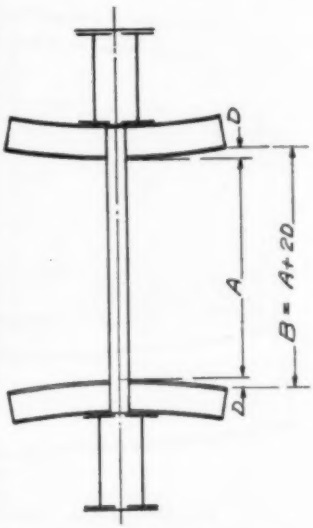


FIG. 9 STRAIN-ROD PRESSURES, WRONGLY APPLIED
(End view, center strain rods, with unsupported platen corners.)

that will effectively close the press at any desired speed. This press is also usually arranged with the semi-automatic control features, so that when the "up" button is pressed, the rotary-pump motor closes the press and takes up most of the slack in the package, building up initial pressure, which automatically, by means of a pressure-operated switch, causes the reciprocating-pump motor to start and the rotary-pump motor to stop. This arrangement makes this press with its two pumps as simple to operate as a press with a single pump. The number and kind of manual controls are the same.

It is not as expensive as type B, either in initial cost or in operation, as hydraulic transmission is used throughout. It is not as well suited to varying heights of packages as type B unless fitted with extra-long cylinder and ram. It can be provided with hand-adjustable "daylight," but this adjustment would require longer

time and would not be justified unless a fairly large number of the off-height packages were to be run. Although the closing speed can be made faster with this type than with the preceding, the lower platen must return the full travel distance before unloading, but return speed can be made as rapid as desired.

(*D*) *Press With Rotary Pump and Intensifier.* This is a smooth and quiet-operating combination, and is shown in Fig. 5. It is free from pulsation and shock. It closes at a high rate of speed and builds up pressure at a slower speed. When the pressure is building up, the gage finger moves steadily across the dial. The absence of pulsation and shock increases the life of the gage and of the valves and packings, and as there are but eleven motivated parts in the system, including the valves, a high efficiency is attained. A unit which exerts a pressure of 450 tons requires only a 3-hp. motor to operate it. As the pressure is built up steadily, perfect regulation can be had and it is simple to operate.

A valve and an electric push-button station are the only controls necessary, and it is usually equipped with a pressure-regulating gage that can be set to the pressure desired for each package before the "start" button is pressed, and when the

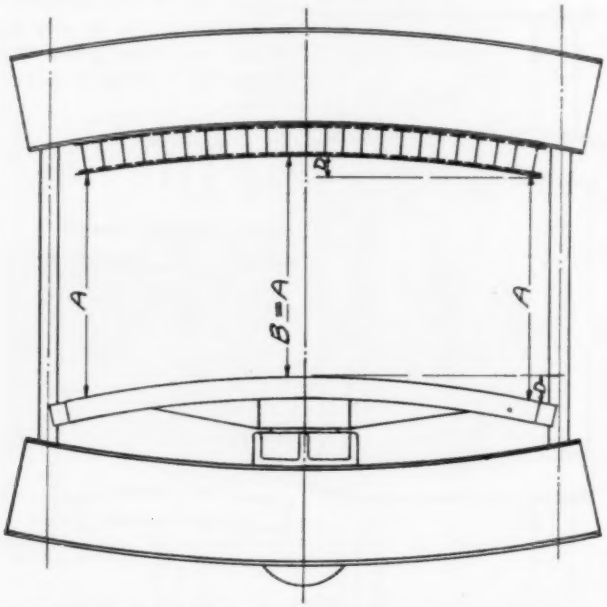


FIG. 10 STRAIN-ROD PRESSURES, CORRECTLY APPLIED
(Side view, outside strain rods, with center piston, showing both platens tending to curve in the same direction.)

pressure reaches the point for which the gage is set, it stops automatically.

SUMMARY OF OPERATING ADVANTAGES

The type D is the most economical press yet devised. Its operating cost is under that of any other type of power press, both in upkeep and power consumption. As all of the operating parts are built in the press frame with the exception of the intensifier, comparatively little floor space is required.

The four types of hydraulic presses described are the types now most generally used, and are all regularly built with an "up stroke;" that is, the cylinder is supported by the bottom frame and is arranged to apply pressure on the work through an upward movement of the ram. This arrangement is most desirable for plywood manufacture as it provides a means for more easily loading and unloading the press and is in other respects more desirable for plywood than inverted-cylinder presses.

RETAINER CLAMPING METHOD

As I-beam retaining clamps are in general use and are placed on the lower press plate previous to loading, the space between the clamp beams provides a convenient place for placing roller bars. These roller bars are made so that their overall height is slightly less than the height of the clamp beams, and when the press platen is raised in the pressing position, the roller bars rest on the face of the platen, and the clamp beams take the entire load. After pressing and tightening the clamps, the platen is lowered. At a point approximately a half-inch above the lowered position of the platen, the roller bars strike on jack stands, which cause them to remain stationary while the platen continues its downward stroke, thereby transferring the weight of the package from the platen to the roller bars. The package (shown in Fig. 6) can then be easily rolled out of the press over an offbearing truck and lowered to the truck by means of the jack stands. The jack stands are usually made to return to the starting or loading position automatically, leaving only the operation of lowering the package on the offbearing truck to manual control, and with the modern improvements, this operation consists of a pull of only 5 lb. on a hand or foot lever.

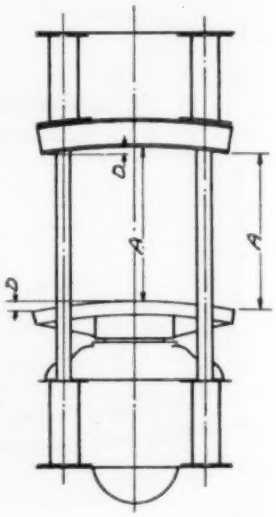


FIG. 11 STRAIN-ROD PRESSURES, CORRECTLY APPLIED
(End view of Fig. 10.)

PRESSURE-GAGE VARIATIONS

There is an impression among some users of pressing equipment in glue rooms that, when the pressure gage on a hydraulic press falls back, the press is losing pressure due to leaking in valves or packings. This is not necessarily the case. Any type of press, whether geared-screw or hydraulic, if provided with a gage to show the actual pressure held on a freshly glued package would show a drop in pressure. This is due mainly to the displacement of glue between the laminations and to a certain amount of compression in the stock, both of which actions continue after the press is stopped, causing the package to become lower and therefore to not offer as much resistance to the pressure applied. The same action takes place in all glue-room presses, but passes unnoticed in electrically gaged screw presses, as the electric gage or ammeter ceases to register as soon as the press movement is stopped, whereas the hydraulic gage registers as long as stock remains under pressure.

FRAME CONSTRUCTION AND PLATEN DESIGN

A feature of major importance in producing good glue joints in plywood manufacture is the method of constructing the frame of the press. The fact that the frame is heavily constructed does not necessarily mean that it is correctly constructed. Excessive deflection in a press frame may be the invisible cause of much spoiled work. It is impossible to build a press entirely free from deflection, but by means of proper construction and use of beams of sufficient depth, the amount of deflection can be minimized and its injurious effects on the work eliminated. There are shown three types of frame construction now in general use, with their deflection curves greatly exaggerated.

The front elevation of a press is shown in Fig. 7, with the pressure both applied and backed at the corners through the

strain rods. This type of construction, when the press is under load, permits the top and bottom members to deflect in opposite directions, causing more pressure to be exerted near the ends of the press than at the center in both directions across the press. This condition can cause a starved joint at the ends of the stock due to the concentrated pressure at this point, while at the center, part of the stock may not even be pressed into contact.

An end elevation of the same construction is shown in Fig. 8. While the same condition exists, it is usually to a much lesser extent, due to the shorter span. If deep presses for wide plywood are built in this manner, however, the deflection in this direction soon reaches serious proportions.

Fig. 7 also represents a front elevation of the type of press using only two strain rods, one at each end of the center. The same deflection curves apply to this press in this direction, and Fig. 9, which is an end elevation of this type of press, shows that under extreme load a large percentage of the pressure can be concentrated at the two points closest to the strain rods, as the deflection in both directions causes the members to curve away from each other in all directions radiating from the strain rods.

The front elevation of the usual method of constructing a hydraulic press is shown in Fig. 10. Although deflection cannot be wholly eliminated, the members for clamping the stock are made to deflect with each other instead of away from each other. This is due to the pressure being applied at the center and backed at the four corners, making it possible to hold the distance between the lower platen and the pressure head the same at all points, thereby delivering uniform pressure at all parts of the work.

An end elevation of the same press with practically the same conditions is shown in Fig. 11.

It is of course understood that the drawings show greatly exaggerated deflection curves, but if one considers a definite condition for an example, the importance of balanced deflection in press-frame construction can be more readily understood.

Let dimension *D*, Fig. 7, equal $^1/_8$ in. when under full-load pressure and *A* equal 30 in.; the *B* would equal 30 plus 2*D*, or $30^1/_4$ in. This means that the work under pressure must either be compressed $^1/_4$ in. more at the ends than at the center or that there is no pressure being applied at the center.

Compare this with a press constructed as shown in Fig. 10 with the same span and using the same members for the top, or in other words, with the same strength. *D* in this case would also equal $^1/_8$ in., and dimension *A* equals 30 in., and as the deflection of both pressure members is in the same direction, dimension *B* also equals 30 in., as well as all other points between the platen and the pressure head. This latter example cites a perfectly proportioned press, but even if the press is not perfectly proportioned, the greatest possible difference between the distances between the pressure members at different points could not possibly be so great as the total deflection of one member, whereas this difference in all other types of presses equals the sum of the total deflection of the top plus the total deflection of the bottom member.

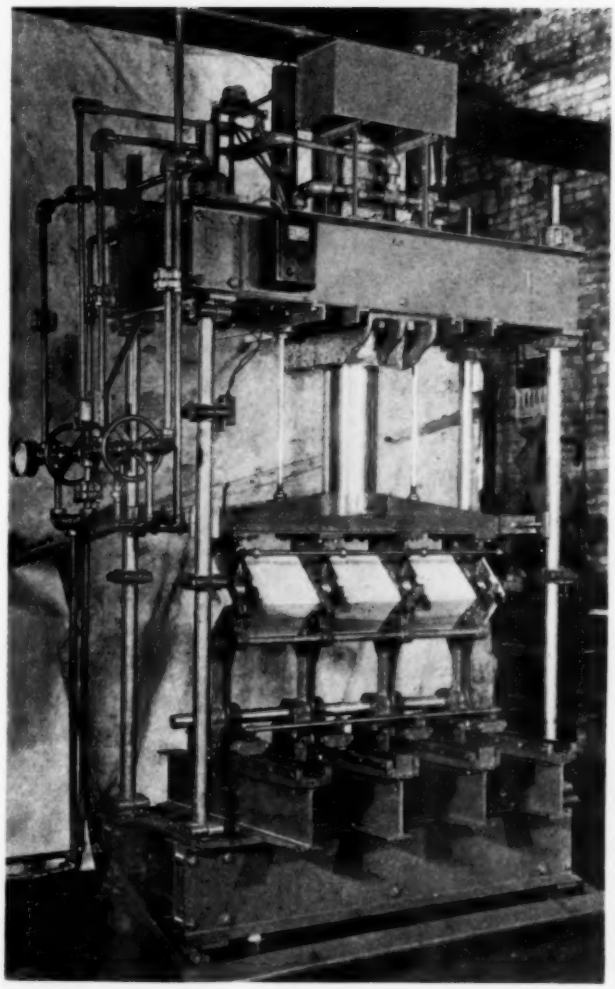


FIG. 12 DOWN-STROKE HYDRAULIC PRESS (INVERTED FROM USUAL TYPE)
(Showing angling platens for gluing up material to make end-block flooring.)

WDI-53-13

Research in Wood Products

By JAMES W. LAWRIE,[1] MILWAUKEE, WIS.

Wood has widely different characteristics, and species may be chosen for a wide range of industrial and construction purposes possessing either high strength-weight ratios or qualities that permit cutting thin veneers, also possessing great ultimate strength or low specific gravity. It varies widely in color, grain structure, and beauty. Wood also has faults and limitations which can be largely corrected through intensive research. Wood expands when wet. Research with the X-ray has shown the reason, and has pointed to a way of rendering wood nearly non-expansive. Wood can be made fire-resistant, but tests must be conducted under both heat and strain conditions to be of practical value. Preventive measures can be taken to lengthen the life of wood by toxic-chemical treatments to eliminate decay. Wood-finishing methods can be greatly improved by scientific adaption of the right principles. There are many other wood qualities that can be improved by modern research methods.



IT IS VERY pleasing to know that the wood industries through the National Lumber Manufacturers' Association are organizing and equipping a research laboratory to undertake the study of wood. Of all the materials which nature has given us, wood was among the earliest used by man in construction work, yet through the ages man has neglected the study of wood and even today is largely in doubt as to what wood is, and why it has the peculiar and interesting physical and chemical properties which make it the indispensable material for variety, beauty, and practicability in construction.

Wide Range of Wood Properties

Wood has certain properties which can be evaluated. Pound for pound, wood, in general, is stronger than steel. It can be obtained in many sizes, in the form of veneer less than 0.01 in. in thickness, in lumber several feet thick, in lengths up to over 100 ft. These are woods that are soft, hard, that are lighter than cork, and heavier than water. Wood varies in color, grain, structure, and beauty. It is moderate in cost, generally available, and can be fitted and used by even unskilled workmen. There is some kind of wood that is useful and usable for almost every construction purpose.

Correcting Wood Faults

Has wood any faults which, if corrected or eliminated, would make it an even more desirable material? It is particularly along the lines of correcting the faults of wood that research should be of assistance to the wood producers and consumers. When wood takes up moisture it expands, when it is heated too highly it chars and burns, when rot fungi get proper conditions for attack they cause rotting, insects eat the wood, and lastly in order to keep the beauty of wood it must be protected from undue wearing.

Why Wood Expands and Contracts

What causes wood, a comparatively soft material, and water, a liquid, when brought together, to expand, and in expanding to exert such enormous forces as to be able to split open large rocks? In fact, the wet wedge is employed commercially to split marble, slate, and other like formations because the lines of splitting can be controlled by proper placement of the wedges. How can two such soft bodies when brought together exert these great forces? Soft wood, liquid water. It seemed that water might react chemically with the cellulose molecules of wood to form a hydrated cellulose with a larger molecule, and that the expansion in size of the molecule is the source of the forces which exist. How can this idea be proved or disproved? If one can prove that there is chemical union or disunion of cellulose and water to form a larger or smaller molecule, and that expansion or contraction is due to these reactions, then why can not the cellulose molecule unite with some other chemical than water to form a permanent union and thereby stabilize the wood so it will no longer contract or expand?

X-rays have been turned to as a means of answering the question as to what is the reaction between water and cellulose, and why are expansion and contraction practically all across the grain of the wood and almost none along the grain? The X-ray work was carried out by Dr. George L. Clark at the University of Illinois. He was able to show definitely that the swelling of wood is a physical and not a chemical reaction. The reason wood expands largely in one direction is that the cellulose in the form of large spiral crystals is packed in close bundles called micelles. Moisture as such cannot penetrate these bundles, but it does penetrate in between the bundles and by capillary force pushes the micelles apart with tremendous force. As the micelle bundles are long in the direction of the grain of the wood and very narrow across the grain the expansion takes place proportionately much more in the cross-grain than in the length. By X-rays it is possible to measure exactly the amount of expansion, the size of the micelles, the number of crystals of cellulose in a micelle, and further the number of molecules of cellulose in the crystals themselves. These established facts have been of great value in pointing out the direction of research to overcome the entrance of moisture between the micelles. Experimentally so far the rate of this entrance of moisture has been cut down to 10 per cent of that of untreated wood taken from the same board. The complete exclusion of moisture from the wood, however, is not looked forward to. Wood is now looked upon as a tank with a large capacity for quickly absorbing or giving up moisture. By use of proper moisture-resisting treatments the rate and to a lesser extent the amount of water or moisture which may penetrate into the wood is cut down, and in this way the wood is stabilized so that under ordinary average conditions there is such slow absorption or evaporation of moisture as not to affect expansion or contraction in wood with continuous changes in humidity. If today humidity is high and tomorrow

---

[1] Director of Chemical Research, A. O. Smith Corp. He received the degree of Ph.D. from the University of Chicago and was engaged as chief chemist for several manufacturing concerns from 1909 to 1919, when he became research and consulting chemist for the E. I. du Pont de Nemours and Co. He joined the A. O. Smith Corp. in 1930.

Contributed by the Wood Industries Division and presented at the Annual Meeting, New York, N. Y., Dec. 1, to 5, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

it is low, there will be no appreciable effect on the treated wood because of the slowness of the rate of absorption or evaporation of the moisture. Over long low humidity or long high humidity periods the contraction or expansion will change, but in considerably lesser total degree than for untreated wood. Apparently one can never hope for a complete freedom from expansion and contraction.

Fire-Resistant Wood

The problem of next greatest interest is probably that of making wood resistant to fire. Again it is realized that no treatment can make wood fireproof, but at best it can be made more or less resistant to fire. There are several factors that must be considered in the study of fire resistance. If wood is to be made only resistant to the passage of fire such as would be the case with partitions and separating walls, then this is a certain type of problem, but if it is to be made resistant to fire and at the same time is under a stress from a load such as is structural timber in a factory, then the question of the effect of the chemicals used in the treatment of the wood becomes at least of as great importance as does the problem of fire resistance. It has been found that many of the chemicals that are used to give wood resistance to burning react on the wood at not very high temperatures and cause a much more rapid destruction of the wood than does the fire itself, and the treated timbers will give way under their loads long before they would be consumed by the fire. This point should be given deep consideration, and methods of testing for fire resistance should at least, in the case of structural timbers, include definite tests on the effect of the chemicals on the wood at varying temperatures up to the ignition temperature. The use of salts soluble in water or aid in fire resistance is wrong in theory. Such salts in nearly every instance are injurious to the wood even at normal temperatures, and chemical action takes place even though very slowly. The salts also tend to leach out gradually or in some cases to volatilize, and therefore gradually render the wood less resistant to fire. Another undesirable feature is that wood treated with water-soluble salts must, in general, be first slowly air dried and later kiln dried, a time-consuming process, and because of time, requiring the carrying in stock of large quantities of treated wood in process.

Preventing Wood Decay

From the tree in the forest to the lumber bought is a pathway of waste and loss. Only about 40 per cent of the tree reaches the consumer. Further than this, the average life of the wood used for all-around construction purposes is only about 10 years. This means that much of the wood lasts only a short time to make up for that portion which lasts a long time. Wood properly taken care of should last for decades. Why the short average life? Wood is subject to attack by fungi which in their life process put out certain chemicals that react on wood to cause it to change in its physical as well as chemical properties. The effect of these changes we call rot. With rot fungi the old paint slogan, "Paint the surface and you save all," is an unhappy misnomer. Rot fungi can work from the inside of the wood out or from an unpainted surface like the back side of a base board through to the painted surface. Again, throughout most of this country and in all tropical countries the ravages of termites, the so-called white flying ants, are causing untold damage to both old and new wooden structures.

Toxics for making wood immune to rot and to attacks by termites and other wood-destroying insects must be toxic to fungi and insects, but should not be toxic to humans. They must be slightly soluble in water or in the liquids of the fungi or insects in order to act as a toxic to them. They should be odorless, practically colorless, should allow for the wood being painted, varnished, lacquered, or given some type of finish to enhance the beauty of permanence of the wood; they must be permanent and must not weaken the physical structure of the wood over a long period of time. Another desirable requirement is that they be easily applied without expensive equipment or the necessity of keeping large stocks of treated lumber on hand because of the long, slow period of drying as is the case when the wood is treated with water solutions of toxic materials. Again the wood should not undergo changes in dimensions due to the treatment, and where complete penetration is not required, the method of treatment should as far as possible sterilize the wood throughout. Where wood has been kiln dried complete sterilization is assured, and if the moisture content is held low then the inside of the wood should remain sterile after the outside is treated. Again, the concentration of toxic materials in the wood must be considerably greater than the "killing concentration" due to the gradual loss of toxic either by volatilization or solution from the wood. Finally, any successful treatment of wood from rot resistance must be reasonable in cost.

Better Finishing Processes

The beauty of wood can be preserved from damage or definite changes by coating its surface with materials which either are less affected by weather and abrasion or which can be readily applied to the surface, such as with varnishes, lacquers, or oils. With some woods the beauty is enhanced by the use of stains, which usually help differentiate the grain of the wood, by fillers which help level the surface, by varnishes, etc. which by differences in defraction bring out more markedly the beauty of wood. Hard woods such as red and white oak are used very extensively for floors. There are several fundamental reasons for the choice of these woods for flooring purposes. The woods are hard and therefore wear well, they have unusual variety and beauty in grain, they do not readily splinter or crack, and they can be easiest enhanced in beauty by varnishing or lacquering.

The problems presented in giving a desirable finish to oak flooring have not been given thorough study in all their aspects. The ordinary practice is to sand the floor, then add the filler coat with a certain amount of rubbing or brushing to remove excess filler. Next shellac, varnish, or lacquer is applied. In many instances the floor is considered as finished, but in some cases a second coat of varnish, etc. is applied and then a coat of wax which is given a so-called polish. The entire procedure is really, at least theoretically, incorrect.

Avoiding Sanding Errors

An analysis of the sanding operation shows that the surface of the wood is torn usually in one direction by means of abrasives. These abrasive materials are not uniform in size nor are they distributed on the sand belt or drum in an absolutely uniform manner. The result of sanding is that the surface of the wood is scored and pressed down in the direction of the movement of the sanding belt. When filler is then applied to such a surface it does not readily enter the open grain of the wood where the wood "whiskers" have been pressed into the grain. The result is that the filler cannot function properly and the average floor when finished presents a very uneven surface. In theory a floor should be given a currying so that the fibers are shredded, but not removed, and they should stand up giving a velvet-like appearance to the surface of the wood. The entire surface of the wood supporting the "whiskers" of the curried wood should be at a uniform level. The next operation should then be the shaving of the whiskers, leaving the surface of the wood level, smooth, and with no matted surface of finely shredded wood. The entire grain of the wood is left open. The appli-

cation of a proper kind of filler to such a surface is not sufficient, however, to insure good filling. The need for filling is to create not only a surface that is level, but also to create a surface which will allow a uniform penetration of the finishing coats of varnish, lacquer, etc. In order to get such a surface it is necessary to work the filler into the soft and open grain of the wood under a pressure which is just great enough to force the filler into the grain, but not great enough to abrade the harder structure of the wood. By the application of a filler that has the proper absorption qualities, and by the use of the required pressure, a surface is created which is entirely different from that of ordinary filled wood.

Wood Fillers

The proper filling of the wood is probably the most important step in the finishing of wood. It is the basic foundation of all the other materials to be applied. It gives a surface to the wood which is smooth and level, and as a result there are no multiplicity of small ridges such as result from the sanding process, and which have little resistance to abrasion. When abrasion starts at the peaks of these ridges, there is a rapid spread of destruction of the finish to large areas, but with the shaving method a surface is presented which is level throughout, and which wears evenly and therefore has a long life.

The question of the use of a seal coat directly after the filler has been the subject of much argument. Seal-coating materials have, in general, been made so as to give a film which has very little penetration into the wood, which is therefore somewhat brittle and subject to chipping by abrasion. The seal coat, however, gives a base for the subsequent finishing coats so that there is almost no penetration into the grain by the finishing coats, and therefore a more even surface is obtained, but they afford a poor surface for adherence of the finishing coats. With the very even surface presented by the newer method of surfacing and filling the wood, the old seal coats are unnecessary, but the use of a film which is intermediate in characteristics between the filled surface of the wood and the finishing coats gives a better set up than the use either of a seal coat or a first finishing coat. The final finish coat should be hard but not brittle. It will therefore stand abrasion well, and with a uniformly even foundation as presented by the previous finishes will wear several times as long as it would when applied under old conditions.

Prefinishing Flooring

The use of oak blocks set in a permanent mastic is rapidly coming into general use especially in homes, apartments, hotels, hospitals, etc. With a floor material of this kind, uniform in size, the prefinishing of the blocks in the factory becomes an interesting problem. It is at once obvious that the usual methods of finishing flooring would be impractical because of the long time period, and because of the very large space required for storage during processing. A study was made of the conditions which would be practical. This study showed that a time limit of two hours from raw to finished block was the maximum that could be allowed. To meet such time conditions it was necessary to develop more exact equipment and new methods of application of the finishing materials. There are no varnishes which will dry within these specified time limits. Lacquers were found not to wear satisfactorily, but with the cooperation of some of the lacquer manufacturers new lacquers were developed which present a hard surface, but which are not brittle. These lacquers age well and retain their hardness and non-brittle properties. With the use of the new methods of surfacing and filling, and the specially designed machinery, it is possible to finish the oak blocks in less than one hour. This finishing process includes surfacing, dust removal, filling, rubbing, removal of dust, first coat, drying, rubbing, second or finish coat, drying, rubbing, wax coat, and polishing.

How Floors Wear

The wear on floors is largely caused by persons walking over the floors. It is not, however, possible to make even fair comparative tests on various finishes on floors by counting the steps of persons walking over the floors. There is a tremendous variation in the weight, pressure, grinding effect of the persons, and in the particular spots where the walking dominates. The weather, dust, and a multitude of other conditions will all have their immeasurable effects. Because of these facts a "walking" machine was designed which gives a uniform pressure of 27 lb. per sq. in. to a disk of sole leather while turning 5 deg. in a circle at the time of contact with the surface. These conditions simulate average conditions of persons walking. The machine operates at the rate of 250 steps per minute, and the samples are run in duplicate, changing the two test pieces every 100,000 steps. Under the old conditions of finishing floors the finish with a good varnish will last for a maximum of 300,000 steps and with a lacquer for about 250,000 steps. With these same finishing materials, but applied by the new methods, the varnish and lacquer coats both stand up for over 1,500,000 steps. Obviously such a test is not quantitative. The wear of the finish is checked by microscopic examination and is therefore open to the error of personal equation, but the results are so outstanding that there is no question about the increased value due to the methods of application. The blocks finished by this method present a beauty which has heretofore been obtained only by the use of high gum varnishes and rubbing by hand.

It has been found, however, for stores, halls, or other places where there are unusual conditions that a polished floor is undesirable. In cases like this after the filler coat has been worked in, it has been found that the treating of the floors with a hot brushing of a solution of drying oils and waxes gives a splendid finish. This solution penetrates into the fibers of the wood and resists to a large degree the effect of moisture or water on the wood, it fills the cells preventing dirt and stain from entering, and it gives a finish that withstands abrasion and wear. It is particularly fine for use in smoking rooms where a varnish or lacquer finish is easily marred by burning cigarettes, etc. This kind of a finish is difficult to burn or char, and any discoloration is usually removed by a slight rubbing with steel wool and the application of some additional treatment.

Conclusion

The foregoing problems are only a few of the many which are apparent in the wood industry. As some measure of success is obtained with one problem other new and important problems suggest themselves. This is the history of research work on all other materials. It is well that it is already showing like history with regard to wood.

A.S.M.E. Record and Index, 1931

THIS section of the Transactions contains a record of the activities of The American Society of Mechanical Engineers for the year 1931, together with an index of its publications. It is issued to every member of the Society in pamphlet form and when bound with the technical papers of the Professional Divisions for the use of libraries and official depositories throughout the world, it completes the permanent record of the Society's activities.

The Record and Index Section of Transactions includes all the reference material likely to be of permanent value and of interest to future searchers. As an index it provides a means of locating the technical information in the Society's publications. It includes the separate indexes of the Transactions and *Mechanical Engineering*, together with a miscellaneous index to reports or other technical publications of the Society that have appeared during the year.

A considerable portion of the Record and Index is devoted to memorial notices of members who have died during the year. Special care has been given to the preparation of these brief biographies with the idea in mind that fitting attention should be given to significant events in the lives of deceased members, even though they may fall outside the immediate field of mechanical engineering.

W. H. WINTERROWD, *Chairman*
L. C. MORROW
S. F. VOORHEES
S. W. DUDLEY
W. F. RYAN
Publications Committee



ROY V. WRIGHT

PRESIDENT OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
1930–1931

Roy V. Wright

AS railroad mechanical engineer and editor for the Simmons-Boardman Publishing Company, Roy V. Wright, president of The American Society of Mechanical Engineers for the term 1930–1931, has given more than thirty years of progressive service to the transportation industry.

Mr. Wright was born at Red Wing, Minn., on October 8, 1876. He attended the public schools of St. Paul, Minn., and the University of Minnesota, from which he received the degree of M.E. in 1898. He was given the honorary degree of Doctor of Engineering by Stevens Institute of Technology in 1931. He is a member of Sigma Xi and Pi Tau Sigma (1931).

Immediately upon graduation he entered the locomotive erecting shop of the Chicago, Milwaukee & St. Paul Railway at South Minneapolis, Minn., as machinist apprentice. For three years he served this railway and the Chicago Great Western as apprentice, draftsman, and chief draftsman. In 1901 he became mechanical engineer for the Pittsburgh & Lake Erie Railroad. This was in the period of early development of the all-steel gondola and hopper freight cars, the P. & L.E. R.R., in the heart of the coal and steel districts, being one of the first railroads to use such cars on a large scale. The locomotive repair shops at McKees Rocks, Pa., were also rebuilt during this period, involving one of the first extensive applications of individual motor drive to machine tools.

The editorial career of Mr. Wright began in 1904, as associate editor of the *American Engineer and Railroad Journal*, now *Railway Mechanical Engineer;* he was made editor of that publication in 1905. In 1910 he became mechanical department editor of the *Railway Age Gazette*, which is now *Railway Age*. In 1911 he was appointed managing editor of that publication, and this position and the editorship of the *Railway Mechanical Engineer*, which he resumed in 1912, he still holds. Since 1910 he has acted as editor of various editions of the Locomotive Cyclopedia, the Car Builders' Cyclopedia, and the Material Handling Cyclopedia. In 1911 he compiled and edited "Railway Shop Kinks" for the International Railway General Foremen's Association. Mr. Wright was elected a director of the Simmons-Boardman Publishing Company in 1915 and was appointed secretary four years later. In December, 1931, he was elected vice-president of the company.

He is also a director and secretary of the Simmons-Boardman Publishing Corporation and the board of the Y.M.C.A. of the Oranges; chairman of the Board of Publication and member of the Administrative, Railroad, and Industrial Com-

mittees of the Home Division of the National Council of the Y.M.C.A.; member of the International Committee of the Y.M.C.A.; trustee of the Silver Bay Association; member of the Silver Bay Human Relations in Industry Conference Committee; member of the board of the Committee on Friendly Relations Among Foreign Students; and a director of the Ampere Bank & Trust Co., East Orange, N.J.

He has been very active in the affairs of the A.S.M.E., which he joined in 1907. He was a member of its Railroad Committee before the formation of the Railroad Division, and a member of the first Metropolitan Section Committee. He served as manager from 1922 to 1925 and as vice-president in 1926 and 1927. He has been a member of the following committees: Meetings of the Society in New York, 1910–1914, and chairman in 1914; Railroad, 1913, 1914, 1915; Meetings and Program, 1919–1923, and chairman in 1921 and 1922; Policy of the Society, 1924; Presentation of Dues Increase, 1924; and Finance, 1926, 1927. From 1928 to 1931 he was a member of the Standing Committee on Awards; he is a member of the Biography Advisory Committee. He was president of the United Engineering Society during 1928 and 1929; was a member of the Committee on the War Memorial to American Engineers at Louvain, Belgium; was a delegate to the First World Power Conference in London in 1924, and honorary vice-president of the American Management Association in representing it at the International Management Congress in Prague, Czechoslovakia, in 1924. He was a member of the American Engineering Council committee which prepared the report on Storage of Industrial Coal, and is a member of the Advisory Board of the Department of Smoke Regulation of Hudson County, N.J. He was one of the organizers of the New York Business Papers Editorial Conference and served as its president in 1918. He is a member of The Franklin Institute, Engineers' Club (New York), Railroad Club of New York, and of the executive committee of the New York Railroad Club.

His contributions in papers and discussions have been an inspiration toward better engineering. He contributed a chapter on Transportation in the symposium "Toward Civilization," edited by Dr. Charles A. Beard and published in the spring of 1930, and he has delivered many addresses on this subject. He was one of the honorary editors of the Professional Divisions' reports on technical developments since 1880, as published in the Fiftieth Anniversary Issue of *Mechanical Engineering*.

The Engineer Militant[1]

By ROY V. WRIGHT

THE engineer has been subjected to harsh and severe criticism in these recent years. While much of this may not have been deserved, it promises to be helpful to the profession—strange as it may seem—if for no other reason than that it is forcing the engineer more thoroughly and critically to examine his part in and contribution to our economic program and progress. Moreover, it is encouraging him also to develop a larger and stronger professional consciousness. Again, it is a clear indication that certain elements, at least, are looking to him for leadership in helping to solve some of our more pressing economic problems.

These are trying days, and in some ways they promise to mark a distinct turning point in the activities of our Society, and in the emphasis on its major objectives. We seem to be at the parting of the ways. With statesmanlike vision and constructive effort we may yet lift the engineering profession to the heights which it should occupy, in the light of its past achievements and of the leadership which it can help to give this present civilization. Stuart Chase has been a harsh critic of the engineer, and yet in his "The Nemesis of American Business" he makes this statement: "Plato once called for philosopher kings. Today the greatest need in all this bewildered world is for philosopher engineers."

What About the Future?

Unfortunately, up to this time the engineer and those with whom he has been associated have been too prone to consider only the immediate results of their work, and have failed to vision the great forces which have been in process of development in our economic, social, and political life as a result of the complications brought about by this machine age, for which the engineer is largely responsible.

Was it a recognition of the engineer's responsibility in these respects that the artist attempted to portray so graphically on one side of the medal commemorating the Fiftieth Anniversary of our Society? Obviously the engineer pictured is looking over and beyond the tools and instruments in the foreground and is visualizing a world raised to higher levels and standards. Can the engineer render a larger or more important service than to insure the total results of his work, direct and indirect, being entirely in the interests of humanity? Those are indeed challenging words on the medal: "What Is Not Yet, May Be."

These problems and others have been brought home to me most forcefully as I have journeyed about among you this past year and have discussed with you some of the larger problems and challenges which are facing us. Quite naturally, therefore, there are a number of important tendencies concerning which I should like to talk with you this evening, if there were more time. I shall attempt rather to comment briefly upon only a few of the high spots.

Civic Responsibilities

We ought to give more serious consideration to the building up of a stronger professional consciousness, which will weld the engineers more closely together and inspire and encourage them to greater and more constructive unified efforts in the interests of humanity. Of all men the engineer should take a large and intelligent interest in helping to control all of the forces which have been set up as the result of our more complicated existence in this era of mass production. The very success of our nation depends upon the intelligence of its citizens and the extent to which they use this intelligence in working out problems of government—and these problems are steadily and rapidly becoming more serious.

Is it not strange, under these conditions, that students are being graduated from our colleges and universities, many of which are supported largely by public funds, without having any clear understanding of the elements of citizenship, or how they can best serve their communities and country? Engineers must take these responsibilities more seriously—as individual citizens or as groups in assisting their localities in city and county planning, or collectively by taking a larger part in public affairs of the state and nation.

We can, for instance, well take pride in and ought to support generously and with enthusiasm the work of the American Engineering Council, which represents the engineer in public affairs and has done so much also to bring the engineering attitude to bear in important economic studies. One of our members, L. P. Alford, was recently awarded the Gantt Medal, partly, at least, because of the outstanding part that he has taken in the making of these engineering economic studies in the public interest. Our President-Elect, Conrad N. Lauer, was mindful of the engineers' responsibility for public service when he founded the Hoover Medal, the purpose of which, in the words of President Hoover, "is to mark the public service of men who have gone outside their strictly professional work to interest themselves in civic and humanitarian affairs."

Elevating Professional Status

Study and discussion of the report on the status of the mechanical engineer, which was so ably prepared under the direction of a committee headed by President-Elect Lauer, indicate that there are possibilities of greatly improving the status of the engineer, which almost automatically should bring home to him more forcefully his responsibilities to the community at large.

If this is to be done thoroughly, however, it means a rebuilding of our structure from the very foundation. One of the evidences of an intelligent appreciation of this is the fact that more and more thought is being given to the better selection of young men who enter our engineering schools. In some instances members of the faculties of engineering colleges are getting in touch with the high-school authorities and are trying to locate, particularly through the science teachers, boys who seem to be particularly well adapted for engineering and who should be encouraged to enter that profession. Stevens Institute of Technology, under the direction of its president, Dr. Harvey N. Davis, also one of our vice-presidents, this last summer gathered a group of junior-high-school students at its camp in northern New Jersey, with a view to giving them a thorough understanding of what engineering is, and of the requirements for success in its various phases; at the same time an attempt was made scientifically to study and analyze each boy, with a view to encouraging those who were specially fitted for engineering to enter that profession.

Our Student Branches

Our Society has done good work with its Student Branches,

[1] Presidential Address, at the Annual Meeting, New York, N.Y., November 30 to December 4, 1931, of The American Society of Mechanical Engineers.

but has not approached the problem with the same type of statesmanship as in the supervision of the Local Sections and the Professional Divisions. The committee in charge of this work is to be congratulated upon the forward step that it is now taking. The experiment with the new set-up is at present confined to 15 Student Branches in the Southeast, but there are good reasons even now to believe that it will be a success and that, when extended to all of our Student Branches, it will more closely integrate the activities of our Society with the engineering colleges. Rightly directed, this means that it can be of very great assistance to the engineering faculties and students in a practical way, and will help greatly to develop a proper professional consciousness among the students; it should also, almost automatically, bring these young men into the membership of the Society. We have to too great an extent neglected these students. It will mean much to them if in the course of their training they can touch elbows with the experienced and practical engineer. Indeed, their future success depends largely upon the extent to which we reach out a helping hand to them.

The Junior Members

We have been so intent and so serious in promoting our technical programs that we have lost sight of some of the large opportunities for strengthening the Society and insuring its future stability. What, for instance, have we done in a constructive way to encourage and help the young graduate engineer? He needs our friendship as well as our helpful interest and encouragement. If he has come to our meetings, too frequently we have paid little or no attention to him, or have given him the cold shoulder. This neglect has not been intentional. We have simply failed to see the young man and to realize how much he needs coaching and encouragement; and too often we have lost him from our membership, possibly never to return. It is a question as to whether any other activity needs our attention so much at this time as the development of constructive and aggressive plans to assimilate the Juniors into our activities. They need us, but we need them with their youth, enthusiasm, aggressiveness, and courage. And then, too, this is one of the very foundation stones for building a larger and stronger professional consciousness.

Technical Meetings Cold and Inefficient

Engineers are not cold-blooded. Indeed, as we come to know each other as individuals, we find that we are intensely human. For some reason, however, our meetings have lacked much of that spirit of warmth and good-fellowship which is so vital if we are to come to understand each other better and to be mutually helpful to as great an extent as possible. Judging from numerous comments that I have received, many members fail to attend meetings because of the lack of this spirit. Some of our Local Sections are deliberately organizing to introduce a spirit of good-fellowship into their meetings.

We have been careless also in attending to the appointments and details which go to make the technical handling of the meetings a success. It is not enough to find some one who will prepare a paper or make an address, send out a formal notice to that effect, and then perfunctorily carry out the program. Staging a successful technical meeting is an art. Incidentally, even an experienced speaker has difficulty in overcoming some of the handicaps that are thoughtlessly placed in his way. Even a machine must be treated with reasonable consideration, but fine discrimination must be used with a human being if he is to give the best that is in him. Oh! how careless and thoughtless some of our members and groups can be—and never even realize it! Surely a profession that prides itself upon eliminating waste and increasing efficiency in industrial operations can master the technique of putting over successful technical programs.

We have not made the best of jobs in selling ourselves, either to the mechanical engineers who are not members of our Society or to the public. We can well take a lesson from the constructive piece of publicity work that has been done by the chemical engineers. Few things will do more to enhance our prestige or develop a larger professional consciousness than a carefully planned publicity campaign—and, personally, I believe this can well be done by the engineering societies jointly. It will cost money, but, wisely directed, should far more than pay for itself.

Engineers, Not Specialists

Up to this point I have carefully refrained from mentioning the mechanical engineer as such. Not that I am not proud of the profession and of this great Society which so ably represents it, with its multitudinous and far-flung activities. Basically however, we are engineers, having the same fundamental training and, in a large way, the same objectives as other branches of engineering. We are seeing recognition of this in the fact that engineering colleges more and more are thinking in terms of the engineer rather than of the specialist. We see it in the extent to which graduate engineers shift from one type of engineering to another. We see it in the tendency for specialized groups of engineers in a community to combine in federations or all-inclusive clubs, each type of engineering functioning as a group within the larger group. We see it again in the extent to which our national engineering societies more and more are engaging in joint activities—in technical meetings, in our Engineering Societies Library, in public service through the American Engineering Council, and now, let us hope, in a joint program looking toward the elevation of the status of the engineer.

Many years ago one of our most honored members, a past-president, Ambrose Swasey, made possible the formation of the Engineering Foundation. With clear vision and unfaltering step he has never lost faith in this joint activity for the promotion of engineering research. Within recent months he has generously added to his already liberal contribution to this project—a challenge to all of us to put forth greater efforts in this worthy joint activity.

Rapid shifts which have taken place in our industrial and economic life in this era of mass production have brought home to us in no uncertain manner the fact that the engineer must be so trained that he can readily adapt himself to new and greatly changed conditions. Indications are that such changes may be even more numerous and radical in the days to come. Just as business and industry should protect the worker by cooperation to provide a continuous training throughout his service life, in order that he may keep up with the advancement of skill in his craft, so the engineer must have a broad basic training in order that he may readily adjust himself to new and unforeseen requirements. This again emphasizes the fact that we should train engineers—not specialists.

Wanted—Militant Engineers!

Vice-President Ralph E. Flanders has recently published a book entitled, "Taming Our Machines." He is not a pessimist. Indeed, those of us who know him well have been impressed by his optimism and his enthuiastic, constructive attitude. He is a builder, not a wrecker. Listen to this challenge from the last chapter of his book:

> If our age of machines cannot assure, on the whole, a steady and considerable flow of sustenance and the opportunity for leisure for all who are able and willing to work, it has failed in the end which it was best adapted to serve. The whole social fabric built upon our machines will then slowly crumble and decay.
>
> The process of decay, however, will be hastened by direct attacks

which are being organized at this moment. The political battles of the present and future are visibly concentrating about our economic failures. If we, who are the sponsors of the machines and of the industrial society built upon them, can successfully organize them to the ends of general human satisfaction, and if we can join to ourselves the other groups whose cooperation is essential, or if we can join ourselves to them in some effective way, then this civilization which we have built will endure. Otherwise it will perish. No mere defensive tactics will avail in this critical undertaking.

Stuart Chase calls for philosopher engineers, but we need something more than this. We need militant engineers—aggressive and constructive—who will enthusiastically devote their energies to building up the engineering profession, raising it to higher levels, and will vigorously take the offensive in helping to find a solution of some of the great economic problems which we are now facing.

The American Society of Mechanical Engineers

THE personnel records and other information presented in this first section of the Record and Index refer, except where otherwise noted, as in the case of Local Section and Student Branch chairmen, to the official year beginning in December, 1930. Such information is purely of historical nature and is included here as a permanent record. Further information was published in the Membership List for 1931.

Officers and Council

Dates in parentheses denote expiration of terms

PRESIDENT

ROY V. WRIGHT

PAST-PRESIDENTS

Terms expire in December

WILLIAM F. DURAND (1931) — CHARLES M. SCHWAB (1933)
WILLIAM L. ABBOTT (1932) — ALEX DOW (1934)
CHARLES PIEZ (1935)

VICE-PRESIDENTS

Terms expire December, 1931	*Terms expire December, 1932*
PAUL DOTY	HARVEY N. DAVIS
RALPH E. FLANDERS	WILLIAM A. HANLEY
ERNEST L. JAHNCKE	THOMAS R. WEYMOUTH
CONRAD N. LAUER	

MANAGERS

Terms expire December, 1931	*Terms expire December, 1932*
CHARLES M. ALLEN	HAROLD V. COES
ROBERT M. GATES	JAMES D. CUNNINGHAM
ELY C. HUTCHINSON	CLARENCE F. HIRSHFELD

Terms expire December, 1933

WILLIAM L. BATT — HAROLD L. DOOLITTLE — HERBERT L. WHITTEMORE

TREASURER	SECRETARY
ERIK OBERG	CALVIN W. RICE

EXECUTIVE COMMITTEE OF COUNCIL

ROY V. WRIGHT, *Chairman* — HARVEY N. DAVIS
CONRAD N. LAUER, *Vice-Chairman* — ROBERT M. GATES
CALVIN W. RICE, *Secretary* — WILLIAM L. BATT

Advisory Members: Chairmen of the Finance Committee, the Local Sections Committee, and the Professional Divisions Committee.

CHAIRMEN OF STANDING COMMITTEES

Representatives on Council but without vote

Finance, F. A. SCHAFF, *Chairman*
DAVID LOFTS, *Vice-Chairman*
Meetings and Program, G. M. EATON
Publications, F. V. LARKIN
Membership, F. A. WALDRON
Professional Divisions, H. W. BROOKS
Local Sections, H. R. WESTCOTT
Constitution and By-Laws, A. D. BLAKE
Awards, H. L. SEWARD
Relations with Colleges, M. C. MAXWELL
Education and Training for Industries, R. L. SACKETT
Library, A. S. MILLER
Standardization, L. K. SILLCOX
Research, WM. REUBEN WEBSTER
Power Test Codes, F. R. LOW
Safety, H. W. MOWERY
Professional Conduct, JOHN HUNTER

EXECUTIVE SECRETARY	EDITOR
C. E. DAVIES	GEORGE A. STETSON

ASSISTANT SECRETARIES

ERNEST HARTFORD
C. B. LEPAGE

Standing Committees

Dates in parentheses denote expiration of terms

FINANCE

F. A. SCHAFF, *Chairman and Representative on Council* (1931)
DAVID LOFTS, *Vice-Chairman* (1932) — JOHN H. LAWRENCE (1934)
KINGSLEY L. MARTIN (1933) — E. A. MULLER (1935)

Council Representatives:
H. V. COES (1931)
ELY C. HUTCHINSON (1932)

MEETINGS AND PROGRAM

G. M. EATON, *Chairman and Representative on Council* (1931)
F. M. FEIKER (1932) — C. P. BLISS (1934)
J. W. PARKER (1933) — R. I. REES (1935)

PUBLICATIONS

F. V. LARKIN, *Chairman and Representative on Council* (1931)
W. H. WINTERROWD (1932) — S. F. VOORHEES (1934)
L. C. MORROW (1933) — S. W. DUDLEY (1935)

(*Personnel of Special Committee, p. 11*)

MEMBERSHIP

F. A. WALDRON, *Chairman and Representative on Council* (1931)
H. W. BUTLER (1932) — OTTO E. GOLDSCHMIDT (1934)
HOSEA WEBSTER (1933) — HENRY A. LARDNER (1935)

PROFESSIONAL DIVISIONS

H. W. BROOKS, *Chairman and Representative on Council* (1931)
W. F. DIXON (1932) — C. B. PECK (1934)
P. T. SOWDEN (1933) — W. A. SHOUDY (1935)

(*Chairmen of Professional Divisions' Executive Committees, p. 11*)

LOCAL SECTIONS

HARRY R. WESTCOTT, *Chairman and Representative on Council* (1931)
CHARLES W. BENNETT (1932) — JILES W. HANEY (1934)
JAMES M. TODD (1933) — WM. LYLE DUDLEY (1935)

(*Chairmen of Local Sections' Executive Committees, p. 11*)

CONSTITUTION AND BY-LAWS

A. D. BLAKE, *Chairman and Representative on Council* (1931)
THOS. C. MCBRIDE (1932) — H. H. SNELLING (1934)
R. S. NEAL (1933) — P. R. FAYMONVILLE (1935)

AWARDS

H. L. SEWARD, *Chairman and Representative on Council* (1931)
A. M. GREENE, JR. (1932) — F. L. EIDMANN (1934)
K. H. CONDIT (1933) — W. L. BATT (1935)

RELATIONS WITH COLLEGES

M. C. MAXWELL, *Chairman and Representative on Council* (1931)
D. B. PRENTICE (1932) — W. L. ABBOTT (1934)
E. F. CHURCH, JR. (1933) — LEWIS F. GORDON (1935)

(*Student Branches and Honorary Chairmen, p. 11*)

EDUCATION AND TRAINING FOR THE INDUSTRIES

R. L. SACKETT, *Chairman and Representative on Council* (1934)
D. C. JACKSON (1931) — H. S. FALK (1933)
JOHN T. FAIG (1932) — C. F. BAILEY (1935)

LIBRARY

ALTEN S. MILLER, *Chairman and Representative on Council* (1931)
GEORGE F. BATEMAN (1932) — E. P. WORDEN (1934)
W. M. KEENAN (1933) — THE SECRETARY, CALVIN W. RICE

RESEARCH

Organized 1909, to supervise all research activities, collaborating with committees of kindred societies, and obtaining results of researches conducted in the U. S. and other countries

W. Reuben Webster, *Chairman and Representative on Council* (1931)

W. H. Fulweiler (1932)
Alex. D. Bailey (1933)
Andrey A. Potter (1934)
F. C. Biggert, Jr. (1935)

STANDARDIZATION

Organized April, 1911, to supervise all standardization activities of the Society. Cooperates with American Standards Association

L. K. Sillcox, *Chairman and Representative on Council* (1932)

H. B. Taylor (1931)
W. S. Monroe (1933)
E. Buckingham (1934)
B. H. Blood (1935)

POWER TEST CODES

Organized December, 1918, to revise and extend the Power Test Codes of the Society. The Codes had been formulated by various committees appointed from 1886 to that date

Fred R. Low, *Chairman and Representative on Council* (1935)

Term Expires 1931

A. G. Christie
P. Diserens
C. E. Lucke
G. A. Orrok
W. M. White

Term Expires 1932

H. Cooke
E. R. Fish
O. P. Hood
H. B. Oatley
W. J. Wohlenberg

Term Expires 1933

E. H. Brown
I. E. Moultrop
G. A. Horne
L. S. Marks
E. N. Trump

Term Expires 1934

C. H. Berry
F. Hodgkinson
D. S. Jacobus
L. F. Moody
E. B. Ricketts

Term Expires 1935

F. R. Low
A. T. Brown
R. H. Fernald
C. F. Hirshfeld
R. J. S. Pigott

SAFETY

Appointed October, 1921, to extend knowledge of safety, to promote cooperation in this field and to supervise all safety code activities of Society with exception of those of Boiler Code group of committees

H. W. Mowery, *Chairman and Representative on Council* (1933)

L. R. Palmer (1931)
L. Price (1932)
T. A. Walsh, Jr. (1934)
M. H. Christopherson (1935)

PROFESSIONAL CONDUCT

John Hunter, *Chairman and Representative on Council* (1931)

William B. Powell (1932)
A. G. Christie (1933)
H. S. Philbrick (1934)
D. Robert Yarnall (1935)

Special Committees

BOILER CODE

Fred R. Low, *Chairman*
D. S. Jacobus, *Vice-Chairman*
C. W. Obert, *Honorary Secretary*
M. Jurist, *Acting Secretary*
H. E. Aldrich
Wm. H. Boehm
Frank S. Clark
Francis W. Dean
W. F. Durand
Edwards R. Fish
Vincent M. Frost
Charles E. Gorton
Arthur M. Greene, Jr.
Frank B. Howell
J. O. Leech
Edward F. Miller
M. F. Moore
I. E. Moultrop
C. O. Myers
James Partington
C. L. Warwick
H. LeRoy Whitney

Honorary Members

Thomas E. Durban
Charles L. Huston
William F. Kiesel, Jr.
H. H. Vaughan

TELLERS OF ELECTION FOR OFFICERS

W. J. Hargest Henry V. Oberg Phillip S. Atkinson

REGULAR NOMINATING COMMITTEE

GROUP	REPRESENTATIVE	ALTERNATE
I	S. J. Teller	R. E. Newcomb F. S. Clark
II	C. P. Tolman	A. A. Adler
III	Harte Cooke	Alfred Iddles F. A. Allner
IV	W. R. Woolrich, *Secretary*	D. E. Kehoe
V	A. E. White, *Chairman*	W. M. Taylor
VI	W. C. Lindemann	C. A. Herrick
VII	Dennistoun Wood	A. J. Dickie R. L. Mahon

LOCAL SECTIONS IN NOMINATING COMMITTEE GROUPS

GROUP I

Boston
Bridgeport
Green Mountain
Hartford
Meriden
New Britain
New Haven
Norwich
Providence
Waterbury
Western Mass.
Worcester

GROUP II

Metropolitan (N. Y.) and Foreign Members

GROUP III

Anthracite-Lehigh Valley
Baltimore
Central Pa.
Ontario
Philadelphia
Plainfield
Rochester
Schenectady
Susquehanna
Syracuse
Utica
Washington, D. C.

GROUP IV

Atlanta
Birmingham
Charlotte
Chattanooga
Florida
Greenville
Houston
Knoxville
Memphis
New Orleans
North Texas
Raleigh
Savannah
Virginia

GROUP V

Akron-Canton
Buffalo
Cincinnati
Cleveland
Columbus
Dayton
Detroit
Erie
Indianapolis
Louisville
Peninsula
Pittsburgh
Toledo
West Virginia
Youngstown

GROUP VI

Chicago
Kansas City
Mid-Continent
Milwaukee
Minneapolis
Nebraska
Rock River Valley
St. Joseph Valley
St. Louis
St. Paul
Tri-Cities

GROUP VII

Colorado
Inland Empire
Los Angeles
Oregon
San Francisco
Utah
Western Washington

REVENUES

W. L. Batt, *Chairman*
James D. Cunningham
Dexter S. Kimball
Ralph E. Flanders
Conrad N. Lauer
Erik Oberg
James L. Walsh

BUDGETING POLICY COMMITTEE

Ralph E. Flanders, *Chairman*
J. D. Cunningham, *Vice-Chairman*
W. L. Abbott
Harold V. Coes
C. F. Hirshfeld
Leo Loeb
Harry R. Westcott

Erik Oberg, *Treasurer*
F. A. Schaff, *Chairman, Finance Committee*
W. L. Batt, *Chairman, Revenues Committee*

ECONOMIC STATUS OF THE ENGINEER

CONRAD N. LAUER, *Chairman*
C. F. HIRSHFELD
DEXTER S. KIMBALL
WM. E. WICKENDEN
HENRY B. OATLEY
W. A. STARRETT
H. L. WHITTEMORE
JAMES M. TODD, *representing Committee on Local Sections*

BIOGRAPHY ADVISORY COMMITTEE

F. R. LOW, *Chairman*
ROY V. WRIGHT, *Vice-Chairman*
GEO. A. ORROK
JOHN R. FREEMAN
RALPH E. FLANDERS

Professional Divisions

For complete personnel of executive and subcommittees consult 1931 Membership List

Division	Executive Committee Chairmen
Aeronautic	ELMER A. SPERRY, JR.
Applied Mechanics	A. L. KIMBALL
Fuels	W. J. WOHLENBERG
Hydraulic	E. M. BREED
Iron and Steel	W. TRINKS
Machine-Shop Practice	F. C. SPENCER
Management	WM. B. FERGUSON
Materials Handling	F. D. CAMPBELL
National Defense	JAMES L. WALSH
Oil and Gas Power	L. R. FORD
Petroleum	W. G. HELTZEL
Power	W. F. RYAN
Printing Industries	E. P. HULSE[1]
Railroad	ELIOT SUMNER
Textile	PAUL A. MERRIAM
Wood Industries	THOMAS D. PERRY

Local Sections

For complete personnel for 1930–1931 and further information consult 1931 Membership List

Section	Executive Committee Chairmen[2]
Akron-Canton	H. E. COOK
Anthracite-Lehigh Valley	F. L. MOORE
Atlanta	L. F. GORDON
Baltimore	F. L. LEILICH
Birmingham	F. H. CROCKARD
Boston	M. D. ENGLE
Bridgeport	K. C. MONROE
Buffalo	C. E. HARRINGTON
Central Pennsylvania	C. L. ALLEN
Charlotte	E. M. POTTER
Chattanooga	NEWELL SANDERS
Chicago	C. B. COLE
Cincinnati	R. E. W. HARRISON
Cleveland	T. F. GITHENS
Colorado	F. H. PROUTY
Columbus	H. M. BUSH
Dayton	C. W. RIPSCH
Detroit	C. S. GLADDEN
Erie	F. M. DARNER
Florida	ERNEST KREHER
Green Mountain	E. R. STUBBS
Greenville	D. J. KERR
Hartford	W. S. PAINE
Houston	S. Q. BATES
Indianapolis	W. M. TAYLOR
Inland Empire	U. B. HOUGH
Kansas City	E. L. HUGHES
Knoxville	J. C. WHITE
Los Angeles	J. T. HOLMES
Louisville	B. M. BRIGMAN
Memphis	W. D. EDWARDS
Meriden	K. W. DECHERD
Metropolitan	A. T. BROWN
Mid-Continent	A. J. KERR
Milwaukee	HANS DAHLSTRAND
Minneapolis	C. F. SHOOP
Nebraska	C. F. MOULTON
New Britain	H. S. HALL
New Haven	G. R. HOLMES
New Orleans	G. R. HAMMETT
North Texas	ELMER SCHMIDT
Norwich	WM. H. BUCK
Ontario	S. L. FEAR
Oregon	M. C. PHILLIPS
Peninsula	F. H. MEYER
Philadelphia	K. M. IRWIN
Pittsburgh	VAN A. REED, JR.
Plainfield	P. F. NYDEGGER
Providence	P. N. KISTLER
Raleigh	L. L. VAUGHAN
Rochester	L. P. WILLSEA
Rock River Valley	W. M. DUNDORE
St. Joseph Valley	C. C. WILCOX
St. Louis	T. A. PECK
St. Paul	E. B. CURRY
San Francisco	E. G. SHEIBLEY
Savannah	D. E. KEHOE
Schenectady	T. W. HOWARD
Susquehanna	W. J. FISHER
Syracuse	Z. G. DEUTSCH
Toledo	C. V. MARTIN
Tri-Cities	W. P. HUNT
Utah	ARTHUR ERICKSON
Utica	WALTER CLEMENT
Virginia	V. SERBELL
Washington, D. C.	I. H. FULLMER
Waterbury	W. K. SIMPSON
Western Massachusetts	R. E. NEWCOMB
West Virginia	C. E. BLIVEN
Western Washington	W. L. DUDLEY
Worcester	J. W. ARMOUR
Youngstown	W. G. ARMSTRONG

[1] Resigned March, 1931; A. C. Jewett appointed to complete term.
[2] Term of office, July, 1930–July, 1931.

Student Branches

For further information consult 1931 Membership List

Name and Location	Honorary Chairmen[3]
Akron, Univ. of, Akron, Ohio	F. S. GRIFFIN
Alabama Polytechnic Inst., Auburn, Ala.	SOLON DIXON
Alabama, Univ. of, University, Ala.	J. M. GALLALEE
Arkansas, Univ. of, Fayetteville, Ark.	J. T. STRATE
Armour Inst. of Technology, Chicago, Ill.	J. C. PEEBLES
Brooklyn, Polytechnic Inst. of, Brooklyn	J. E. SETCHELL
Brown Univ., Providence, R. I.	J. A. HALL
Bucknell Univ., Lewisburg, Pa.	FRANK E. BURPEE
California Inst. of Technology, Pasadena	W. H. CLAPP
California, Univ. of, Berkeley, Calif.	H. B. LANGILLE
Carnegie Inst. of Tech., Pittsburgh, Pa.	T. G. ESTEP
Case School of Applied Science, Cleveland	C. L. TUVE
Catholic Univ. of America, Washington	G. H. HEINE
Cincinnati, Univ. of, Cincinnati, Ohio	J. W. BUNTING
Clarkson College of Tech., Potsdam, N. Y.	JESS H. DAVIS
Clemson College, Clemson College, S. C.	C. P. PHILPOT
Colorado Agricultural College, Fort Collins	L. D. CRAIN
Colorado, Univ. of, Boulder, Colo.	FRANK S. BAUER
Columbia Univ., New York, N. Y.	F. L. EIDMANN
Cooper Union, New York, N. Y.	G. F. BATEMAN
Cornell Univ, Ithaca, N. Y.	C. O. MACKEY
Delaware, Univ. of, Newark, Del.	J. L. COYLE
Detroit, Univ. of, Detroit, Mich.	F. J. LINSENMEYER
Drexel Inst., Philadelphia, Pa.	C. V. HAHN
Florida, Univ. of, Gainesville, Fla.	P. O. YEATON
George Washington Univ., Washington, D.C.	WALTER B. LAWRENCE
Georgia School of Technology, Atlanta, Ga.	RAY M. MATSON
Harvard Univ., Cambridge, Mass.	L. S. MARKS
Idaho, Univ. of, Moscow, Idaho	H. F. GAUSS
Illinois, Univ. of, Urbana, Ill.	C. W. HAM
Iowa State College, Ames, Ia.	R. A. NORMAN
Iowa, State Univ. of, Iowa City, Ia.	RALPH M. BARNES
Johns Hopkins Univ., Baltimore, Md.	A. G. CHRISTIE
Kan. State Agricultural College, Manhattan	J. P. CALDERWOOD
Kansas, Univ. of, Lawrence, Kan.	RALPH S. TAIT

[3] In office December 31, 1931.

Name and Location	Honorary Chairmen
Kentucky, Univ. of, Lexington, Ky.	Carter C. Jett
Lafayette College, Easton, Pa.	Paul B. Eaton
Lehigh Univ., Bethlehem, Pa.	M. C. Stuart
Louisiana State Univ., Baton Rouge, La.	W. Whipple
Louisville, Univ. of, Louisville, Ky.	H. H. Fenwick
Lowell Textile Inst., Lowell, Mass.	H. J. Ball
Maine, Univ. of, Orono, Me.	Harry Watson
Marquette Univ., Milwaukee, Wis.	John E. Schoen
Mass. Inst. of Technology, Cambridge	C. E. Fuller
Michigan College of Mining and Technology, Houghton, Mich.	R. R. Seeber
Michigan State College, East Lansing, Mich.	Lawrence N. Field
Michigan, Univ. of, Ann Arbor, Mich.	R. S. Hawley
Minnesota, Univ. of, Minneapolis, Minn.	B. J. Robertson
Mississippi A. & M. College, A. & M. College	O. D. M. Varnado
Missouri School of Mines and Metallurgy, Rolla, Mo.	R. O. Jackson
Missouri, Univ. of, Columbia, Mo.	J. R. Wharton
Montana State College, Bozeman, Mont.	R. T. Challender
Nebraska, Univ. of, Lincoln, Neb.	C. A. Sjogren
Nevada, Univ. of, Reno, Nev.	Wayne Buerer
Newark College of Engineering, Newark, N. J.	J. Ansel Brooks
New Hampshire, Univ. of, Durham, N. H.	Edward T. Donovan
New York, College of City of, New York	A. Bruckner
New York Univ., New York, N. Y.	Arthur C. Coonradt
North Carolina State College, Raleigh, N. C.	J. M. Foster
North Carolina, Univ. of, Chapel Hill, N. C.	E. G. Hoefer
North Dakota Agricultural College, Fargo, N. D.	Robert M. Dolve
North Dakota, Univ. of, Grand Forks, N. D.	A. J. Diakoff
Northeastern Univ., Boston, Mass.	J. W. Zeller
Notre Dame, Univ. of, Notre Dame, Ind.	C. C. Wilcox
Ohio Northern Univ., Ada, Ohio	John A. Needy
Ohio State Univ., Columbus, Ohio.	W. T. Magruder
Oklahoma A.&M. College, Stillwater, Okla.	V. W. Young
Oklahoma, Univ. of, Norman, Okla.	E. F. Dawson
Oregon State Agricultural College, Corvallis	R. E. Summers
Pennsylvania State College, State College	C. L. Allen
Pennsylvania, Univ. of, Philadelphia, Pa.	John A. Prior
Pittsburgh, Univ. of, Pittsburgh, Pa.	S. J. Tracy, Jr.
Porto Rico, Univ. of, Mayaguez, P. R.	Ramon Gil
Pratt Inst., Brooklyn, N. Y.	R. B. Dale
Princeton Univ., Princeton, N. J.	A. M. Greene, Jr.
Purdue Univ., W. Lafayette, Ind.	H. L. Solberg
Rensselaer Polytechnic Inst., Troy, N. Y.	G. K. Palsgrove
Rhode Island State College, Kingston, R. I.	Royal L. Wales
Rice Inst., Houston, Tex.	J. V. Pennington
Rose Polytechnic Institute, Terre Haute, Ind.	Carl Wischmeyer
Rutgers Univ., New Brunswick, N. J.	R. C. H. Heck
Santa Clara, Univ. of, Santa Clara, Calif.	G. L. Sullivan
South Dakota, Univ. of, Vermilion, S. D.	M. W. Davidson
Southern California, Univ. of, Los Angeles	T. T. Eyre
Stanford Univ., Stanford University, Calif.	C. N. Cross
Stevens Institute of Technology, Hoboken, N. J.	K. Davidson
Swarthmore College, Swarthmore, Pa.	G. A. Bourdelais
Syracuse Univ., Syracuse, N. Y.	S. T. Hart
Tennessee, Univ. of, Knoxville, Tenn.	D. White
Texas, A. & M. College of, College Station	J. A. Trail
Texas Technological College, Lubbock, Tex.	C. L. Svensen
Texas, Univ. of, Austin, Tex.	Carl J. Eckhardt, Jr.
Tufts College, Tufts College, Mass.	E. MacNaughton
U. S. Naval Academy, Post Graduate School, Annapolis	P. J. Kiefer
Utah, Univ. of, Salt Lake City, Utah	W. J. Cope
Vanderbilt Univ., Nashville, Tenn.	F. J. Reed
Vermont, Univ. of, Burlington, Vt.	F. H. Sibley
Villanova College, Villanova, Pa.	J. S. Morehouse
Virginia Polytechnic Inst., Blacksburg, Va.	N. W. Conner
Virginia, Univ. of, Charlottesville, Va.	A. F. Macconochie
Washington, State College of, Pullman, Wash.	F. W. Candee
Washington Univ., St. Louis, Mo.	R. R. Tucker
Washington, Univ. of, Seattle, Wash.	R. H. G. Edmonds
West Virginia Univ., Morgantown, W. Va.	C. R. Jones
Wisconsin, Univ. of, Madison, Wis.	J. W. McNaul
Worcester Polytechnic Inst., Worcester	Kenneth G. Merriam
Wyoming, Univ. of, Laramie, Wyo.	Robt. Rhoads
Yale Univ., New Haven, Conn.	S. W. Dudley

A.S.M.E. Representatives on Joint Activities

Dates in parentheses denote expiration of term

AMERICAN ASSOCIATION FOR THE ADVANCEMENT OF SCIENCE

SECTION M, ENGINEERING

R. L. Daugherty — Wm. B. Gregory

Representatives or alternates appointed according to geographical location of meetings

AMERICAN ENGINEERING COUNCIL

Terms expire January 1, 1932

L. P. Alford, New York, N.Y.
Thomas D. Campbell, Hardin, Mont.
William B. Ferguson, Newport News, Va.
Charles E. Ferris, Knoxville, Tenn.
John Lyle Harrington, Kansas City, Mo.
William H. Kenerson, Providence, R.I.
John H. Lawrence, New York, N.Y.
Richard C. Marshall, Jr., Chicago, Ill.
A. A. Potter, Lafayette, Ind.
Max Toltz, St. Paul, Minn.

Terms expire January 1, 1933

Roy V. Wright, *Chairman*, New York, N.Y.
W. S. Conant, New York, N.Y.
E. O. Eastwood, Seattle, Wash.
B. E. Hull, Houston, Tex.
Jos. W. Roe, New York, N.Y.
Warner Seely, Cleveland, Ohio.
W. Trinks, Pittsburgh, Pa.
E. N. Trump, Syracuse, N.Y.
W. R. Webster, New York, N.Y.
D. Robert Yarnall, Philadelphia, Pa.

AMERICAN SOCIETY OF SAFETY ENGINEERS

ENGINEERING SECTION, NATIONAL SAFETY COUNCIL

H. W. Mowery — Ernest Hartford

AMERICAN STANDARDS ASSOCIATION

C. P. Bliss (1931)[4] — Cloyd M. Chapman (1933)
Lewis K. Sillcox — K. H. Condit (*alternate*)
C. B. Le Page (*alternate*)

EDUCATIONAL RESEARCH COMMITTEE

THE ENGINEERING FOUNDATION

Harvey N. Davis

THE ENGINEERING FOUNDATION

H. Hobart Porter (1932)[5] — A. E. White (1933)
D. Robert Yarnall (1934)

GANTT MEDAL BOARD OF AWARD

W. L. Conrad — Conrad N. Lauer
Dexter S. Kimball

GUGGENHEIM MEDAL BOARD OF AWARD

H. I. Cone (1931) — E. E. Aldrin (1933)
W. F. Durand (1932) — George W. Lewis (1934)

HERBERT HOOVER MEDAL BOARD OF AWARD

Ambrose Swasey — Dexter S. Kimball
Calvin W. Rice

INSTITUTE OF HUMAN RELATIONS

ADVISORY COMMITTEE

H. M. Crane — H. E. Howe
Ralph E. Flanders — J. P. Jordon
E. W. Rice, Jr. — Calvin W. Rice, *Ex-Officio*

INTERNATIONAL ELECTROTECHNICAL COMMISSION

U. S. NATIONAL COMMITTEE

C. H. Berry — F. Hodgkinson
H. N. Davis — E. C. Hutchinson
P. Diserens — F. R. Low
W. F. Durand — I. E. Moultrop

[4] Represents the A.S.M.E. on the Executive Committee of the A.S.A.
[5] A.S.M.E. representatives from Board of United Engineering Trustees.

INTERNATIONAL MANAGEMENT CONGRESS COMMITTEE OF THE U. S. A.

L. P. Alford — Calvin W. Rice

JOHN FRITZ MEDAL BOARD OF AWARD

The terms of office expire in October of each year

Charles M. Schwab (1931) — W. L. Abbott (1933)
Alex Dow (1932) — Charles Piez (1934)

JOSEPH A. HOLMES SAFETY ASSOCIATION

O. P. Hood

LIBRARY BOARD

MONOGRAPHS ADVISORY COMMITTEE

W. A. Shoudy — S. W. Dudley

NATIONAL RESEARCH COUNCIL

DIVISION OF ENGINEERING

R. J. S. Pigott (June, 1931) — A. E. White (June, 1932)
R. L. Streeter (June, 1933)

ALFRED NOBLE PRIZE

L. P. Alford

THE PRESIDENT'S EMERGENCY COMMITTEE ON EMPLOYMENT

A.S.M.E. COOPERATING COMMITTEE

Ralph E. Flanders, *Chairman* — Ely C. Hutchinson
Dexter S. Kimball — C. H. Crawford
J. W. Roe — W. A. Starrett
Ex-Officio: Roy V. Wright, President, A.S.M.E.
Calvin W. Rice, Secretary, A.S.M.E.

REGISTRATION OF ENGINEERS

D. S. Kimball, *Chairman* — C. F. Hirshfeld
Virgil M. Palmer, *Vice-Chairman* — J. M. Todd
Paul Doty — Calvin W. Rice (*alternate*)

SPIRIT OF ST. LOUIS MEDAL BOARD OF AWARD

P. deC. Ball — H. I. Cone
W. F. Durand — J. L. Doolittle
W. B. Mayo — Orville Wright

UNITED ENGINEERING TRUSTEES, INC.

H. Hobart Porter (1932) — H. V. Coes (1933)
Edwards R. Fish (1934)

WESTERN SOCIETY OF ENGINEERS

WASHINGTON AWARD

James D. Cunningham (June, 1931) — Julius L. Hecht (1932)

GENERAL ASSIGNMENTS[6]

American Bureau of Welding, Advisory Board of the National Research Council, James Partington.
American Bureau of Welding, Advisory Committee on Structural Steel Welding, Geo. A. Orrok.
American Bureau of Welding, Gas Welding Committee, James Partington and C. W. Obert.
International High Commission, Advisory Committee to the U. S. Section, concerned with a preliminary study and compilation of a report on the use of Uniform Weights and Measures, Ralph E. Flanders.
National Screw Thread Commission, Luther D. Burlingame.
Standardization Survey Committee, C. M. Schwab, H. Birchard Taylor (alternate).
U. S. Department of Commerce, National Committee on Metals Utilization, Stanley G. Flagg, Jr.
U. S. Department of Commerce, National Conference on Street and Highway Safety, E. J. Posselt.
U. S. Shipping Board, Fuel Conservation Board, H. L. Seward.

[6] The Society is also represented on a number of technical committees sponsored by other organizations.

Exchange of Courtesies

Special arrangements have been made between the A.S.M.E. and the following organizations whereby their members will be mutually accorded such courtesies as the use of reading and writing rooms, assistance in securing general information and in arranging to visit industrial plants, receipt of mail, etc.

Austria	
Vienna	Osterreichischer Ingenieur und Architekten Verein
Belgium	
Bruxelles	Federation des Associations Belges d'Ingenieurs
	Société Belge des Ingenieurs et des Industriels
Canada	
Montreal	The Engineering Institute of Canada
Cuba	
Havana	Sociedad Cubana de Ingenieros
Czechoslovakia	
Prague	Association of Czechoslovakian Engineers and Architects in Praha
	Masarykova Akademie Prace
	Spolek Ceskoslovenskych Inzenyru
Denmark	
Copenhagen	Dansk Ingeniorforening
England	
London	British Association for the Advancement of Science
	British Engineering Standards Association
	British Engineers Association
	Chemical Society
	The Institute of Fuel
	The Institute of Marine Engineers
	The Institute of Metals
	The Institution of Automobile Engineers
	The Institution of Civil Engineers
	The Institution of Electrical Engineers
	The Institution of Engineering Inspection
	The Institution of Mechanical Engineers
	The Institution of Mining Engineers
	Institution of Naval Architects
	The Institution of Structural Engineers
	International Electrotechnical Commission
	The Iron and Steel Institute
	Junior Institution of Engineers
	The Royal Aeronautical Society
	The Society of Engineers
Newcastle-on-Tyne	North East Coast Institution of Engineers and Shipbuilders
	North of England Institute of Mining and Mechanical Engineers
South Kensington	Newcomen Society
France	
Paris	l'Association Technique Maritime et Aeronautique
	Société des Ingènieurs Civils de France
	Société Francaise des Electriciens
Germany	
Berlin	Verein deutscher Ingenieure
Italy	
Rome	Sindacato Nazionale Fascista delgi Ingegneri
Japan	
Tokyo	Kogakkai
Mexico	
Mexico City	Asociacion de Ingenieros y Arquitectos de Mexico
Netherlands	
The Hague	Koninklijk Instituut van Ingenieurs

Norway
Oslo.....Den Norske Ingeniorforening

Scotland
Glasgow.....Institution of Engineers and Shipbuilders in Scotland

South Africa
Johannesburg.....South African Institution of Engineers

South America
Colombia.....Sociedad Colombiana de Ingenieros

Sweden
Stockholm.....Svenska Teknologforeningen

Switzerland
Winterthur.....Verein Schweizerischer Maschinen-Industrieller

UNITED STATES OF AMERICA

California
Los Angeles.....Engineering and Architects Association

District of Columbia
Washington.....Society of American Military Engineers

Florida
Gainesville.....Florida Engineering Society

Illinois
Chicago.....Western Society of Engineers

Louisiana
New Orleans.....Louisiana Engineering Society

Maryland
Baltimore.....Baltimore Engineers Club

Massachusetts
Boston.....Boston Society of Civil Engineers

Michigan
Detroit.....Detroit Engineering Society

Missouri
St. Louis.....Engineers Club of St. Louis

New York
New York.....American Gas Association
American Institute of Electrical Engineers
American Institute of Mining and Metallurgical Engineers
American Society of Civil Engineers
American Society of Refrigerating Engineers
American Society of Safety Engineers
Chemists Club Library
Columbia University Library
New York Railroad Club
Rochester.....Rochester Engineering Society
Syracuse.....Technology Club of Syracuse
Utica.....Mohawk Valley Engineers' Club

Ohio
Cleveland.....Cleveland Engineering Society
Dayton.....Engineers' Club of Dayton

Pennsylvania
Philadelphia.....Engineers Club of Philadelphia
Pittsburgh.....Engineers Society of Western Pennsylvania
Scranton.....Engineers Society of Northeastern Pennsylvania
Scranton Engineers Club

Rhode Island
Providence.....Providence Engineering Society

Tennessee
Nashville.....Engineering Association of Nashville

Washington
Seattle.....Pacific Northwest Society of Engineers
Spokane.....Associated Engineers of Spokane

Awards

The Committee on Awards, given on page 9, has supervision of the awards of the Society under the direction of the Council. Awards and special funds are administered as specified in the deeds of gift or as may be determined by the Council from time to time. The following awards come within the jurisdiction of the Society. A list of the recipients of awards previous to 1931 will be found in the 1930 Record and Index, pages 18–20.

Honorary Membership, to which persons of acknowledged professional eminence are elected by unanimous vote of Council under the provisions of the By-Laws and Rules. A list of honorary members is given on page 15.

Life Membership, which may be conferred by the Council for distinguished service to the Society.

A.S.M.E. Medal, established by the Society in 1920 to be presented for distinguished service in engineering and science. May be awarded for general service in science having possible application in engineering. Award for 1931 to Albert Kingsbury for his research and development work in the field of lubrication.

Holley Medal, instituted and endowed in 1924 by George I. Rockwood, Past Vice-President of the Society, to be bestowed for some great and unique act of genius of engineering nature that has accomplished a great and timely public benefit.

Melville Medal, established in 1914 by the bequest of Rear-Admiral George W. Melville, Honorary Member and Past-President of the Society, to be presented for an original paper or thesis of exceptional merit, presented to the Society for discussion and publication, to encourage excellence in papers. May be presented annually. Award for 1931 to A. E. Grunert for his paper entitled "Comparative Performance of a Pulverized-Coal-Fired Boiler Using Bin System and Unit System of Firing."

Spirit of Saint Louis Medal, endowed by members of the Society and citizens residing in St. Louis, Mo., to be awarded for meritorious service in the field of aeronautical engineering. This medal will be awarded at the discretion of the Council of the Society at approximately three-year periods upon the recommendation of a Spirit of Saint Louis Medal Board of Award made up of six members, each appointed for a term of nine years and the terms of two members expiring at each three-year period. The St. Louis Section and the Aeronautic Division will each nominate three members. The first medal was awarded to Daniel Guggenheim in 1929.

Worcester Reed Warner Medal, provision for which was made in the will of Worcester Reed Warner, Honorary Member of the Society, is to be a gold medal to be bestowed on the author of the most worthy paper received, dealing with progressive ideas in mechanical engineering or efficiency in management.

Junior Award, annual cash award of $50, established in 1914, from a fund created by Henry Hess, Past Vice-President of the Society, to be presented, together with an engraved certificate, for the best paper or thesis submitted by a Junior Member. Award for 1931 to M. K. Drewry for his paper entitled "Radiant-Superheater Developments."

Student Awards, two annual cash awards of $25 each, established in 1914, from a fund created by Henry Hess, Past Vice-President of the Society, to be presented, together with engraved certificates, for the best papers or theses submitted by Student Branch Members. One award for 1931 to Jules Podnossoff of the Polytechnic Institute of Brooklyn for his paper entitled "Pressure and Energy Distribution in Multi-Stage Steam Turbines Operating Under Varying Conditions."

Charles T. Main Award, annual cash award of $150, established in 1919 from a fund created by Charles T. Main, Past-President of the Society, to be awarded to a student of engineering, preferably a member of a Student Branch of the Society, for the best paper within the general subject of the "Influence of the Profession upon Public Life." The exact subject is assigned by the Committee on Awards, subject to the approval of the Council, and is announced each year through the Honorary Chairmen of the Student Branches. Award for 1931 to Robert E. Klise of the University of Michigan for his paper on "Interchangeability—Its Development and Significance in Industry."

SCHOLARSHIPS AND LOAN FUNDS

Max Toltz: Loan Fund of $15,000 established by Major Max Toltz, former member of the Council of the Society, the income to be used for assistance to students.

John R. Freeman: Fund of $25,000 established in 1926 by John R. Freeman, Past-President of the Society, the income to be used for travel scholarships and research. Scholarships have been awarded as follows: Herbert N. Eaton and Blake R. Van Leer, 1927; Robert T. Knapp, 1929; and Reginald Whitaker, 1931.

Woman's Auxiliary: Scholarship or Fellowship offered by the Woman's Auxiliary to the Society to assist sons and daughters of members or worthy students of mechanical engineering.

JOINT AWARDS

The A.S.M.E. participates in three joint awards, the Herbert Hoover Medal, the John Fritz Medal, and the Washington Award.

The Herbert Hoover Medal

The Hoover Gold Medal was instituted in 1930 to commemorate the civic and humanitarian achievements of Herbert Hoover and the first award was made to Herbert Hoover on April 8, 1930. From time to time the medal will be awarded by engineers to a fellow engineer for distinguished public service.

The trust fund creating the award was the gift of Conrad N. Lauer, of Philadelphia. The American Society of Mechanical Engineers is the custodian of the fund, which is administered by a Board of Award composed of representatives appointed by the American Society of Civil Engineers, the American Institute of Mining and Metallurgical Engineers, The American Society of Mechanical Engineers, and the American Institute of Electrical Engineers.

The John Fritz Medal

The John Fritz Medal was established in August, 1902, by the professional associates and friends of the late John Fritz, Past-President and Honorary Member of the A.S.M.E., to perpetuate the memory of his achievement in industrial progress.

The medal is awarded not more than once each year for notable scientific or industrial achievement, with no restrictions on account of sex or nationality. The award is made by a Board of sixteen, four representatives from each of the four national engineering societies, the A.S.C.E., A.I.M.E., A.S.M.E., and A.I.E.E.

The Washington Award

The Washington Award was founded in 1916 by John Watson Alvord, to be awarded annually by the Western Society of Engineers upon the recommendation of a Commission composed of nine representatives of that society and two representatives of each of the four national engineering societies, the A.S.C.E., A.I.M.E., A.S.M.E., and A.I.E.E. It is bestowed in recognition of devoted, unselfish, and preeminent service in advancing human progress through engineering.

The award for 1931 was made to Ralph Modjeski for his contribution to transportation through superior skill and courage in bridge design and construction.

Honorary Members

HONORARY MEMBERS IN PERPETUITY

Alexander Lyman Holley, Founder of the Society. Died 1882.

John Edson Sweet, Founder of the Society. Died 1916.

Henry Rossiter Worthington, Founder of the Society. Died 1880.

DECEASED HONORARY MEMBERS

	ELECTED	DIED
Horatio Allen	1880	1889
Sir William Arrol	1905	1913
Sir Benjamin Baker	1886	1907
Johann Bauschinger	1884	1893
Sir Henry Bessemer	1891	1898
Sir Frederick Joseph Bramwell	1884	1903
John Alfred Brashear	1908	1920
Gustave Canet	1900	1908
Andrew Carnegie	1907	1919
Daniel Kinnear Clark	1882	1896
Rudolph Julius Emmanuel Clausius	1882	1888
Sir John Coode	1889	1892
Peter Cooper	1882	1883
Carl Gustaf Patrik de Laval	1912	1913
Rudolph Diesel	1912	1913
James Dredge	1886	1906
Victor Dwelshauvers-Dery	1886	1913
Thomas Alva Edison	1904	1931
Alexandre Gustave Eiffel	1889	1923
Marshal Ferdinand Foch	1921	1929
Sir Charles Douglas Fox	1900	1921
John Fritz	1900	1913
Major - General George Washington Goethals	1917	1928
Franz Grashof	1884	1893
Otto Hallauer	1882	1883
Charles Haynes Haswell	1905	1907
Friedrich Gustav Herrmann	1884	1907
Gustav Adolph Hirn	1882	1890
Joseph Hirsch	1889	1901
Ira N. Hollis	1928	1930
Robert Woolston Hunt	1920	1923
Benjamin Franklin Isherwood	1894	1915
Henri Léauté	1891	1916
Erasmus Darwin Leavitt	1915	1916
Anatole Mallet	1912	1919
Charles H. Manning	1913	1919
Rear-Admiral George Wallace Melville	1910	1912
The Honorable Sir Charles Algernon Parsons	1920	1931
Charles Talbot Porter	1890	1910
Auguste C. E. Rateau	1919	1930
Sir Edward J. Reed	1882	1906
Franz Reuleaux	1882	1905
Henri Adolphe - Eugene Schneider	1882	1898
C. William Siemans	1882	1883
Viscount Ei-ichi Shibusawa	1929	1931
Henry Robinson Towne	1921	1924
Henri Tresca	1882	1885
Francis A. Walker	1886	1897
Worcester Reed Warner	1925	1929
Sir William Henry White	1900	1913
George Westinghouse	1897	1914

LIVING HONORARY MEMBERS

	ELECTED
Sir John Audley Frederick Aspinall	1911
William Wallace Atterbury	1925
Mortimer Elwyn Cooley	1928
Charles de Fréminville	1919
Rear-Admiral Robert Stanislaus Griffin	1920
Nathanael Greene Herreshoff	1921
Honorable Herbert Clark Hoover	1925
Masawo Kamo	1929
Henri Le Chatelier	1927
Grande Ufficiale Ing. Pio Perrone	1920
Calvin Winslow Rice	1931
Palmer C. Ricketts	1931
Charles M. Schwab	1918
Ambrose Swasey	1916
Elihu Thomson	1930
William Cawthorne Unwin	1898
Samuel Matthews Vauclain	1920
Oskar von Miller	1912
Right Honorable Lord Weir	1920
Orville Wright	1918
Sir Alfred Fernandez Yarrow	1914

Past Officers[7]

ALEXANDER LYMAN HOLLEY, Chairman of the Preliminary Meeting for Organization of The American Society of Mechanical Engineers.
Died January 29, 1882

PRESIDENTS

ROBERT HENRY THURSTON Died October 25, 1903	1880–1882
ERASMUS DARWIN LEAVITT Died March 11, 1916	1883
JOHN EDSON SWEET Died May 8, 1916	1884
JOSEPHUS FLAVIUS HOLLOWAY Died September 1, 1896	1885
COLEMAN SELLERS Died December 28, 1907	1886
GEORGE H. BABCOCK Died December 16, 1893	1887
HORACE SEE Died December 14, 1909	1888
HENRY ROBINSON TOWNE Died October 15, 1924	1889
OBERLIN SMITH Died July 19, 1926	1890
ROBERT WOOLSTON HUNT Died July 11, 1923	1891
CHARLES HARDING LORING Died February 5, 1907	1892
ECKLEY BRIXTON COXE Died May 13, 1895	1893–1894
EDWARD F. C. DAVIS Died August 6, 1895	1895
CHARLES ETHAN BILLINGS Died June 4, 1920	1895
JOHN FRITZ Died February 13, 1913	1896
WORCESTER REED WARNER Died June 25, 1929	1897
CHARLES WALLACE HUNT Died March 27, 1911	1898
GEORGE WALLACE MELVILLE Died March 17, 1912	1899
CHARLES HILL MORGAN Died January 10, 1911	1900
SAMUEL T. WELLMAN Died July 11, 1919	1901
EDWIN REYNOLDS Died February 19, 1909	1902
JAMES MAPES DODGE Died December 4, 1915	1903
AMBROSE SWASEY Cleveland, Ohio	1904
JOHN RIPLEY FREEMAN Providence, R. I.	1905
FREDERICK WINSLOW TAYLOR Died March 21, 1915	1906
FREDERICK REMSEN HUTTON Died May 14, 1918	1907
MINARD LAFEVER HOLMAN Died January 4, 1925	1908
JESSE MERRICK SMITH Died April 1, 1927	1909
GEORGE WESTINGHOUSE Died March 12, 1914	1910 1911
EDWARD DANIEL MEIER Died December 15, 1914	
ALEXANDER CROMBIE HUMPHREYS Died August 14, 1927	1912
WILLIAM FREEMAN MYRICK GOSS Died March 23, 1928	1913
JAMES HARTNESS Springfield, Vt.	1914
JOHN ALFRED BRASHEAR Died April 8, 1920	1915
DAVID SCHENCK JACOBUS New York, N. Y.	1916
IRA NELSON HOLLIS Died August 15, 1930	1917
CHARLES THOMAS MAIN Boston, Mass.	1918
MORTIMER ELWYN COOLEY Ann Arbor, Mich.	1919
FRED J. MILLER Center Bridge, Pa.	1920
EDWIN S. CARMAN Cleveland, Ohio	1921
DEXTER SIMPSON KIMBALL Ithaca, N. Y.	1922
JOHN LYLE HARRINGTON Kansas City, Mo.	1923
FREDERICK ROLLINS LOW New York, N. Y.	1924
WILLIAM FREDERICK DURAND Stanford Univ., Calif.	1925
WILLIAM LAMONT ABBOTT Chicago, Ill.	1926
CHARLES M. SCHWAB New York, N. Y.	1927
ALEX DOW Detroit, Mich.	1928
ELMER AMBROSE SPERRY Died June 16, 1930	1929
CHARLES PIEZ Chicago, Ill.	1930
ROY V. WRIGHT New York, N. Y.	1931

NOTE: According to the Constitution, Article C7, Sec. 2, the last five surviving Past-Presidents are members of the Council.

[7] A list of past vice-presidents, managers, treasurers, and secretaries will be found in the 1930 Record and Index, pages 10–12.

Report of Council

THE Society made significant progress during the year 1930–1931 as a result of the cumulative efforts of many individuals and active groups organized in committees. This progress, which covered a wide range of professional engineering achievement, was carried on during a period of industrial and economic stress which made the work of the individual members of the Society much more difficult and the results of it therefore much more praiseworthy.

A noteworthy feature of the year's development was the untiring and enthusiastic attitude of the President of the Society, Dr. Roy V. Wright, who during his term of office visited every one of the 71 Sections and 66 of the 108 Student Branches. Dr. Wright aroused the members of the Society to the need for more thorough understanding by them of the economic and social implications of the work of the engineer. The net result was the quickening of interest in the problem which will undoubtedly result in broadening and deepening the professional life of the Society. Dr. Wright is entitled to the sincere gratitude of every member.

Another notable factor was the opportunity provided for members to participate in Society activities. Four meetings of the Society outside of New York City, 17 national meetings of Professional Divisions, 471 meetings of Local Sections, and 143 technical committee meetings mark a high point in the number of gatherings. This policy of providing more opportunities for members throughout the nation to take part in Society work is a logic corollary of President Wright's recent campaign to arouse interest in that work.

A unique achievement of the technical committees was the survey of screw-thread practice in the United States, sponsored by the S.A.E. and the A.S.M.E. with the National Bureau of Standards cooperating. Over 163 firms manufacturing and using threaded products were visited by the Committee's representative and approximately 7600 bolts, nuts, and screws were collected and shipped to the National Bureau of Standards for measurement in an effort to judge the extent to which the American National Standard Screw Thread has measured up to commercial requirements and in what respect revisions should now be made.

An important program for future progress has grown out of the Report of the Committee on the Economic Status of the Engineer, a fact-finding study which appeared in the September issue of *Mechanical Engineering* and which has provided the basis for developing the professional position of the engineer.

The problem of affiliating the student members more closely with the work of the Society was brought nearer to a satisfactory solution during the year. An auspicious beginning was made at the French Lick Meeting shortly after the opening of the fiscal year where an unusually successful student session was held. The year closed with the initiation of a program for the organization of Student Branches as fundamental units of the Society. The plan that has been decided upon offers great promise for the future development of the engineering profession.

The Pure Air Committee, organized to focus the influence of the engineering profession on reducing smoke, dust, and fumes, was a logical and important addition to the list of similar committees that are devoted to the public interest.

The accompanying table shows the changes in membership during the year, resulting in a total of 20,040 as of November, 1931. The Society mourns the loss from the distinguished group of Honorary Members of Thomas Alva Edison, Member of the Society since 1880, Honorary Member since 1904; the Honorable Sir Charles Algernon Parsons; and Viscount Ei-ichi Shibusawa.

Meetings of the Council were held during the Annual Meeting in December, 1930, at the Semi-Annual Meeting, Birmingham, in April, 1931, the Hartford Meeting in June, and by special representation at the Kansas City Meeting in September. The Executive Committee of the Council has met more frequently and has attended to urgent and detailed Council business.

Presidential appointments and special representations, outside of the regular routine appointments for the year, have totaled more than 100, including those for the Seventy-Fifth Anniversary of the Verein deutscher Ingenieure, attended by a group of about 70 from this country, the Faraday Centenary in London, and several century and half-century anniversaries of foreign and American Colleges and societies, as well as International Congresses and similar celebrations.

A gift has come to the Society from the bequest of C. B. F. Waller of Philadelphia, Member of the Society.

Coordinating the public and joint activities of its 26 engineering member societies, the American Engineering Council has done some notable work in the programs for the relief of unemployment, both through the President's Committee and the A.E.C. plan to be carried out throughout the United States by engineering societies and their local sections. A report on technological unemployment submitted to the Department of Labor by a special committee of the societies through the A.E.C. has been favorably received. Bulletins of the A.E.C. record its most important work, in which the Society participates.

The reports that follow are worthy of careful study.

CALVIN W. RICE, *Secretary*.

MEMBERSHIP CHANGES—OCTOBER 1, 1930, TO SEPTEMBER 30, 1931

	Membership		Losses				Additions			Totals		Net Changes
	Oct. 1, 1930	Oct. 1, 1931	Transferred from	Resigned	Dropped	Died	Transferred to	Elected	Reinstated	Loss	Gain	
Honorary Members	24	25	...	...	...	1	1	1	..	1	2	1+
Life Members	74	70	...	...	5	1	...	...	..	5	1	4−
Members	8955	8968	...	142	131	135	137	276	8	408	421	13+
Associates	632	602	6	25	15	6	1	20	1	52	22	30−
Associate-Members	4175	4127	98	82	176	18	82	234	10	374	326	48−
Juniors (20)	1558	1668	74	112	147	1	442	...	2	334	444	110+
Juniors (10)	4593	4549	486	88	243	11	...	779	5	828	784	44−
	20,011	20,009	664	449	712	177	664	1310	26	2002	2000	2−

FINANCE

The year 1930–1931 brought a number of problems in the financing of the Society. The reduction in income from advertising necessitated reductions in expenditures as compared to the previous year; but without serious curtailment of Society activities. By the provision of the By-Laws, the income from initiation fees is not included in the estimate of income when the budget of expenditures is prepared and is placed in a special reserve. The expenditures ex-

ceeded the other income of the Society by $9896.58, which the Council authorized to be taken from accumulated surplus reserve.

While the earning power of our invested securities has not declined, the market value has declined somewhat, making it prudent to ear-mark a portion of our reserves as Reserve against Depreciation of Securities; $15,000 has been set aside for this purpose, merely in the event of an emergency that might require the sale of some securities in a depreciated market.

The Society has built up a reputation for paying bills and has found it to advantage, toward the end of the year, to borrow for short periods rather than to sell investments. This financing is necessary as the Mechanical Catalog billings are not made until September 30, the last day of the fiscal year, while expenditures against that income are incurred throughout the year. The amount borrowed this year was $170,000, an increase of $50,000 over the previous year, necessary to pay bills outstanding at the close of the fiscal period and because of an increase in needs for cash to operate The Engineering Index.

The Engineering Index passed through its fourth year and made an excellent showing in the face of the present situation. The income was $108,508.64, the expenses $113,379.71, showing an operating deficit of $4871.07. Means have been found for producing the Index more economically, and it is expected that during the coming fiscal year the Index will begin to repay the advances that have been made for its development.

Total Income

The following table gives a comparison of the total income of the Society for the fiscal year ending September 30, 1930:

	1929–1930	1930–1931
Initiation Fees	$ 30,430.17	$ 25,863.27
Membership Dues	314,815.87	314,828.31
Advertising and Publication Sales (less bad debts)	439,704.08	377,132.72
Interest and Discount	23,601.18	22.080.10
Miscellaneous Sales	5,529.10	3,364.35
	$814,080.40	$743,268.75

The 1929–1930 fiscal year was exceptional because of the large income from the special Fiftieth Anniversary issue of *Mechanical Engineering*.

Expenditures

During the 1930–1931 year a simplified scheme of accounting was installed, which provided that for control purposes the expenditures of the Society should be segregated into three major classes: Committee Expenses, Printing and Distributing Publications, and Expenses of the Administrative Office.

Committee Expenses are those which are incurred by direct action of the committees or Council. The Administrative Office Expenses are those incurred in the office of the Secretary to carry out the work of the various committees and to execute policies approved by the Council. The expenditures according to this classification are as follows:

	1929–1930	1930–1931
Committee Expenses	$114,380.01	$116,073.62
Reserve for Engineering Index Development	5,000.00	.....
Publications (Printing and Distribution only)	242,847.38	230,932.77
Administrative Office	412,804.34	380,295.57
	$775,031.73	$727,301.96
To Reserve	39,048.67	15,966.79
	$814,080.40	$743,268.75

The use to which these expenditures are put in carrying on the work of the Society is shown in the following tabulation:

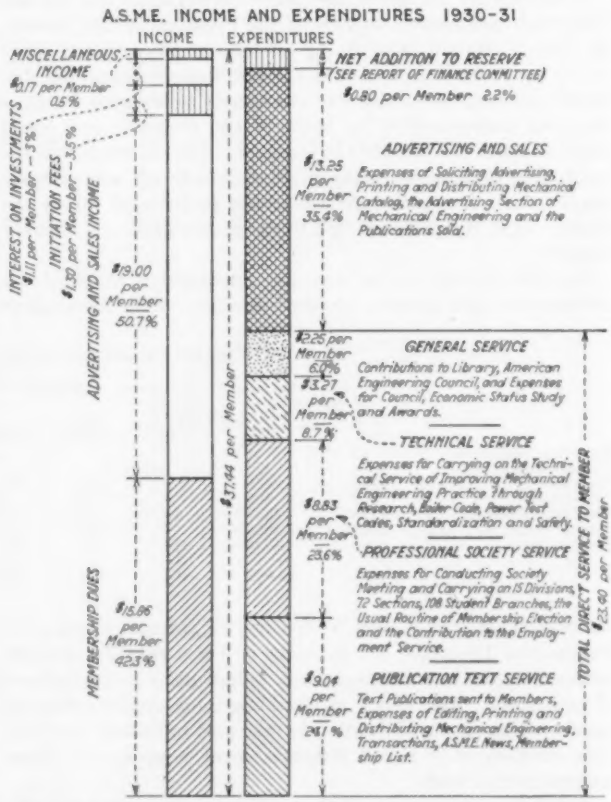
	1929–1930	1930–1931
General	$ 41,754.43	$ 44,630.18
Society Activities	171,470.63	175,241.50
Technical Committees	66,475.18	64,883.97
Text Publications (Printing, Editorial, and Distribution)	188,232.12	179,390.56
Advertising and Sales (Printing, Soliciting, and Distribution)	302,099.37	263,155.75
Reserve for Engineering Index Development	5,000.00	.....
	$775,031.73	$727,301.96

General Expenses include those for Council, the Library, American Engineering Council, the study of the Committee on the Economic Status of the Engineer, Awards, and miscellaneous. The increase of nearly $3000 was used by the Council (including the traveling expenses of the President) and in the study of the Earnings of Mechanical Engineers under the auspices of the Committee on the Economic Status of the Engineer.

Expenses for Society Activities include those for Meetings, Local Sections, Professional Divisions, Student Branches, Employment Service, and for the staff to support these activities and to carry on the routine of securing and handling membership applications for the Membership Committee. The increase of $3800 was due to an increase of $1000 in direct expenditure by the Professional Divisions and to an increase in staff support to Sections and Divisions in the Mid-Continent area.

Technical Committee expenses include the direct expenses to the International Electrotechnical Commission ($1320), American Standards Association ($1500), grants in aid of research ($1286.55), and staff expenses to support the Boiler Code, Research, Standardization, Power Test Codes, and Safety Committees. The decrease from the previous year is in grants for research and in the staff.

Text publications includes the expenses for printing, distributing, and editing Transactions, the *A.S.M.E. News*, Membership List, and the text pages of *Mechanical Engineering*. In 1929–1930 the



Note: The average membership during 1930–1931 was 19,878. The income from membership dues is made up of income from about 14,300 members at $20 each and from 4400 Juniors at $10 each. First year Juniors who pay partial dues, honorary members, life members, etc., make up the remainder of the membership

April issue of *Mechanical Engineering*, devoted to the Fiftieth Anniversary, entailed unusual expenses, which were provided by increased advertising income. The decrease in 1930–1931 is due to the absence of these special expenditures and to fewer pages in Transactions.

Expenses of Advertising and Sales include (1) printing and distributing the Mechanical Catalog and the advertising pages of *Mechanical Engineering*, (2) cost of securing advertising income, (3) printing and distributing publications sold, and (4) the expenses of the publication sales department. The decrease in this item is due to the reduced number of advertising pages sold and printed.

Income and Expenses per Member

The chart above shows the income and expenses for 1930–1931 reduced to a per member basis.

Balance Sheet

The accompanying comparative balance sheet explains the status of our liabilities and assets and shows the changes that have taken place during the fiscal year.

F. A. Schaff, *Chairman* J. H. Lawrence
David Lofts E. A. Muller.
K. L. Martin

A.S.M.E. COMPARATIVE BALANCE SHEET

ASSETS

	September 30, 1930		September 30, 1931	
CURRENT ASSETS				
Cash in Bank and on Hand	$ 6,965.99		$ 12,674.63	
Accounts Receivable	166,150.97		186,905.04	
(Money due the Society from members for dues, from advertisers in Mechanical Catalog and Mechanical Engineering, from sale of Engineering Index Service and from sale of Publications. Increase of $20,754.07 due to inclusion of Engineering Index item.)				
Inventories	114,873.36		140,239.56	
(Supplies at cost, publications in process of manufacture and completed publications in the stock room for sale. Increase of $25,366.20 due to drop in expected sale of index and large orders completed just before close of the fiscal period.)				
TOTAL CURRENT ASSETS		287,990.32		339,819.23
INVESTMENTS				
Trust Funds	164,035.45		166,764.94	
(Funds entrusted to the Society for specific purposes. Net increase of $2,729.49 due to accrued interest and addition of Waller Fund less expenditures of $2000 for Hoover Medal.)				
Invested Reserve	343,381.25		343,381.25	
(Securities purchased from proceeds of initiation fees and excess of income over expenditures in previous years.)				
Advances				
Physical Properties of Steam	15,000.00		15,000.00	
(A loan at interest to the Special Research Committee in Charge.)				
Engineering Index Service	110,981.45		112,916.80	
(A loan at interest for the establishment of the Engineering Index Service. Increase of $1,935.35 due to operating loss of $4,871.07 less depreciation paid of $2,935.72.)				
TOTAL INVESTMENTS		633,398.15		638,062.99
FIXED ASSETS				
United Engineering Trustees, Inc.	493,352.60		496,948.48	
(One-quarter interest in building. Depreciation is carried on the books of U.E.T., Inc. Increase of $3,585.88 on basis of changed value ordered by U.E.T., Inc.)				
Furniture and Equipment	22,327.74		24,094.93	
Library Books	1.00		1.00	
Engineering Index	1.00		1.00	
TOTAL FIXED ASSETS		515,682.34		521,045.41
DEFERRED CHARGES				
50th Anniversary	29,399.66		24,399.66	
(Advanced in 1930, being repaid at rate of $5000 per year.)				
Society Activities	15,820.03		3,001.41	
(Money paid in advance for activities provided in budget of next fiscal year.)				
TOTAL DEFERRED CHARGES		45,219.69		27,401.07
TOTAL ASSETS		$1,482,290.50		$1,526,328.73

LIABILITIES

	September 30, 1930		September 30, 1931	
CURRENT LIABILITIES				
Accounts Payable	$ 20,405.85		$ 0.00	
(Amounts billed to Society and due. Reduction of $20,405.85 possible by larger borrowing.)				
Unfilled Commitments	43,937.28		32,391.15	
(Commitments for printing and distributing Mechanical Catalog. Bills not due but income from Catalog is credited during current fiscal period. Reduction of $11,546.13 due to decreased cost of Catalog.)				
Gleason Gift for Thurston Biography	460.82		460.82	
Notes Payable	120,000.00		170,000.00	
(Borrowing from bank for current financing. Increase of $50,000 needed for accounts payable and for current financing on Engineering Index.)				
TOTAL CURRENT LIABILITIES		184,803.95		202,851.97
CUSTODIAN FUNDS		23,757.09		27,715.93
(Funds in the custody of the Society to be expended for research at the direction of some thirty Committees which collected the funds.)				
DUES PAID IN ADVANCE		4,349.81		3,572.36
TRUST FUNDS		164,035.45		166,764.94
RESERVES				
Staff Retirement Reserve Fund	26,232.32		26,232.32	
Initiation Fee Reserve Fund	169,288.78		195,152.05	
(By-Laws require that initiation fees be placed in reserve fund. Increase of $25,863.27 is amount of initiation fees received in 1930–31.)				
Previous Surplus Invested	147,860.15		106,996.88	
(Reduction of $40,863.27, $25,863.27 of which was to provide additional working capital during 1930–1931 and $15,000 of which was transferred to establish a Reserve against Depreciation of Securities.)				
Engineering Index Development Reserve	25,000.00		25,000.00	
Reserve Fund Against Depreciation of Securities	0.00		15,000.00	
TOTAL RESERVES		368,381.25		368,381.25
CAPITAL				
In Fixed Assets	515,682.34		521,045.41	
Working Capital	221,280.61		235,996.87	
TOTAL CAPITAL		736,962.95		757,042.28
TOTAL LIABILITIES		$1,482,290.50		$1,526,328.73

MEETINGS AND PROGRAM

Five meetings were held during the year under the direct supervision of the Committee on Meetings and Program, as follows: French Lick, Ind., October 13–15, 1930, Annual Meeting in New York, December 1–5, 1930, Semi-Annual Meeting in Birmingham, Ala., April 20–23, 1931, Hartford, Conn., June 1–3, 1931, and Kansas City, Mo., Sept. 7–9, 1931.

French Lick Meeting

Excellent technical sessions and an attractive outdoor program made this meeting an enjoyable occasion. A Student Session proved to be one of the most interesting of the technical events; papers were presented by students representative of six universities on such varied topics as power industry, cooperative education, research in mechanical engineering, machine tools, mechanical engineering in the coal industry, heat treatment of nickel steel welds, and instruction in flying to engineering students. Dr. C. E. Grunsky, President of the American Engineering Council, delivered an inspiring message at a luncheon for the students and Council members, and addresses of interest were also given at the informal dinner and banquet.

Annual Meeting

The Fifty-First Annual Meeting broke all records for attendance—2562, which exceeded the attendance of the previous meeting by almost 400. An event which attracted a large attendance was the general conference on employment which was addressed by Col. Arthur S. Woods, Chairman of the President's Emergency Committee for Employment.

An Exhibition of Art by Engineers, initiated at this meeting, proved so interesting and delightful and aroused such enthusiasm that it was decided to repeat it at the next Annual Meeting. The formal opening of the exhibition was preceded by an hour of music by the "Bridge Engineers' Trio," composed of prominent consulting bridge engineers. More than 200 exhibits were on display, covering sculpture, paintings, prints, line drawings, and photographs by 78 professional engineers.

Another innovation at this meeting was the delivery of three lectures on "Talking With an Audience" by Dr. S. Marion Tucker, Head of the Department of English, The Polytechnic Institute, Brooklyn, which preceded the technical sessions on three mornings and which were so well attended that the Committee has made arrangements to have Dr. Tucker continue these talks at the next meeting.

At the usual New Members' Dinner, the outstanding features were the presentation of special badges to 14 fifty-year members of the Society, the award of the John Fritz Medal to Admiral David Watson Taylor, and the address of the evening given by Elliott Dunlap Smith, director of industrial engineering at Yale University, on the subject "Engineering Encounters Human Nature."

Birmingham Meeting

Southern hospitality and an excellent program of technical papers, well discussed, made the Birmingham meeting a splendid success.

The high spot of the meeting was the dinner at which addresses were given by Roy V. Wright and Dr. Harrison E. Howe. At this meeting also a session with papers contributed entirely by students proved of great interest. A special all-day excursion was arranged to the properties of the Tennessee Coal, Iron, and Railroad Company and a stop for a barbecue luncheon provided a delightful interlude. A second all-day excursion was arranged for the last day of the meeting to the Jordan Dam of the Alabama Power Company, where a modern high-head development was inspected.

Hartford Meeting

Technical papers of high excellence, noteworthy discussions on economic problems, entertainment features of unusual quality, and instructive plant visits marked the Hartford Meeting. A session on stabilization of employment proved one of the leading attractions and provided an opportunity for the exchange of views on many economic subjects. Howell Cheney of Cheney Brothers led this discussion.

A Lamb Bake which the Local Committee arranged was a great success, as was a musicale followed by addresses at the new Horace Bushnell Memorial Auditorium. Among those who spoke were the Honorable Hiram Bingham, Senator from Connecticut, whose subject was "The Engineer in Public Life," and F. M. Ryan of the Bell Telephone Laboratories, who gave an illustrated lecture on "Airplane Telephony." At the informal banquet, President Wright and E. E. Wilson, President of the Chance Vought Corporation and the Sikorsky Aviation Corporation, delivered addresses.

Kansas City Meeting

The participation of a number of organizations, including this Society, in the first Power Conference in the Southwest was an arrangement which offered an excellent opportunity for exchange of ideas on the different phases of power problems. The special program of this Society arranged by the Local Committee contributed in no small way to the success of the meeting.

In addition to these meetings, a session was held at the Annual Meeting of the American Society for Testing Materials, sponsored jointly by that Society and the A.S.M.E. Special Research Committee on Effect of Temperature on the Properties of Metals. The Committee on Meetings and Program was represented at the annual meeting of the Society for the Promotion of Engineering Education at Purdue in June by J. W. Parker, member of the Committee, who gave a talk on "The Need for Speech Training for Engineers."

Respectfully submitted,

G. M. Eaton, *Chairman*
F. M. Feiker
J. W. Parker
C. P. Bliss
R. I. Rees.

MEMBERSHIP

The Committee on Membership held 22 meetings during the fiscal year 1930–1931.

The number of applications considered in the transaction of its work and a summary showing the action taken follow:

New applications pending, October 1, 1930	292
Transfers pending, October 1, 1930	94
Applications received during fiscal year 1930–1931	1529
Total applications handled during year 1930–1931	1915
Recommended for membership	1648
Transfers denied	4
Deferred	1
Withdrawn	44
New applications pending, October 1, 1931	179
Transfers pending October 1, 1931	39
Total applications handled during year 1930–1931	1915

Those recommended for membership were divided into the following grades:

Members	287
Transfers to Member	153
Associates	21
Transfers to Associate	1
Associate-Members	260
Transfers to Associate-Member	98
Juniors	338
Juniors (R5, Rule 1)	490
Total recommended	1648
Transfers	252
Total new members recommended	1396

During the fiscal year 1930–1931 the Membership Committee made the following recommendations:

Elections declared void	187
Resignations accepted	465
Dropped from membership	712
Total loss	1364

Respectfully submitted,

F. A. Waldron, *Chairman*
H. W. Butler
H. Webster
O. E. Goldschmidt
H. A. Lardner.

PUBLICATIONS

During the past year the Publications Committee has been seriously concerned with an attempt to hold its expenditures within its budget. An increase in the activities of the Professional Divisions resulting in an overwhelming number of papers has made careful selection of material for publication more essential and has retarded the prompt

appearance of some papers in the Transactions. It is unfortunate that the number of papers awaiting publication should have increased as a result of limited funds, in view of the fact that the editorial staff had caught up with its work to a point where the prompt publication, one of the benefits that had been expected to accrue from the new scheme of issuing Transactions in sections, became feasible otherwise. Because of limited funds the final publication of many papers ready for printing has been postponed, and sections of Transactions have appeared containing a greatly reduced number of papers.

A study of the costs of publication under the new scheme, based upon experience since 1926, has rendered valuable information on the amount of the expenditure to which the Society is committed in the acceptance of a paper for preprinting for a meeting with assignment of final publication to the Transactions. This information has been passed on to the Professional Divisions with a request that only papers of the highest quality be recommended for printing in Transactions. Further study of the problem of quality and quantity of papers is necessary, and it is hoped that cooperation between the Publication Committee and the committees of the Society which secure papers will result in a satisfactory solution of the problem.

Financial stringency also resulted in the publication of a 2-page *News* throughout the summer and a reduction in text pages in *Mechanical Engineering* from 80 to 72 for the months of July, August, and September. As a further measure of economy the Publications Committee recommended to Council that the printing of the Membership List for 1932 be omitted.

The Record and Index was issued as a section of the 1930 volume of Transactions upon the recommendation of the Publications Committee and with the consent of Council. This resulted in a saving in printing and distribution costs.

The Committee and the Staff have spent much time and study in the development of *Mechanical Engineering*. The size of type of the text pages was enlarged. A new format was worked out and decided upon and will be adopted with the January, 1932, issue. The functions and relationships of the A.S.M.E. periodical publications, the *News*, *Mechanical Engineering*, and Transactions, have been thoroughly discussed and are being brought to the attention of the members through the Local Sections. In the meantime, *Mechanical Engineering* is undergoing a change in editorial function that should enhance its value and significance.

The Committee is pleased to report the publication by the Engineering Societies Monographs Committee of the first of a series of books to be issued under a joint arrangement with the four Founder Societies, the Engineering Societies Library, and the McGraw-Hill Book Company; this is Nádai's "Plasticity," translated, with revisions and additions, from the German. The text was secured and sponsored by the Applied Mechanics Division.

The Mechanical Catalog was issued September 30 and contained data on the products of 488 companies, as compared with 574 in 1930.

The Engineering Index had reached the fortunate position of meeting current expenses when subscriptions were adversely affected by general business conditions. A committee of Council was appointed at the Hartford Meeting to make a thorough study of the Index.

Respectfully submitted,

F. V. Larkin, *Chairman* S. F. Voorhees
W. H. Winterrowd S. W. Dudley.
L. C. Morrow

PROFESSIONAL DIVISIONS

The activities of the Professional Divisions during the year have been especially significant. Five new developments were gotten under way, the most outstanding of which was the First National Meeting held by the Applied Mechanics Division at Purdue University. The program was coordinated with the summer meeting of the Society for the Promotion of Engineering Education. The success of the meeting gives promise of gathering into the Society all engineers interested in this field and also indicates that the Society will be recognized internationally as the American forum for activities in this field.

The Petroleum Division continued its splendid activities around Tulsa in cooperation with the Mid-Continent Section. It is expected that reports of tests conducted on steam equipment for drilling rotary-drilled oil wells will be published early in 1932. Two new developments occurred in the East under the auspices of the Division. The first was a Petroleum Refining Meeting held in Elizabeth, N. J., May 5 and 6, which was very successful and is resulting in the development of a very valuable standard for testing the efficiency of oil-refining tubular heat-transfer equipment. The other development was the holding of a one-day lubrication engineering meeting at Pennsylvania State College on May 22, which brought together more than 100 specialists in this field. This was the first time the Society has held a technical meeting solely on this subject.

Another new activity launched during the year was the work of the Process Industries Committee, which is developing a Division. Its program included a cooperative meeting with the American Institute of Chemical Engineers at Swampscott, Mass., in June.

The fifth new activity deals with maintenance engineering, which for the first time was given special prominence in divisional programs. A successful national meeting on maintenance engineering was held in Cleveland during April under the auspices of the Management Division as part of the Industrial Congress.

In addition to these meetings initiated during the year, the four-point program of the Professional Divisions continues to function as may be seen by the accompanying table. This table of the Divisions' activities shows papers secured and presented by them during the year. This does not include papers presented by them at usual Metropolitan Section meetings or Division progress reports, but does include papers for Section meetings in which the Division assisted financially.

Special Committee Activities

The Professional Divisions are gradually developing a fifth activity which might be called "Special Committees." Of these the following have been especially active during the year:

The Elimination of Waste Committee of the Management Division continued its work to stimulate industrial plants to hold elimination-of-waste campaigns, especially during the month of April. This year the Committee went to considerable expense to develop a system of posters and information which could be used by small companies in holding campaigns. This waste-campaign unit sold for the small sum of $3, and more then 130 companies made use of the unit in holding campaigns. During the year the Committee also compiled a booklet published by the Society called "Waste Materials Dictionary" which gave information on the use of waste items. Some 50 companies contributed items to the Dictionary.

The Oil-Engine-Power Cost Committee of the Oil and Gas Power Division continued its splendid work and again compiled a report on the cost of oil power. This Committee has the active cooperation of users all over the country. The Committee should be especially commended on its splendid report, which has met with wide approval.

The Pure Air Committee, which was organized this year through the aid of the Fuels Division, already has a subcommittee at work. The Committee is broadening the scope of work of the former Smoke Abatement Committee of the Fuels Division, and has upon it representatives of several other organizations. It gives promise of becoming a very valuable activity and one which will warrant the close cooperation of the whole Society.

The Manufacturers Committee of the Materials Handling Division has a subcommittee developing tentative standards for crawlers, locomotive cranes, and industrial cranes. This Committee has a splendid field of activities, which if given proper support by industry should accomplish valuable work. The Division also has a special committee at work on analyzing the field of various types of conveyors.

This year the Hydraulic Division's special committee on the translation of important foreign hydraulic papers began its work, which promises to be of considerable value.

The Petroleum Division has a number of committees active in the Mid-Continent Section and a newly formed committee which developed out of the Elizabeth meeting.

The Aeronautic Division has formed a committee to study wage-incentive practice in its industry. The Applied Mechanics Division has subcommittees stimulating developments in vibration, plasticity, and impact. The Wood Industries Division is forming a subcommittee to report on construction and design of doors, and also has a newly formed research survey committee.

Division Meetings

In addition to the new national meetings gotten under way, other Divisions have continued to function as usual.

The Printing Industries Division deserves special recognition for its meeting last March at the Government Printing Office in Washington in joint cooperation with the United Typothetae of America and the Government Printing Office. More than 500 attended this two-day meeting. The entire proceedings of the gathering were published by the Government Printing Office, as part of its splendid cooperation. This meeting was called "The Second Conference of Technical Experts in the Printing Industry" and it gave very definite evidence of the fact that the Printing Industries Division has succeeded in winning for itself recognition from the entire printing industry as the neutral forum for the discussion of its engineering problems. The

Division deserves a great deal of credit for the development of these conferences and for the recognition it has won the Society in this field.

The Aeronautic Division held its national meeting at Baltimore, Md., under the auspices of the Baltimore Section, Baltimore Engineers Club, and other local societies, which made possible the largest technical meeting ever held in aeronautics. As many as 50 papers were presented, covering almost every branch of aeronautics. The local program was especially well arranged and added much to the interest and pleasure of the meeting. During the year the Aeronautic Division also assisted Local Sections in Toronto, Buffalo, Cleveland, Philadelphia, and Detroit in holding meetings of more than local interest. In addition, the Aeronautic Division through the Metropolitan Section held a memorial meeting in the city of New York in February to honor the services rendered to aeronautics by the late Daniel Guggenheim, Glenn H. Curtiss, and Chance M. Vought. The program was arranged by the Division, but the meeting was a cooperative one in which several other societies participated. During the year the Division initiated the formation of a standards committee on aircraft safety inspection.

The Materials Handling and Management Divisions held their cooperative meeting in April in Cleveland as part of the National Industrial Congress. The Elimination of Waste Committee sponsored two sessions under the auspices of the Management Division; while the holding of a Maintenance Meeting as part of the Congress has already been mentioned. The Materials Handling Division held an extensive national meeting, covering its field very thoroughly. In spite of the depression the attendance at the Congress from the outside was larger than at the one held the year before in Chicago.

The Fuels Division held its fourth national meeting in February in Chicago in coordination with the Midwest Power Conference.

The Iron and Steel Division cooperated with the Detroit Section in a one-day meeting held in May. The Division also cooperated in the National Metal Congress held in Boston in September.

The Oil and Gas Power Division held its usual successful national meeting at the University of Wisconsin in June. This is the first time the Division has held a national meeting in the West, and its success is especially gratifying.

The Wood Industries Division held its first eastern national meeting in New York in October, 1930. Encouraged by the success of this meeting, the Division decided to hold its October, 1931, meeting at Winston-Salem, N. C. Although the membership of the Society is rather small in that neighborhood, the Division held a very successful national meeting, winning the hearty support and cooperation of the Southern Furniture Manufacturers Association. This gave promise of future benefit to the Society, as woodworking industries in the South have increased rapidly, even during the depression.

The Textile Division held a meeting jointly with the local engineering society and our Section in Providence, R. I., on October 6, and also a Southern Meeting as part of the Birmingham Spring Meeting.

The Machine Shop Practice Division cooperated in the National Metals Congress in Boston on September 22, and during the year has formed two promising subcommittees, one on foundry practice and the other on machine design. At the Boston meeting a conference was called on wear of metals, and the Division, jointly with the Iron and Steel Division, recommended the initiation of a research committee on the subject.

The Railroad, Power, and National Defense Divisions continued to function as adequately as their field demanded, using the regular Society meetings for their activities.

PROFESSIONAL DIVISIONS ACTIVITIES IN 1931—FOUR-POINT PROGRAM

		Aeronautic	Applied Mechanics	Fuels	Hydraulic	Iron and Steel	Machine Shop Practice	Management	Materials Handling	National Defense	Oil and Gas Power	Petroleum	Power	Printing Industries	Process Committee	Railroad	Textile	Wood Industries	Total, 1931	Total, 1930
1	*Society Meetings*[1] (Semi-Annual, Regional, Annual)																			
	(a) Number of sessions	4	2	3	2	1	2	4	3	0	2	3	4	2	1	3	2	1	39	43
	(b) Number of papers	9	4	6	5	2	3	7	7	0	5	6	9	5	1	5	4	2	80	87
2	*Division Meetings*																			
	(a) Number of meetings	3[2a]	1	1	0	1[2d]	1[2d]	1[2b]	1[2c]	0	1	2	0	1	1	0	2	1	17	14
	(b) Number of sessions	22	4	6	0	1	1	7	8	0	6	5	0	4	1	0	2	4	71	51
	(c) Number of papers	66	17	8	0	0	3	20	23	0	17	13	0	12	1	0	4	11	195	159
	(d) Attendance, members	460	170	500	0	100	75	..	450	0	160	260	0	200	35	0	110	170	2690	3190
	(e) Attendance, non-members	360	50	300	0	200	25	..	220	0	80	180	0	300	100	0	50	120	1985	1795
	(f) Attendance, total	820	220	800	0	300	100	..	670	0	240	410	0	500	135	0	160	290	4645	4985
1 and 2																				
	(a) Total Division sessions held	31	6	9	2	2	3	11	11	0	8	8	4	6	2	3	6	6	118	94
	(b) Total Division papers presented	75	21	14	5	2	6	27	30	0	22	19	9	17	2	5	8	13	275	246
3	*Research*																			
	(a) Survey Committee at work	0	0	1	1	0	0	0	1	0	1	1	0	1	0	1	1	1	9	
	(b) Executive Committee acting as Survey	1	1	0	0	1	1	1	0	1	0	0	1	0	0	0	0	0	7	8
	(c) Projects being developed at present	0	3	0	0	0	0	1	2	0	0	6	0	0	0	0	0	0	12	13
	(d) Special Research Committee authorized during year at work	1	0	0	0	0	1	0	0	0	0	1	0	0	0	0	0	0	3	8
4	*Progress Reports*	1	1	1	1	1	1	1	1	0	1	1	1	1	0	1	1	1	15	14
5	*Enrolment in Divisions, 1930.*	2575	3803	3818	2015	2250	3951	6255	2608	*	3084	1354	4033	363	0	1250	609	727		38,695
6	*Enrolment in Divisions, 1931.*	2587	3785	3829	2012	2245	4019	6301	2535	*	3137	1317	4029	400	0	1210	590	730	38,721	

[1] Includes Annual, French Lick, Birmingham, and Kansas City Meetings.
[2a] National Meeting, Baltimore, May, 1931; Toronto Meeting, March, 1931; Cleveland Meeting, Sept., 1931. [2b] and [2c] National Meetings were held with Management Congress. [2d] National Meetings were held with Metal Congress.
* All members are registered in National Defense Division.
NOTE: Table includes meetings held from October 8, 1930, to October 6, 1931.

The Railroad, Power, and National Defense Divisions continued to function as adequately as their field demanded, using the regular Society meetings for their activities.

Future Developments

During the year of 1932 it is expected that greater attention will be given to subcommittee activities of the Divisions, warranted by the increased financial support granted by Council. Several of the Divisions have already taken advantage of this by beginning to develop valuable subcommittee activities.

H. W. Brooks, *Chairman* C. B. Peck
W. F. Dixon W. A. Shoudy
P. T. Sowden

LOCAL SECTIONS

New Section

The members in Eastern Connecticut, with the approval of Council, organized the seventy-second Section of the Society, known as the Norwich Section. This was accomplished in time for their delegate to participate in the Local Sections Conference at the 1930 Annual Meeting. During the year this Section held four successful meetings.

Prospective Sections

Most of the Industrial areas of the United States have been provided with Local Sections which are now functioning. Sections at Ft. Wayne, Ind., Central Illinois, with headquarters probably at the University of Illinois, Nashville, Tenn., and Central Wisconsin, are probable developments in the near future.

Membership Chart

The number of meetings held by Sections during 1930–1931 is shown in Appendix I. This table also details the membership of the respective Sections on October 1, 1930, and October 1, 1931, the net increase, the percentage of membership increase, and the fact that 92 per cent of the entire membership is now residing within the territory of established Local Sections.

New Membership

The potential membership of the Society is enormous. Even with 20,000 members it can be conservatively stated that not more than half of the qualified mechanical engineers are enrolled. The most economical and dignified method of membership development would be for each of our 20,000 members to encourage one engineering associate, whom he knows to be well qualified, to file his application. If members would seriously undertake to do this, the membership could be doubled at little expense. The saving effected could be well applied to other Society services. Normal losses in membership include, on the average, 150 deaths, 350 resignations, and 800 who allow membership to lapse, making a loss of 1300 members or about 7 per cent.

Meetings

The Local Sections have cooperated in the development of National Meetings of the Society, particularly in the arrangements for the entertainment and plant visitations program at Birmingham, where the 1931 Semi-Annual Meeting was held; and for the National Regional Meetings held at French Lick Springs, Ind., October, 1930; at Hartford, Conn., June, 1931, and at Kansas City in September, 1931. Local Sections cooperated with the Professional Divisions in holding National Divisional Meetings as follows:

National Divisional Meetings, October 1, 1930–September, 1931

October 6–8, 1930	Petroleum	Tulsa, Okla.
October 16–17, 1930	Wood Industries	New York, N. Y.
October 22, 1930	Textiles	Greenville, S. C.
February 11–13, 1931	Fuels	Chicago, Ill.
March 16–17, 1931	Printing Industries	Washington, D. C.
April 13–16, 1931	Management and Materials Handling	Cleveland, Ohio
May 12–14, 1931	Aeronautics	Baltimore, Md.
June 15–16, 1931	Applied Mechanics	Purdue Univ., Lafayette, Ind.
June 23–26, 1931	Oil and Gas Power	Univ. of Wisconsin, Madison

In addition, Local Sections held 471 meetings of their own as shown in Appendix I.

Cooperation With Professional Divisions

The Sections also have availed themselves of each opportunity to cooperate with Professional Divisions by planning their programs to cover the subjects represented by the Professional Divisions; this practice is followed particularly in the larger Sections, like the Metropolitan and Philadelphia. Individual Sections have given fine support to many of the Professional Divisions by cooperating with them in various undertakings.

Scope of Service

The services available to members through Local Sections vary to some extent with the activity of the particular Section. The extent of activities is governed by conditions peculiar to the Section, the number of members, and its geographical location. A wide range of activities include the Employment Service, opportunities to study public speaking through classes arranged by the Local Section, opportunities for inspection of industrial plants, contacts with, and often, privileges of local engineering societies and clubs; opportunities for civic service, etc., besides the usual engineering meetings. Individual members with worthwhile ideas will usually find that their Local Executive Committee will welcome their interest and assistance on special committees.

Tours and Transportation

The Committee on Local Sections arranged for reduced fares to the Annual, Semi-Annual, and other national meetings of the Society during the year, thereby saving thousands of dollars to the members of the Society. Also, after the Council had accepted the invitation of the V.D.I. to have members of the Society participate in its Seventy-Fifth Anniversary Celebration at Cologne, June 26–30, 1931, it successfully planned, developed, and conducted a tour to Europe which was participated in by a large number of members and their wives, there being 85 from the United States in attendance at the Cologne Meeting of the V.D.I. Considerable international goodwill was built up through official visits by the party to several other societies. The Institution of Mechanical Engineers entertained at a tea in their rooms, and the party was also guests of officers of the Society of Civil Engineers of France, and of the Faculty of the École Centrale des Arts et Manufactures, which they visited. Thus the engineering societies of Germany, England, and France renewed their friendship with the A.S.M.E. Important industrial plants were also visited during the trip.

Elimination of Waste

Six sections cooperated with the Management Division in organising campaigns and meetings on the subject of "Elimination of Waste," as follows:

Section	Date of Meeting	Subject
Cleveland	April 13–17, 1931	National Division Meeting on Management, Materials Handling, and Maintenance
Detroit	November 17, 1930	Management as Related to a Medium-Size Plant
Hartford	April 7, 1931	Elimination of Waste in the Industrial Plant
Metropolitan	February 6, 1931	The Scope of Management in Our Future Business Development
Metropolitan	April 3, 1931	Product Design for Increased Utility and Improved Marketability
Philadelphia	October 28, 1930	Factory Management

Foreign Relations

In addition to the foreign relations developed in connection with tours, there has been built up and functioning for several years a service to foreign visitors and to members of our Society or members of other engineering societies traveling either in the United States or in other countries. Some idea of the extent of this service is contained in the résumé (Appendix II) covering the three-year period 1928–1931.

In addition to this service, from June 1, 1930, to October 1, 1931, 18 members of the Society were given letters of introduction to various organizations, societies, and plants in ten different countries of Europe. During the same period 27 members of the Society were given letters enabling them to gain admittance to various plants and works which they desired to visit. Prior to June, 1930, no records were kept of introduction service rendered the members of the A.S.M.E.

Employment

The administration of the Engineering Societies Employment Service, in so far as it pertains to our Society, was assigned by the Council during the year to the Committee on Local Sections. All Local Sections rendered special and valuable service at the beginning of the year 1931 by responding to the request of the A.S.M.E. Committee on Employment, of which Mr. Ralph E. Flanders was the Chairman, and which cooperated closely with the President's Emergency Committee for Employment in accordance with Col. Arthur Wood's personal appeal to the members of the Society at the 1930 Annual Meeting. Excellent progress was made during the year by the reestablished Volunteer Employment Committee, composed of members of the four Founder Societies out of employment. These men visited various concerns to urge upon them the employment of engineers and jobs were located as a result of their activity. Also, more than 30,000 letters were sent to various industrial concerns in the United States, soliciting opportunities for employment. During the year the Service placed 559 members in positions, and 231 non-members were placed in positions for which we could not find members with the necessary qualifications, making a total of 790 placements. These were divided as follows:

New York:	Members	391
	Non-Members	110
Chicago:	Members	63
	Non-Members	47
San Francisco:	Members	105
	Non-Members	74

However, because of the falling off in positions open, the income derived by the Employment Service from contributions made by members placed in positions was so reduced that it was necessary for the four societies not only to renew on April 1, 1931, their financial support of the Service, but to increase the budget by 50 per cent. Plans are progressing for the continuation during the ensuing year of the Employment Service with its three offices in Chicago, New York, and San Francisco, which will require finances of double the amount previously contributed. Members generally will be requested to be particularly alert to find opportunities for the very large number of members of the Society out of employment. The coming winter threatens to be one of the most difficult periods in the history of our country and a corresponding degree of importance should be attached to the Employment Service in its efforts to assist members to "help find a job for an engineer."

Mid-West and Mid-Continent Offices

The Mid-West Office located at 205 West Wacker Drive, Chicago, Ill., and the Mid-Continent Office, 503 Tulsa Bldg., Tulsa, Okla., have made much progress in rendering service to the Student Branches and Local Sections in those areas of the country. (See Appendixes III and IV.)

Local Sections Visited by the Committee

During the year President Wright has either visited or has definitely scheduled a visit to every Local Section of the Society. This is the first time that this will have been accomplished. Other presidents have made notable contributions in this direction and have visited a majority of the Sections, but President Wright will be the first to meet with all of them. In addition, the Committee on Local Sections visited Local Sections as shown in Appendix V.

Local Sections Financial Policy

The Local Sections Conference of 1930 adopted a recommended financial policy of the Society pertaining to Local Sections. Copies may be obtained from the Secretary's office if desired.

Harry R. Westcott, *Chairman* — Jiles W. Haney
Charles W. Bennett — William Lyle Dudley.
James M. Todd

APPENDIX II VISITORS FROM FOREIGN COUNTRIES TO A.S.M.E., JUNE, 1928, TO SEPTEMBER 30, 1931

Visits were made to the A.S.M.E. by 208 people from 22 countries, as follows:

Australia	1	Austria	5	Belgium	1
China	1	Czechoslovakia	16	Denmark	3
England	62	France	19	Formosa	1
Germany	64	Holland	7	Hungary	1
Italy	4	Japan	4	Mexico	2
Norway	4	Poland	2	Roumania	1
Russia	1	Scotland	3	Sweden	3
Switzerland	3				

Of the 62 visitors from England, 34 were students of The Liverpool University, sponsored by the Liverpool Engineering Society, Liverpool, England. Each of the past three summers approximately 12 of these students have worked in the engine room of a transatlantic liner, and in accordance with our arrangements have visited several power or other plants in New York and vicinity during the period the steamship was in New York harbor. Their interest in different phases of engineering is shown by the following figures as to the type of plants they requested permission to visit:

11 electrical engineering
11 mechanical engineering
9 civil engineering
1 general engineering
1 internal combustion engine
1 naval architecture

According to our records Germany sent 64 visitors, of which practically one-third were young engineering graduates sponsored by The German Students Cooperative Association. We have aided a number of them to obtain employment in the United States.

For each visitor, we prepared letters of introduction to various companies whose plants and factories they desired to visit.

The number recorded, 208, represents only a portion of the number of visitors served. In many instances when we arranged inspection trips for a party of foreign engineers we kept a record of the leader of the party only. We have served several visitors for whom no record was entered. In the past three years this Society has served possibly 300 engineers of other countries.

APPENDIX I MEMBERSHIP IN LOCAL SECTIONS, OCTOBER 1, 1931

Section	Membership Oct. 1, 1930	Oct. 1, 1931	Increase	Per cent	Meetings, 10/30-10/31
Akron-Canton	163	170	7	4.3	9
Anthracite-Lehigh Valley	334	296	—38	—11.3	11
Atlanta	113	107	— 6	— 5.3	6
Baltimore	225	228	3	1.3	9
Birmingham	99	92	— 7	— 7.0	3
Boston	780	776	— 4	— 0.5	8
Bridgeport	200	198	— 2	— 1.0	8
Buffalo	244	240	— 4	— 1.6	11
Central Pennsylvania	113	112	— 1	— 0.8	2
Charlotte	43	40	— 3	— 6.9	4
Chattanooga	24	22	— 2	— 8.2	2
Chicago	1145	1160	15	1.3	8
Cincinnati	343	327	—16	— 4.6	12
Cleveland	435	416	—19	— 4.3	8
Colorado	96	97	1	1.0	8
Columbus	85	89	4	4.2	6
Dayton	102	119	17	16.6	1
Detroit	554	557	3	0.5	11
Erie	95	100	5	5.2	4
Florida	68	67	— 1	— 1.4	3
Green Mountain	48	48	..	..	1
Greenville	41	39	— 2	— 4.8	2
Hartford	151	142	— 9	— 5.9	10
Houston	144	148	4	2.3	3
Indianapolis	147	149	2	1.3	4
Inland Empire	25	23	— 2	— 8.0	7
Kansas City	136	143	7	5.1	12
Knoxville	49	52	3	6.1	3
Los Angeles	445	452	7	1.5	9
Louisville	44	39	— 5	—11.3	5
Memphis	28	32	4	22.2	1
Meriden	34	32	— 2	— 5.8	5
Metropolitan	4925	5060	135	2.7	31
Mid-Continent	167	184	17	10.1	6
Milwaukee	298	297	— 1	— 0.3	11
Minneapolis	74	75	1	1.3	6
Nebraska	34	40	6	17.6	6
New Britain	53	46	— 7	—13.2	4
New Haven	145	134	—11	— 7.5	9
New Orleans	90	101	11	12.2	1
North Texas	69	72	3	4.3	4
Norwich[1]	...	39	39		4
Ontario	163	191	28	17.2	10
Oregon	75	71	— 4	— 5.3	8
Peninsula	80	85	5	6.2	2
Philadelphia	1316	1276	—40	— 3.0	14
Pittsburgh	611	608	— 3	— 0.4	4
Plainfield	320	321	1	0.3	9
Providence	200	212	12	6.0	7
Raleigh	30	27	— 3	—10.0	5
Rock River Valley	84	79	— 5	— 5.9	5
Rochester	138	140	2	1.4	7
St. Joseph Valley	61	64	3	4.9	6
St. Louis	308	299	— 9	— 2.9	10
St. Paul	52	48	— 4	— 7.6	2
San Francisco	423	438	15	3.5	10
Savannah	20	21	1	5.0	1
Schenectady	234	244	10	4.2	9
Susquehanna	70	70	..	..	4
Syracuse	126	135	9	7.1	5
Toledo	95	95	..	..	12
Tri-Cities	76	78	2	2.6	6
Utah	37	37	..	..	7
Utica	35	34	— 1	— 2.8	2
Virginia	183	183	..	..	1
Washington, D. C.	200	202	2	1.0	7
Waterbury	92	95	3	3.2	8
Western Massachusetts	145	153	8	5.5	6
Western Washington	116	107	— 9	— 7.7	9
West Virginia	66	73	7	10.6	1
Worcester	197	191	— 6	— 3.0	10
Youngstown	84	82	— 2	— 2.3	6
	18045	18219	174	0.9	471

Total membership in Society, Oct. 1, 1930, 20,016; Oct. 1, 1931, 19,979.
Total number of Student Branches, 108.
[1] New Section, December, 1930.

APPENDIX III MID-WEST OFFICE

The first field office of the Society was established in Chicago, Ill., in October, 1929. In April, 1930, R. R. Leonard, a member of the New York staff, was transferred to the newly established office to act as a representative of the Field Department in the following states covered by the office: Illinois, Indiana, Iowa, Kansas, Kentucky, Michigan, Minnesota, Missouri, Nebraska, Ohio, and Wisconsin. Within these states 21 Local Sections and 27 Student Branches function. They were visited and given cooperation in working out the details of operation of this new venture of the Field Department.

In addition to personal visits contacts with each group were made on numerous occasions by telephone or correspondence. Stress has been laid on the facility of a nearby branch office of the Headquarters organization to render prompt assistance. The intimate contact of the Mid-West Office with the headquarters staff greatly facilitates action in regard to the problems of the Local Sections and Student Branches.

Local Sections have been aided in the preparation of local programs, as well as in their participation in regular Society and Division meetings. The headquarters staff has been assisted in the operation of meetings within the territory of the Mid-West Office. Student Branches have been helped in program set-ups, speakers have been obtained, and advice on general activities has been supplied. Through intimate contact with each of the groups, Local Section and Student Branch, it has been possible to use the experiences of one group to the advantage of other groups. Information on the activities of individual members has been obtained, and as a result members capable of offering valuable committee assistance have been discovered and some of these have already been enlisted for service. New activities of Professional Divisions have been initiated as a result of information uncovered through this more intimate contact with organized groups.

In the Chicago area especially, engineering and other organizations, as well as individuals, have found the Mid-West Office a convenient channel through which information on Society activities and general engineering information may be obtained. The office has received a number of calls for Society publications, and in a few instances valuable time has been saved through its ability to supply them immediately.

Local Sections and Student organizations alike have expressed enthusiastic approval of this new service. They have accepted it with obvious pleasure and have shown marked willingness to offer suggestions for improvement of conditions among all such groups. A few Sections have even asked for an extension of the service by placing the mid-west representative in temporary quarters within the Section for periods ranging from one week to two months. Others have expressed the opinion that a beneficial move would be to establish offices of the character of the Mid-West Office in other parts of the country.

The Mid-West Office has taught the organized groups within its territory that the close proximity of the office is favorable to quick interchange of information, and they have expressed the desire to have the greatest possible service through that point.

There is no doubt that the new service has effected economies, but merely from the standpoint of closer contact with the organized groups within the territory covered by the office and the increased good-will resulting from the more intimate contacts, it would seem that the Standing Committee on Local Sections has been more than justified in its creation of the Mid-West Office.

APPENDIX IV MID-CONTINENT OFFICE

The activities of this office are concerned principally with the problems of the petroleum industry. The Steam Subcommittee of the Production Committee has compiled and sent in to the Mid-Continent Office its report on "Tests on Steam Equipment Used in Drilling a Rotary Drilled Oil Well" and a second report on "Tests on Slush Pumps and Boilers Used in Drilling a Rotary Drilled Oil Well." These two reports, as well as all others on rotary drilling, will be issued under the auspices of the recently appointed Special Research Committee on Rotary Drilling of Oil Wells. A report by the Electrical Subcommittee on Petroleum Production was also completed.

The Fluid Meters Subcommittee has completed the layout plans for tests to determine the accuracy of existing coefficients and correction factors for viscosity, gravity, and temperature of oils flowing through orifices and flow nozzles. The necessary equipment needed to perform these tests, which will cover a period of two years, is now being assembled through the generous cooperation of several manufacturing companies. Prof. W. H. Carson, Head of the Department of Mechanical Engineering at the University of Oklahoma, will have charge of conducting the tests.

The Unfired Pressure Vessel Committee has sent in to the Boiler Code Committee its recommendations for the use and construction of unfired pressure vessels used in the petroleum industry, as well as its views on the revised draft pertaining to unfired pressure vessels.

The Automatic Pumping Station Committee and the Economic Analysis of Automatic Pumping Station Committee designed and constructed an automatic pumping station for use on oil pipe lines, which was exhibited at the Seventh International Petroleum Exposition at Tulsa, Okla., in October, 1930. As a result of this experimental station six such stations have been put into operation during the year. Two joint meetings have been held since then at which operation problems of the latest automatic stations were discussed and corrections suggested. Several changes and improvements of equipment will be made as a consequence of these meetings. The Economic Analysis Committee has collected data on the salvage values of various equipment, insurance, and tax rates. The Committee will compile, from reports sent in from several automatic stations, a thorough report on improvements made in equipment and operation since the exhibition at the oil show.

A campaign has been started among the pipe-line companies to have each appoint a Waste Elimination Committee and to have a representative on the A.S.M.E., Mid-Continent Section, Waste Elimination Committee.

Members are beginning to make use of this office as a medium for obtaining information pertaining to technical problems confronting them in their business.

HARRY R. WESTCOTT, *Chairman* J. W. HANEY
C. W. BENNETT W. L. DUDLEY.
J. M. TODD

APPENDIX V LOCAL SECTIONS VISITED BY MEMBERS OF THE COMMITTEE ON LOCAL SECTIONS, 1930–1931

Section	H. R. Westcott, *Chairman*	C. W. Bennett	J. M. Todd	J. W. Haney	W. L. Dudley	E. Hartford, *Secretary*
Birmingham	Apr. 1931	Apr. 1931	Apr. 1931	Apr. 1931	Apr. 1931	Apr. 1931
Chicago	...	...	...	...	...	Sept. 1931
Hartford	June 1931	...	...	...	...	Nov. 1930
Kansas City	Sept. 1931	...	...	Sept. 1931	...	Sept. 1931
Knoxville	Apr. 1931	...	...	...	...	...
Meriden	Feb. 1931	...	...	...	...	...
Metropolitan	...	...	...	...	...	(Resident)
Mid-Continent	...	...	...	...	...	Sept. 1930; Apr. 1931 Sept. 1931
Nebraska	...	...	...	(Resident)	...	...
New Britain	Feb. 1931	...	...	...	...	...
New Haven	(Resident)	...	...	...	...	Apr. 1931
New Orleans	...	...	(Resident)	...	...	...
Pittsburgh	...	(Resident)	...	...	...	...
Providence	Feb. 1931	...	...	...	...	...
Rochester	...	...	...	...	...	Sept. 1931
St. Louis	...	...	...	Apr. 1931	Apr. 1931	Sept. 1931
Utica	...	...	...	...	...	Mar. 1931
Washington, D. C.	...	...	...	...	...	Jan. 1931
Waterbury	Feb. 1931	...	...	...	...	...
Western Massachusetts	Feb. 1931	...	...	...	...	...
Western Washington	...	...	...	...	(Resident)	...
Worcester	Feb. 1931	...	...	...	...	...

Due to extensive visits to Local Sections by the President this year, the Committee considerably curtailed its usual travel.

AWARDS

This Committee has met nearly every month, except during the summer season.

In order to read all papers submitted for the various awards, the Committee arranged a new plan whereby the best papers of each group were selected by a member and circulated to other committee members.

The Committee made the following recommendations to Council for the Awards for 1931:

(1) The A.S.M.E. Medal to Albert Kingsbury for his research and development work in the field of lubrication.

(2) The Melville Medal to A. E. Grunert for his paper entitled "Comparative Performance of a Pulverized-Coal-Fired Boiler Using Bin System and Unit System of Firing."

(3) The Junior Award to M. K. Drewry for his paper entitled "Radiant-Superheater Developments."

(4) The Charles T. Main Award to Robert E. Klise of the University of Michigan for his paper on "Interchangeability—Its Development and Significance in Industry."

(5) One Student Award to Jules Podnossoff of the Polytechnic Institute of Brooklyn for his paper entitled "Pressure and Energy Distribution in Multi-Stage Steam Turbines Operating Under Varying Conditions."

The Committee recommended only one Student Award this year, due to the caliber of the eligible papers received.

The Committee announces that the subject determined for the Charles T. Main Award for 1932 is as follows: "Apprenticeship and Vocational Training—Their Effect on Industry and Public Life."

It is fortunate for students of engineering, in need of financial assistance, that the Max Toltz Fund has been available. The demand for loans has been unusually heavy this year and the Committee approved fourteen loans to students from the Fund. These loans amounted to $2625.00, more than twice as much as was loaned last year.

The Committee has given some thought to a codification of its practice in judging papers and has a statement of this in preparation. It is hoped that this statement will also be of value to the Professional Divisions as a guide to them in recommending papers to the Committee on Awards.

The statements in the several publications of the Society describing the various awards have been studied and harmonized by the Committee this year.

The will of the late Worcester Reed Warner provided a generous bequest for a medal to be known as the Worcester Reed Warner Medal. It is to be a gold medal to be bestowed on the author of the most worthy paper received, dealing with progressive ideas in mechanical engineering or efficiency in management. The Committee has selected an artist and will soon be ready to have the dies made. The hearty cooperation of Mrs. Warner and Mr. Swasey in assisting the artist and the Committee is much appreciated.

Respectfully submitted,

H. L. Seward, *Chairman*
A. M. Greene, Jr.
K. H. Condit
F. L. Eidmann
W. L. Batt.

CONSTITUTION AND BY-LAWS

The Committee on Constitution and By-Laws serves in an editorial capacity where amendments to the Constitution, By-Laws, and Rules are required or new policies necessitate changes.

The most important change made during this past year is in the amendment to the Constitution affecting the initiation fees in the several grades, approved by the entire membership on a ballot which closed November 10, 1930. The amendment went into effect on its announcement at the Annual Meeting in December, 1930. The new wording follows:

Article C5 Fees and Dues

Sec. 1 The initiation fee for membership in each grade shall be:

Member	$35*
Associate	35*
Associate-Member	25†
Junior	10†
Promotion from Junior‡ to Associate-Member	15
Promotion from Associate-Member to Member	10
Promotion from Junior to Member or Associate	25

* Formerly $25.
† No change.
‡ Promotion from Junior to higher grade, formerly $10.

With the recommendation of the committees concerned with membership, publications, and Local Sections, the Council has reworded part of B14 (3) in order to provide for continuing to send the *A.S.M.E. News* to members after the expiration of the three months' limit for payment of dues (in advance). Other publications are not mailed. Other proposals for rewording the By-Laws have been submitted to Council, but not yet acted upon.

The Publications Committee as a matter of economy recommended, and the Council approved, the future publication of the Constitution, By-Laws, and Rules as a separate pamphlet. No new edition has been printed since that of the Record and Index of 1929.

At the request of the Council the Committee has under consideration the editing of the By-Laws and Rules to bring into better relation certain sections of the rules for various committee work, and to condense other sections. However, this work is not completed, as all Standing Committees of the Council have in process the review of their own special rules; considerable headway has been made, but the work must be carried forward into the new administrative year.

Respectfully submitted,

A. D. Blake, *Chairman*
Thos. C. McBride
R. S. Neal
H. H. Snelling
P. R. Faymonville.

RELATIONS WITH COLLEGES

Student Branches were established at the University of Alabama, Tuscaloosa, Ala., Harvard University, Cambridge, Mass., and the University of South Dakota, Vermilion, S. D., during the year, bringing the total number of Student Branches to 108.

There were 867 Student Branch meetings reported during the school year 1930–1931, an increase of 230 over last year, or 36 per cent. President Wright visited 45 Student Branches during the school year 1930–1931 and 17 Branches in the fall of 1931.

Membership in the Student Branches increased by 660 members over the 1929–1930 school year. Twenty-two joint meetings with Local Sections were held.

Student Branch Conferences were held at the Annual Meeting, the Birmingham Meeting, and the Kansas City Meeting. These conferences were well attended. The Birmingham Conference and the program therefor was arranged by a special committee appointed for the purpose. More than 125 undergraduates were present to hear several excellent papers presented by members of Student Branches.

At the Birmingham Semi-Annual Meeting of the Society an even more important event occurred. Mr. M. C. Maxwell, Chairman of the Committee this year, presented a new policy for Student Branch activities. This policy had been approved at a meeting of the Committee held in New York early in April. The policy was approved in principle by Council. It provides that undergraduates be admitted to a grade of membership designated "Student Member," that application for this grade be made upon a regular form provided by the Society, and that a fee of $3 accompany this application. The application, which must bear the endorsement of the Honorary Chairman, is passed upon by the Membership Committee of the Society and then by Council in the same way as all other applications. Upon election, the Student is provided with a regular Membership Card bearing the designation "Student Member." The Student is also provided with a Student Membership badge bearing the insignia of the Society. Each Student Branch having fifteen or more "Student Members" is entitled to receive reimbursement for necessary expenses to the amount of $25 per annum and each continues to receive the Transactions of the Society as issued and annually a bound volume of Transactions. One copy of *Mechanical Engineering* is sent for each fifteen Student Members, and each Student Member receives semi-monthly a copy of the current issue of the *A.S.M.E. News*. The latter will be developed to provide sufficient space to record the interesting events and progress of the Student Branches and their members. The plan provides for mileage for one member of each Student Branch to a Regional Student Branch Conference, and for transfer to Junior Membership in the Society, upon the endorsement of three Faculty members, with suspension of the first year's dues.

The Council decided at its Hartford meeting to provide $2850 as a special fund to inaugurate the new policy in fifteen colleges in the South (east of the Mississippi River). President Wright appointed Mr. Lewis F. Gordon of Atlanta, Georgia, to membership on the Committee on Relations with Colleges for a five-year period, thereby giving further recognition to the southern Student Branches and assuring them of active leadership by the Committee. As the year closes, Chairman Maxwell, Mr. Gordon, and Ernest Hartford, Secretary to the Committee, have arranged an itinerary which will bring them to each of the fifteen colleges to consult with their faculties and officers during October, 1931.

Respectfully submitted,

Maxwell C. Maxwell, *Chairman*
D. B. Prentice
E. F. Church, Jr.
Wm. L. Abbott
Lewis F. Gordon.

EDUCATION AND TRAINING FOR THE INDUSTRIES

Meetings of the Committee were held during the Annual Meeting, and a proposed joint meeting with the Management Division was discussed with President-Elect Wright.

The Committee considered it unwise to attempt to survey training processes in industry under present conditions. The papers arranged for the December, 1930, meeting were:

(1) "Apprentice Training in Virginia," by C. F. Bailey, Newport News Shipbuilding and Dry Dock Company.

(2) "The Apprentice Training Movement in Wisconsin Industry," by Harold S. Falk, Vice-President and Works Manager, The Falk Corporation.

(3) "Opportunities for Aviation Training," by J. S. Marriott, Chief of Inspection Service, Aeronautics Branch, U. S. Department of Commerce.

Mr. Charles H. Colvin, President, The Pioneer Instrument Company, Brooklyn, New York, presided at this meeting. About 200 were in attendance.

The following program was arranged for the Hartford Meeting, Monday, June 1, 1931:

(1) "Training Foremen to Deal With Human Problems in Industry," by John A. McCarthy, State Supervisor of Trade and Industrial Education, New Jersey Department of Public Instruction.

(2) "Training Apprentices and Foremen," by Frances Perkins, Industrial Commissioner, New York Department of Labor.

A joint program with the Management Division has been arranged for the 1931 Annual Meeting, as follows:

(1) Opening Address, by Roy V. Wright, President of the Society.

(2) "Foremanship Training," by U. J. Lupien, Cheney Brothers.

(3) "Modern Industrial Training," by George A. Seyler, Lunkenheimer Company.

Respectfully submitted,

R. L. Sackett, *Chairman*
D. C. Jackson
John T. Faig
Harold S. Falk
C. F. Bailey
S. S. Edmands } *Associate-Members.*
W. S. Conant }

LIBRARY

The Library Committee represents the A.S.M.E. on the Library Board of the Engineering Societies Library. The report of the Library Board for 1930 shows that there has been no lessening in the use of the library during the depression, but that, on the contrary, there has been a slight increase.

More than forty thousand members of the Founder Societies used the Library during 1930. Forty per cent of these were non-visitors, assisted by loans of books, photoprints, and searches. Twenty per cent more photoprints were called for than in 1929.

Gifts from members of the Founder Societies and other friends numbered 12,517 books and pamphlets. The main collection was increased by 7418 volumes and now contains 139,916 volumes and pamphlets.

During the year a new bookstack was erected, which eliminated serious overcrowding and provided some room for growth.

At the annual meeting of the Library Board in January, 1931, the Chairman of your Library Committee was elected Chairman of the Library Board.

Respectfully submitted,

Alten S. Miller, *Chairman*
George F. Bateman
E. P. Worden
Walter M. Keenan
Calvin W. Rice, *Secretary.*

RESEARCH

The Standing Committee on Research reports 29 cooperative research committees wholly or jointly sponsored by the Society on October 1, 1931. Two new committees were organized during the year, one on rotary drilling of oil wells and the other on critical pressure steam boilers.

It is gratifying to note that although a severe business depression existed throughout the year the total contributions from industry and other sources to the various committees and the expenditures by these committees were slightly greater than those of the previous record year—$53,000 being contributed and $51,000 spent. This money covered the salaries of 25 research workers on whole or part time at 10 institutional laboratories, and materials and special equipment in several cases.

Committee Activity

The personnel of the 29 committees totals 445 men, 45 per cent of whom are non-members of the Society. Most of these committees have held one or more business meetings during the year; in all a total of 43 meetings were held. In addition, they have sponsored either alone or jointly 10 technical sessions for the presentation of their reports and related papers at meetings of the A.S.M.E. and other societies.

Experimental investigations are being financed either in whole or in part at the following institutions: National Bureau of Standards, Washington, D. C., and Columbus, Ohio; U. S. Bureau of Mines, Pittsburgh, Pa.; Massachusetts Institute of Technology; University of Michigan; Union College; University of Illinois; Ohio State University; University of Minnesota; and Carnegie Institute of Technology. Several field investigations have also been under way. Many more institutional and industrial laboratories have been cooperating with the committees without financial assistance.

Financial Record

Contributions to the various programs from industry, Engineering Foundation, and similar sources have totaled more than $53,000 this year as against $48,000 last year. The committees have thus continued to maintain 25 full- and part-time research workers on their programs. Expenditures for this work amounted to $51,000, somewhat more than last year. The Society, in addition, contributed about $14,000 for administrative and secretarial assistance this year toward the organization, maintenance, and development of this cooperative research work. It should be noted that the ratio between industrial contributions and Society contributions still continues to be better than three to one.

Publications

Technical progress reports on their work and related papers have been presented by the committees and published by the Society and other organizations throughout the year. These filled 39 pages of *Mechanical Engineering* and 387 pages of the Transactions alone. As was done last year, these papers are being bound together in a single volume for sale under the title "A.S.M.E. Research Reports and Papers, 1931." In addition, the following separate Research publications developed by the committees were published for sale by the Society:

Fluid Meters, Part 2—Description of Meters
Dynamic Loads on Gear Teeth
Bibliography on Deterioration of Condensing Equipment
Supplement to a Bibliography on the Effect of Temperature on the Properties of Metals.

The Bibliography on Riveted Joints being prepared by A. E. R. deJonge for the industry is in printer's proof form and now comprises over 500 selected and abstracted references.

Prof. W. M. Coates reports that the manuscript of the bibliography and monograph on the Stress Analysis of Pressure Vessels is about completed.

Items of current interest on the activities of the various committees have appeared from time to time in the *A.S.M.E. News*.

Approved Research Publications of the A.S.M.E. The following research publications have been issued by the Society to date:

Bibliography on Riveted Joints
Bibliography on Mechanical Springs
Report on Fluid Meters, Part 1—Their Theory and Application (Third Edition)
Bibliography on Effect of Temperature on Properties of Metals
Bibliography on Woods of the World—Exclusive of the Temperate Region of North America and with Emphasis on Tropical Woods
Bibliography on Cutting of Metals
Fluid Meters, Part 2—Description of Meters
Dynamic Loads on Gear Teeth
Bibliography on Deterioration of Condensing Equipment
Supplement to a Bibliography on the Effect of Temperature on the Properties of Metals.

International Cooperation

The Research Committee can report considerable progress in bringing about closer contact between the Society's research committees and similar groups abroad, particularly through the Institution of Mechanical Engineers and the Department of Scientific and Industrial Research in England and through the Verein deutscher Ingenieure in Germany. Reports of collateral work are exchanged regularly, and the men active in these fields are being brought closer together through international conferences as well as personal visits. These contacts are profitable to the members of the A.S.M.E. as well as to those of the foreign societies in many ways.

Committee Reports

Lubrication. A. E. Flowers, Chairman. The experimental studies at the Bureau of Standards have been continued during the year with three full-time and several part-time workers in the employ of the

Committee, an arrangement made possible by contributions totaling more than $4000 from industry and Engineering Foundation. These studies include (1) further investigation of viscosity under high pressure, (2) journal bearing investigations, and (3) oiliness research. Three other laboratories have cooperated with the committee, namely, (*a*) Pennsylvania State College, on experimental test of hydrodynamic theory of lubrication, (*b*) Kingsbury Machine Works, on pressure distribution in oil films, and (*c*) Research Department of Westinghouse Electric and Manufacturing Company, on waste-packed bearings, ring oiling, and forced lubrication.

A direct result of this activity was the Annual Meeting session held on December 5, 1930, at which Mr. Albert Kingsbury read his paper "On Problems in the Theory of Fluid-Film Lubrication With Experimental Method of Solution," and the Fifth Report of the A.S.M.E. Special Research Committee on Lubrication was presented.

Preparations for a similar session at the 1931 Annual Meeting are now in progress, and it is known that papers on "Optimum Conditions in Journal Bearings" will be read by Mr. Kingsbury and on "Pressure Distribution in Oil Films of Journal Bearings" by Messrs. S. A. McKee and T. R. McKee.

The Chairman published a paper on "Modern Lubrication Studies" in the May 26, 1931, issue of *Power*.

Mr. L. J. Bradford, a member of the Committee and Professor of Machine Design, Pennsylvania State College, read a paper on "Bearing Design in the Light of Oil-Film Pressure Investigation" at the Lubrication Meeting of the Lubrication Engineering Committee of the A.S.M.E. Petroleum Division, State College, Pa., May, 1931. He will also present a paper on "Oil Film Pressure" at the coming Annual Meeting.

In July, 1931, the Committee was represented at the International Congress on Lubrication, Strasbourg, France, by Mr. Winslow H. Herschel, a member of the staff of the Lubrication Section of the National Bureau of Standards.

Fluid Meters. R. J. S. Pigott, Chairman. "Fluid Meters, Part 2, Description of Meters," has been completed and published by the Society. Draft of Part 3, on the "Influence of Installation," was fully discussed by the Committee at its meeting on September 24, in Pittsburgh, Pa., and will be revised further.

The day following this meeting, the members of the Committee met with the Gas Measurement Committee of the American Gas Association. They discussed the revision of the table of coefficients now employed by the American Gas Association and arranged for joint review and promulgation of future tables.

During the summer months a subcommittee was organized in the Tulsa area to conduct tests to determine the accuracy of existing coefficients and correction factors for viscosity, gravity, and temperature in the measurement of oil. Two meetings have been held. A program has been laid out and contributed equipment is now being set up at the mechanical engineering laboratory of the University of Oklahoma under the direction of Prof. W. H. Carson. A delegation from the A.S.M.E. Special Research Committee on Fluid Meters attended the annual Southwestern Gas Measurement Course held at the University of Oklahoma in Norman, Okla., April 21–23.

The Special Research Committee on Fluid Meters will hold a session at the 1931 Annual Meeting of the Society. Two papers have been prepared for presentation at this session.

Thermal Properties of Steam. Alex Dow, Chairman. Throughout the year, the experimental part of the Committee's program was continued at the Massachusetts Institute of Technology under Dr. F. G. Keyes and Mr. L. B. Smith, and at the Bureau of Standards under the immediate direction of Dr. N. S. Osborne with Messrs. H. F. Stimson, D. C. Ginnings, and E. F. Fiock assisting.

Dr. Keyes and Mr. Smith have spent the greater part of the year designing and constructing a new stainless-steel bomb with which to make check measurements on the volume of saturated and superheated steam. In a similar way the experimenters under Doctor Osborne have designed and completed the construction of a new steel calorimeter. Early in the coming year it is expected that measurements of saturation pressures and temperatures up to 2500 lb. per sq. in. will be begun with this apparatus, and the design of a new flow calorimeter for superheat measurements will be started.

No international meeting was held this summer, but correspondence has passed between members of the Committee and the laboratories in Great Britain, Germany, and Czechoslovakia which are at work on this problem and others interested in the results attained. One of the subjects of discussion was the date of the Third International Steam Conference, which is to be held in the United States. The date finally decided upon is September 12–17, 1932.

The usual Steam Tables Research Session was held during the 1930 Annual Meeting of the Society at which papers and reports were presented by Messrs. Keyes, Smith, Osborne, and Keenan.

In September, 1931, the Society published an inexpensive abridgment of the Keenan "Steam Tables and Mollier Diagram," intended particularly for the use of engineering students and publishers desiring to reproduce these tables in abstract.

Strength of Gear Teeth. Ralph E. Flanders, Temporary Chairman. The main Research Committee appointed Mr. R. E. Flanders to the chairmanship to fill the vacancy caused by the death of Mr. Wilfred Lewis.

"Dynamic Loads on Gear Teeth," a complete report of the Committee's investigation at the Massachusetts Institute of Technology during the past five years, was finished during the year by Prof. Earle Buckingham and published in pamphlet form by the Society, in July, 1931.

Cutting of Metals. F. C. Spencer, Chairman. This has been a year of reorganization and readjustment for this special research committee. At its December, 1930, meeting, an executive committee consisting of the three officers was created, and chairmen of subcommittees on Bibliography and Correlation, Cutting Fluids, Metal Cutting Materials, Physics of Cutting Metals, and Properties of Materials, were appointed or reappointed.

The Subcommittee on Cutting Fluids presented a preliminary report prepared by Prof. O. W. Boston, who is in charge of a part of this investigation in progress at the University of Michigan. This report, in the form of a paper, was presented at the Hartford Meeting of the Society in June, 1931. At this session Mr. T. G. Digges of the Bureau of Standards also read a paper, entitled "The Influence of Chemical Composition on Heat Treatment of Steel Forgings on Machinability With Shallow Lathe Cuts."

Chairman Nichols spent three months in Europe during the spring and summer. While there, he visited a number of manufacturing plants in Germany and gathered data for the Committee. Before sailing, he resigned the chairmanship of the Committee in favor of Mr. F. C. Spencer.

Before the close of the fiscal year additional funds were made available to the Subcommittee on Cutting Fluids, Mr. Carl Oxford, Chairman, to permit the continuance of the experimental program at the University of Michigan under Professor Boston.

At the December, 1931, Annual Meeting of the Society, the Committee will hold a joint session with the Machine Shop Practice Division at which a report of the Subcommittee on Metal Cutting Materials will be presented.

The Committee has caused to be translated, and is revising for publication by the Society, a book by Prof. K. Gottwein on "Kühlen und Schmieren bei der Metallbearbeitung."

A comprehensive Bibliography on Milling, being compiled by Prof. J. D. Hoffman at Purdue University, is nearing completion.

Mechanical Springs. J. R. Townsend, Chairman. The chairman and secretary visited personally the prominent spring manufacturers to determine the scope of their spring problems and to ascertain what the Committee might do to help solve them. This survey brought out the need for more information on the fatigue qualities of full-size springs, for the determination of correct spring constants for those materials most widely used in springs, and for the standardization of a new code of design employing these spring constants. Two of the large spring manufacturers have offered their laboratories for the fatigue tests. The program covering railroad-size helical springs has been completed and tests have been begun in these laboratories. A similar program of fatigue tests on wire springs is being prepared. In December, 1930, and in June of this year, Prof. M. F. Sayre reported before meetings of the Society the progress of his investigation on certain characteristics of spring materials, which is being financed by the Committee at Union College. Prof. J. B. Reynolds, at Lehigh University, has continued his investigations of conical springs and will present a report in December, 1931. The Committee's bibliography of current springs literature has continued to be distributed to members and financial contributors.

Effect of Temperature on the Properties of Metals. H. J. French, Chairman. (Joint Committee with A.S.T.M.) The outstanding contribution of this Committee for the year was the Symposium on the Effect of Temperature on the Properties of Metals held during the June, 1931, Annual Meeting of the American Society for Testing Materials in Chicago. This symposium amounted to an authoritative statement of the engineering needs for metals to withstand high temperatures and the properties of the available metals. Some fifty authors contributed 28 papers. These papers with discussion will be printed in an 800-page bound volume and sold by the two societies. Additional papers were contributed by the Committee to the meetings of the two societies during the year. The A.S.M.E. also published for the Committee an up-to-date "Supple-

ment to the Bibliography on the Effect of Temperature on the Properties of Metals."

Prof. H. F. Moore of the University of Illinois has submitted a report of fatigue tests on a low-carbon steel at high temperatures, an investigation financed by the Committee. Progress has been made by the various industrial laboratories cooperating with the Committee in studies of high-temperature effects on mechanical properties and structural stability of cast, wrought, and stainless steel. Progress can also be reported in the development of a test code for high-temperature tension tests.

Nearly half of the required amount of $20,000 has been subscribed by the National Electric Light Association, the American Petroleum Institute, the Engineering Foundation, and a group of alloy casting manufacturers to the support of a proposed enlarged research program for the Committee.

Condenser Tubes. A. E. White, Chairman. The Committee's progress report on the study of the effect of air both in the entrained and dissolved condition upon condenser-tube deterioration was presented at the Annual Meeting, December, 1930. Under the direction of Mr. B. Houghton, Vice-Chairman of the Committee, a number of examinations have been made of these conditions in the condensers of several steam generating stations according to a procedure set up by the Committee. Further experimental work has also been carried on for the Committee.

A comprehensive bibliography of the causes and prevention of corrosion and other forms of deterioration in condenser tubes was presented to the Society and published in pamphlet form during the year. The Committee will present a progress report at the 1931 Annual Meeting.

Boiler Feedwater Studies. S. T. Powell, Chairman. (Joint Committee with A.B.M.A., A.R.E.A., A.S.T.M., A.W.W.A., and N.E.L.A.) The Committee, resuming activities after the recovery of the Chairman from serious illness, has concentrated on research studies on priming and foaming of boiler waters, and on the standardization of water analysis applicable to boiler feed and other industrial uses.

The Committee, in April, 1931, completed negotiations whereby a fellowship was established at the Ohio State University under the direction of Prof. C. W. Foulk, to continue the studies on priming and foaming of boiler waters. Professor Foulk will present a paper at the Boiler Feedwater Session at the December, 1931, Annual Meeting of the Society.

Subcommittee No. 8 on the Standardization of Water Analysis Methods has been reorganized under the chairmanship of Mr. C. H. Fellows, and close contact has been established with similar groups in the American Society for Testing Materials, the National Electric Light Association, and the American Water Works Association. To facilitate the work of formulating standard test methods, the Committee has established a fellowship at the University of Michigan. Mr. Fellows' Subcommittee will present at the coming Annual Meeting a progress report including studies of methods for determining alkalinity of boiler water.

Report No. 21 of the Subcommittee on Bibliography was distributed in May, 1931, to Committee members and to contributors.

Boiler Furnace Refractories. W. A. Carter, Chairman. The Committee held a session at the 1930 Annual Meeting of the Society at which papers were presented on "Comparative Resistance of Refractories to Coal-Ash Slags," by R. K. Hursh, "Action of Slags on Firebrick and Boiler Furnace Settings," by T. A. Klinefelter and E. P. Rexford, and "Furnace-Gas Compositions and Temperatures in Underfeed-Stoker-Fired Boilers and Their Effect on Boiler Settings," by Albert C. Pasini and Edward M. Sarraf. The authors of the third paper are not members of the Committee.

The U. S. Bureau of Mines published Bulletin No. 334 on "A Study of Refractories Service Conditions in Boiler Furnaces"—a summary of papers and reports of work done for the Committee during the past five years. Copies of this publication were distributed to the members of the Committee and to the contributors.

A special report on "The Effects of Cahokia, Trenton Channel, and Lowellville Slags on Their Respective Bricks Under Reducing and Oxidizing Atmospheres," by E. P. Rexford and E. J. Vachuska also was distributed to the same groups for their information.

Tests have been continued at the Columbus Branch of the National Bureau of Standards by E. P. Rexford, A.S.M.E. Research Associate, with the assistance of Mr. Vachuska, under the direction of Mr. T. A. Klinefelter, superintendent of the station. These include an investigation of power-house refractories, studies of samples from the Connors Creek Station, Roseville Brick Plant, Deepwater Installation, and others; studies on the multiple-component system; on power-house samples; on firing brick and slag samples in reducing atmospheres; and on the effects of reducing atmospheres on the slag action on firebrick.

The laboratory slag tests at the Battelle Memorial Institute include reports on various refractories (ST1 to ST8) with Lowellville, Cahokia, Chicago, Nanticoke, Peoria, and Detroit coal ashes. These tests were reported monthly by Mr. H. E. Simpson, and mimeographed copies have been distributed to the members of the Committee. The Institute is supporting this phase of the work. Similar tests have been made by another method at the University of Illinois by Prof. R. K. Hursh, and mimeographed copies of his reports also have been sent to the members of the committee.

Elevator Safeties. M. H. Christopherson, Chairman. Tests on undercar safeties have occupied the time of the Committee's research associates at the Bureau of Standards during the year. Special test apparatus, installed in the test hoistway, was further refined during use.

The bi-monthly meetings of this Committee were continued for the purpose of reviewing the test data and preparing recommendations for use in the revision of the Safety Code for Elevators.

In July, 1931, the Committee sponsored a conference of wire rope manufacturers and users to discuss the possibility of establishing standards for inspection and condemnation of ropes in service. The program of the A.S.M.E. Wire Rope Research Committee was adopted as a method of procedure.

Worm Gears. Earle Buckingham, Chairman. Present industrial conditions have materially slowed down the progress of much of this Committee's program. Tests and other work are now being carried on in three or four plants, but the test data available is not yet sufficient to justify any further report at this time.

Welding of Unfired Pressure Vessels. E. H. Ewertz, Chairman. (Joint Committee with the American Welding Society.) A report of tests on a certain type of light-gage welded tank at Lehigh University has been made. The results obtained have been incorporated in the A.S.M.E. Boiler Code. The Committee is now considering the desirability of disbanding, as its work appears to have been completed.

Saws and Knives. A. S. Kurkjian, Chairman. Owing to the lack of financial support from the woodworking industries and manufacturers of saws and knives, this Committee has been inactive during the year.

In March, 1930, Engineering Foundation made a grant of $1000 toward the Committee's proposed experimental program, but because of the Committee's inactivity, $500 yet due on the grant was transferred to the Lubrication Committee to help meet its current research expenses.

Existing Supplies of Hardwoods. Additional sample logs of 40 different tropical species were received in New York from Central America, British Guiana, Colombia, Venezuela, and Panama. They were cut into test blanks by two importing concerns and shipped to the University of Michigan for tests.

A representative of the Tropical Plant Research Foundation, with the aid of the Pack Forest Trust and the A.S.M.E., made a personal canvass of principal wood-using and importing concerns in the United States in an effort to interest them in contributing to an investigation of the abundance and qualities of the general utility woods of the northern portion of the American tropics. He reports no success because of existing business conditions.

Strength of Vessels Under External Pressure. W. D. Halsey, Chairman. The Committee completed its survey of foreign and domestic design practice and performance records on jacketed vessels, and compiled it into useful form. At the December, 1930, meeting of the Committee, Mr. D. B. Wesstrom, member of the Committee, presented an analysis of this data in the form of tables and comparative curves which indicated that a large amount of experimental work needs to be done to verify existing formulas. Commander Saunders, member, also reported on extensive tests that have been carried on at the Model Basin, Navy Yard, Washington, D. C., on cylindrical structures subjected to collapse, and from which a new formula has been developed. He later discussed this work in a paper before the Society's Hartford Meeting, June, 1931. Mr. T. M. Jasper, member, reported at the same meeting on an investigation of the collapsing strength of steel tubes.

Absorption of Radiant Heat. W. J. Wohlenberg, Chairman. The Committee is continuing to collect and analyze the meager data available from the power industry on the absorption of radiant heat in installed furnaces.

Velocity Measurement of Fluid Flow. W. F. Durand, Chairman. The apparatus described in a previous report which required two identical sound-detecting devices suffered from the disadvantages of producing troublesome reflected sound waves. Professor Hartig has accordingly invented an improved apparatus which has the

inherent property of suppressing waves which ordinarily would give rise to disturbing reflections. A preliminary form of this new apparatus was set up and tested with moving air during the months of July and August, 1931, with encouraging results. The preliminary test suggested refinements and improvements which are now being incorporated in a more suitable design which is adapted for measuring the velocity of gases and liquids. It is planned to test the new apparatus by measuring the velocity of water, in connection with which adequate means for checking the magnitude of the velocity are available in the University of Minnesota Experimental Laboratory, where the work is in progress.

Measures of Management. W. E. Freeland, Chairman. The Committee conducted a conference at the Society's June Meeting in Hartford on the possibility of establishing formulas for the selection and replacement of manufacturing equipment. Some fifty manufacturing executives attended and brought out in the discussion the many practical elements that must be considered in developing such formulas.

The studies on this phase of the Committee's program are being carried on at the Massachusetts Institute of Technology, under the joint sponsorship of this Committee and of a similar committee of the Materials Handling Division. A report, "Economic Life of Equipment," by H. O. Vorlander and F. E. Raymond, will be presented at the December, 1931, Annual Meeting.

Diesel Fuel Oil Specifications. L. H. Morrison, Chairman. The Committee has not made much progress in formulating specifications or in research work during the year, due to reorganization and the entrance of allied technical organizations into the project. The formulation of specifications is to be handled by a Sectional Committee organized under the procedure of the American Standards Association under the sponsorship of the American Society for Testing Materials but based on data to be supplied by the A.S.M.E. Special Research Committee on Diesel Fuel Oil Specifications. A joint committee has been organized with the Society of Automotive Engineers to make a study of combustion in a Diesel engine cylinder. Efforts are now being made to raise funds from industry for this research. The A.S.M.E. Committee has in addition two subcommittees, one studying the physical and chemical properties of Diesel fuels in the laboratories of the larger oil companies in an attempt to discover the characteristics of motor fuel unsatisfactory for engine operation, and the other carrying out field tests in the shops of engine manufacturers in an effort to discover the effect of viscosity and Conradson carbon on the operation of Diesel engines.

Wire Rope. W. H. Fulweiler, Chairman. The Committee believes that a definite relation can be established between the number of broken wires and the wear of a wire rope and its remaining useful life. To test this opinion approximately 500 copies of the Committee's program and data sheets have been distributed to persons believed to be interested in collecting data on ropes in service and in sending samples from discarded ropes to the National Bureau of Standards, Engineering Mechanics Section, for testing. Arrangements have been completed with the Bureau of Standards for testing the samples and correlating the results of the tests with the collected data. Engineering Foundation is financing this phase of the investigation. In July, the Committee participated in a conference with wire rope manufacturers and users called by the Sectional Committee on the Safety Code for Elevators. The purpose of this conference was to discuss the possibility of framing rules for inspection and condemnation of ropes in service on elevators at the present time. Discussion revealed that a great deal of technical information is needed on wire rope life in service before such rules can be established. The program of the Wire Rope Committee was adopted as a method of procedure in obtaining such data.

Airplane Vibration With Special Reference to Instruments. Alexander Klemin, Chairman. The Committee has been inactive since the presentation of its report entitled "Airplane Instrument Vibration" at the December, 1930, Annual Meeting.

Heavy Duty Anti-Friction Bearings. W. Trinks, Chairman. Several concerns have cooperated with the Committee in the conduct of extensive tests on rolling mills. Three reports have been distributed during the year, in mimeographed and blueprinted form, to the members of the Committee and to contributors: these were (1) forces in the cold rolling of non-ferrous metals, (2) an analysis of German data on the forces which occur in the hot rolling of mild steel, and (3) analyses of data on roll neck forces which have been obtained in enormous quantities from the 35-in., 28-in., and 22-in. mills of the Timken Steel and Tube Company in hot rolling alloy steels.

The Committee held two meetings during the year to discuss the program and the progress made. Arrangements have been completed for a joint session with the Iron and Steel Division at the 1931 Annual Meeting at which several papers resulting from the Committee's activity will be presented.

Removal of Ash as Molten Slag From Powdered-Coal Furnaces. P. B. Rice, Chairman. Experimental investigations, under the auspices of this Committee at the Bureau of Mines, on the addition of fluxes as a means of removing ash as molten slag from powdered-fuel furnaces, were completed during the year, and a report will be presented at the December, 1931, Annual Meeting of the Society. A study is now being made of the selective separation of ash in slag furnaces. Constructors, manufacturers, and operators of power plants and power-plant equipment are financing the Committee's program.

Automatic Oil Pipe Line Pumping Stations. W. G. Heltzel, Chairman. The Committee's experimental station was completed in time for demonstration during the seventh International Petroleum Exposition in Tulsa, Okla., October, 1930. As a result of this investigation one of the more prominent oil distributing companies has during the year built six similar stations into its system.

The Committee has recently completed a survey of the functioning of these stations from which data will be used to redesign and rebuild the experimental station along greatly improved lines.

Methods and Apparatus for Noise Measurement. W. B. White, Chairman. The Committee is confining itself to the development of an objective scale for measurement and apparatus for the determination of sound values in a manner universally acceptable. A résumé of the Committee's first report appeared in the May, 1931, issue of *Mechanical Engineering.*

This committee participated with other interested organizations in the symposium on sound measurement conducted by the North Eastern District of the American Institute of Electrical Engineers at Rochester, N. Y., April 29–May 2, 1931.

The project has developed steadily and the general facts and ideas accumulated have been circulated to the committee members. In the spring the Committee expects to submit a report which will contain a tentative, well-considered formula.

Rotary Drilling of Oil Wells. D. L. Trax, Chairman. Organized originally in January, 1930, to investigate the operating efficiencies of steam equipment used in the rotary drilling of oil wells. The Committee's scope was later enlarged to include a study of electrical, gas engine, and Diesel-engine-driven equipment designed for the same purpose.

The study of steam drilling was made first, and one of the oil companies in the Oklahoma field made it possible for the Committee to conduct its tests during the drilling of a new well. Special metering and other apparatus necessary for the investigation were contributed by local concerns, and the tests were all directed by Prof. W. H. Carson, Head of the Department of Mechanical Engineering of the University of Oklahoma. The work was completed during the summer months and an extensive report has been prepared for publication.

Critical Pressure Steam Boiler. A. A. Potter, Chairman. This new committee was organized in August, 1931, to advise with the Engineering Staff of Purdue University in a study of the action of steam in a boiler at and above the critical pressures. Tests have been conducted on a 3500 lb. experimental steam generator and auxiliaries recently installed in the Mechanical Engineering Laboratory of the University with the cooperation of several manufacturers and users of steam-generating equipment. This investigation is expected to supplement in a practical way the program of the Special Research Committee on Thermal Properties of Steam. A paper by members of the Committee will be presented at the 1931 Annual Meeting of the Society. This will describe results of preliminary investigations including automatic control, pressure-temperature relations, and heat transfer.

Physical Constants of Refrigerants. (Joint sponsorship with American Society of Refrigerating Engineers. No personnel yet appointed.) Owing to the depressed condition of business, efforts to collect funds from industry to support a proposed research at the Bureau of Standards on the determination of the physical constants of certain refrigerants were not successful. The project is therefore being held in abeyance till conditions improve.

Respectfully submitted,

W. Reuben Webster, *Chairman*
Walter H. Fulweiler
Alex D. Bailey
Andrey A. Potter
F. C. Biggert, Jr.

STANDARDIZATION COMMITTEE

The A.S.M.E. Standardization Committee presents the following as its report for the year October 1, 1930, to October 1, 1931.

Organization

During the past year the Society accepted one sole sponsorship, that for the Standardization of Specifications for Leather Belting, January, 1931, and one joint sponsorship with the American Foundrymen's Association for the Standardization of Foundry Equipment, February, 1931. To date the Society has accepted sponsorship or joint sponsorship for a total of 29 sectional committees under the procedure of the American Standards Association. These sectional committees have subdivided themselves into 238 subcommittees and subgroups in order that the work assigned to them may be carried forward more promptly and effectively. The total number of members of these committees is now 1090, of which 397 are members of the A.S.M.E.

Organization meetings of the following three Sectional Committees were held: Pressure and Vacuum Gages in December, 1930; Specifications for Leather Belting in May, 1931; and Foundry Equipment in May, 1931. The reorganization meeting of the Sectional Committee on Plain Limit Gages for General Engineering Work was held in December, 1930, at which time it was decided to change the name to Allowances and Tolerances for Cylindrical Parts and Limit Gages. In all, 81 standards committee meetings were held during this year.

Standards Approved

Four standards were completed under the A.S.A. procedure and published in pamphlet form this past year. These are the standards for 25 Lb. Cast Iron Flanges and Flanged Fittings; Track Bolts; Woodruff Keys, Keyslots, and Cutters; and Symbols for Heat and Thermodynamics.

Publicity

During the year 33 pages of text pertaining to the Society's work in standardization were published in *Mechanical Engineering*. Part of this material was published in the Engineering and Industrial Standardization section of *Mechanical Engineering*, and the remainder consisted of the reproductions of preliminary proposals issued from time to time by the committees. Thirty-two hundred copies of tentative draft standards in printer's proof form were distributed to consumer and producer interests and to the technical press during the year.

Mechanical Standards Advisory Council

The Mechanical Standards Advisory Council, organized in December, 1928, to advise this industry and the American Standards Association on questions relative to the initiation and development of dimensional standards and material specifications of special interest to these groups, held a meeting in December, 1930. At this meeting it was voted to increase the personnel of the Executive Committee from seven to nine members; this vote included the recommendation that three members be elected each year for a term of three years.

Since this meeting the new executive committee has been elected by letter ballot and now consists of the representatives of the American Gas Association; American Society for Testing Materials; American Society of Mechanical Engineers; American Institute of Bolt, Nut and Rivet Manufacturers; Heating and Piping Contractors National Association; Manufacturers' Standardization Society of the Valve and Fittings Industry; National Electric Light Association; National Machine Tool Builders Association; and the Society of Automotive Engineers. The Council now consists of 24 member organizations.

Sectional Committee and Subcommittee Reports

Shafting. C. M. Chapman, Chairman. In January, 1931, favorable reactions were received from the committee members on the proposed extension of the shafting diameters and flat key series for shafts larger than 6 in.

The four existing standards, namely, Cold Finished Shafting with an addition covering recommended diameters for forged shafting larger than 6 in., Square and Flat Stock Keys (1925), including the extension for large flat keys, Plain Taper Stock Keys (1927), and Gib-Head Taper Stock Keys (1927) were then tentatively combined under a single cover with appropriate introductory notes and title.

During the review of the combined standard by the committee members certain questions were raised in connection with the larger sizes of shafting and keys, and in order to obtain authoritive information on these points a questionnaire was distributed to the larger manufacturers and users of this material. The answers to this questionnaire and their bearing on the proposed extension to the shaft and key sizes will be discussed at the next meeting of the Sectional Committee, scheduled for the third week in October.

Subcommittee on Woodruff Keys. L. C. Morrow, Chairman. The American Standard for Woodruff Keys, Keyslots, and Cutters was approved by the American Standards Association in December, 1930, and is now available in pamphlet form.

Allowances and Tolerances for Cylindrical Parts and Limit Gages. E. J. Kearney, Chairman. The reorganization meeting of the Sectional Committee on Plain Limit Gages for General Engineering Work was held in December, 1930, letters having been sent to 62 interested organizations requesting them to appoint representatives. The appointments made up to the time of the meeting were discussed and several persons were added as "members at large," bringing the personnel to a total of 43 members. Mr. C. E. Rundorff, of the Buick Motor Company, Flint, Mich., was elected secretary.

After reviewing the work of the original Committee and the extent to which the original standard had been accepted by industry, several alternative tolerance systems were discussed.

The plan and scope of the Committee's work were analyzed and it was recommended that certain changes be made in the wording of the "Scope." It was also recommended that the Committee's name be changed to "Sectional Committee on the Standardization of Allowances and Tolerances for Cylindrical Parts and Limit Gages."

The appointment of subcommittees to facilitate handling the various phases of the work was left to the discretion of the chairman, who has been in touch with the members regarding a plan for the activities of the Committee, and it is expected that another meeting will be held before the end of the calendar year.

Ball and Roller Bearings. F. W. Gurney, Chairman. The American Standard on Ball and Roller Bearings of the Single Row Type and Recommended Practice for the Wide Type was completed by this Sectional Committee and approved by the American Standards Association in April, 1930. The Standardization of Metric Thrust and Roller Bearings is being contemplated by the Committee as its next activity.

Gears. B. F. Waterman, Chairman. While no tangible results in the form of American Standards have been produced this year by the Sectional Committee on the Standardization of Gears, several of its subcommittees have made considerable progress toward this end.

Subcommittee on Nomenclature. E. W. Miller, Chairman. A decision was reached at the September, 1930, meeting of the A.G.M.A. Committee on Nomenclature that the series of multiple letter symbols be designated as "abbreviations" and that a series of single letter symbols be developed for inclusion in the proposed recommended practice of the A.G.M.A. covering symbols, abbreviations, and definitions. During the year the A.G.M.A. Committee and this Subcommittee prepared a proposal based on this idea. Copies of the latter were generally distributed for criticism and comment. To coordinate its work with the recommendations of the Sectional Committee on Scientific and Engineering Symbols and Abbreviations three members of the latter Committee are to be appointed to confer with the Subcommittee on Nomenclature.

Subcommittee on Spur Gear Tooth Form. H. J. Eberhardt, Chairman. The letter ballot of the Sectional Committee on the Proposed American Standard for the Tooth Form of the 14½ Deg. and 20 Deg. Full Depth Involute System was completed during August. Chairman Eberhardt is now reviewing the comments received with these ballots, after which it will be submitted to the A.G.M.A. and A.S.M.E., the sponsor organizations, for approval. This standard when finally approved by the A.S.A. will be combined and printed in one pamphlet with the first standard prepared by the Subcommittee, covering the tooth form of the stub tooth systems.

Subcommittee on Helical Gears. A. A. Ross, Chairman. During the past year agreement on the fundamentals of its report on herringbone gears, with the exception of tooth load formulas, was reached by the Subcommittee. The subject of tooth load formulas is receiving careful consideration but it will be necessary for the Subcommittee to await the completion of certain tests at the Massachusetts Institute of Technology before definite recommendations on this part of the subject can be made. One informal meeting of several members of the Committee was held in Buffalo at the time of the 1931 meeting of the A.G.M.A.

Subcommittee on Inspection of Gears. G. L. Markland, Chairman. The Subcommittee's present proposal is in printer's proof form, dated March, 1929. It was based largely on the recommended practice of the A.G.M.A. Recently the A.G.M.A. Committee on this subject has made a careful review of this material

and in turn proposed a redraft at the May, 1931, meeting of the Association. This revised proposal will be given due consideration by the Subcommittee on Inspection.

Subcommittee on Materials. T. D. Lynch, Chairman. In April, 1931, Mr. Chester B. Hamilton, Chairman of the A.G.M.A. Committee on Gear Materials, advised Chairman Lynch that his Committee had approved, with a few minor changes, the proposed American Standard for Gear Materials and Blanks developed by this Subcommittee. Chairman Lynch is, accordingly, drafting a finally revised proposal.

Pipe Flanges and Fittings. C. P. Bliss, Chairman. The proposed Standard for 25 Lb. Cast Iron Pipe Flanges and Flanged Fittings approved in 1930 by this Sectional Committee and the Sponsors was designated by the A.S.A. as an American Standard in April, 1931. In July, 1931, the proposed standard for 800 Lb. Hydraulic Cast Iron Pipe Flanges and Flanged Fittings was submitted to the A.S.A. for approval and designation as an American Standard. The group of this Sectional Committee's published standards are: 250, 400, 600, 900, and 1350 Lb. Steel Pipe Flanges and Flanged Fittings; 125 and 250 Lb. Cast Iron Screwed Fittings; 150 Lb. Malleable Iron Screwed Fittings; 125 and 250 Lb. Cast Iron Long Turn Sprinkler Fittings; 25 Lb. Cast Iron Pipe Flanges and Flanged Fittings; 125 Lb. Cast Iron Pipe Flanges and Flanged Fittings; 250 Lb. Cast Iron Pipe Flanges and Flanged Fittings. In accordance with action taken at the meeting of the Sectional Committee on December 5, 1930, an additional Subcommittee, No. 9 on Port Openings, was appointed. The Chairman reports progress on the various subcommittees as follows:

Subcommittee No. 1 on Cast Iron Flanges and Flanged Fittings. A. M. Houser, Chairman. The proposed American Standard for 800-Lb. Hydraulic Cast Iron Pipe Flanges and Flanged Fittings, approved last year by the Sectional Committee, was submitted in July, 1931, to the American Standards Association. This action was taken after final agreement had been reached on the paragraphs pertaining to "Marking" and "Materials." These paragraphs were revised at the Sectional Committee meeting held in December, 1930, to meet the wishes of the Manufacturers Standardization Society of the Valve and Fittings Industry.

The standard for 25 Lb. Cast Iron Pipe Flanges and Flanged Fittings prepared by this Subcommittee and submitted to the American Standards Association in September, 1930, was approved and designated as an American Tentative Standard in April, 1931.

A first draft of a proposed American Standard for Ammonia Pipe Flanges and Flanged Fittings was approved as a recommended practice by the Refrigerating Machinery Association at its spring meeting, in May, 1931. Since then it has been re-edited and will be submitted as a preliminary draft to the Subcommittee upon receipt of approval of certain editorial changes from the Refrigerating Machinery Association.

Subcommittee No. 2 on Screwed Fittings. S. G. Flagg, Jr., Chairman. The proposed standard for Pipe Plugs of Cast Iron, Malleable Iron, Cast Steel and Forged Steel distributed in October, 1930, to industry for criticism and comment was discussed at the December, 1930, meeting of the Sectional Committee. At that time it was referred back to Subcommittee No. 2 for further consideration in the light of the comments received. It is expected that the Subcommittee will meet for this purpose at an early date.

Subcommittee No. 3 on Steel Flanges and Flanged Fittings. C. P. Bliss, Chairman. During the year this Subcommittee held two meetings, one in October, 1930, and the second in April, 1931. Difficulty was experienced in setting the "pressure-temperature" ratings for the 150 lb. standard, with the result that action was postponed pending further investigation by the Manufacturers Standardization Society of the Valve and Fittings Industry and the Committee members. The 250 lb. standard was rerated to 300 lb., with the metal thickness of the fittings increased where necessary, all other dimensions remaining unchanged. The 1350 lb. standard was rerated to 1500 lb., with increased metal thickness and, in sizes over 4 in., revised flange thicknesses. Before final action is taken the revised proofs are being submitted to the A.P.I. and N.E.L.A., and their comments will be available in the near future.

Subcommittee No. 5 on Face-to-Face Dimensions of Ferrous Flanged Valves. F. H. Morehead, Temporary Chairman. No actual progress in the work of this subcommittee is reported this year. In January a further request for the delimiting of the scope of this Subcommittee's activities was expressed by the Manufacturers Standardization Society of the Valve and Fittings Industry. The scope now covers the standardization of face-to-face dimensions for certain types of cast iron valves in a range of sizes for pressures of 125 and 250 lb. and for certain steel valves for 150 and 300 lb. pressure.

Subcommittee No. 6 Malleable Iron or Steel Brass Seat Unions. C. P. Bliss, Temporary Chairman. The work of this Subcommittee during the past year has been limited to the consideration of the tolerances and allowances proposed by the A.R.A. Subcommittee on Locomotive Design and Construction. This question is still in the hands of a manufacturers' subgroup.

Subcommittee No. 8 on Marking of Pipe Fittings. W. S. Morrison, Temporary Chairman. Although a good response was received to the questionnaire on "Marking" sent out by this Subcommittee late in 1929, action has been postponed pending the results of the investigation of this subject being made by the Manufacturers Standardization Society of the Valve and Fittings Industry, which should be available shortly.

Subcommittee No. 9 on Port Openings. W. W. Hubbard, Chairman. This Subcommittee of six members was appointed in April, 1931, in accordance with action of the Sectional Committee at its meeting in December, 1930. Although it has not as yet held its first meeting, some progress toward definition of its scope of activity has been made through correspondence.

Bolt, Nut, and Rivet Proportions. A. E. Norton, Chairman. The Sectional Committee on the Standardization of Bolt, Nut, and Rivet Proportions through the activities of nine subcommittees has developed and approved seven standards which have received recognition as American Standards by the A.S.A. These are the standards for Small Rivets; Tinners', Coopers', and Belt Rivets; Wrench Head Bolts and Nuts and Wrench Openings; Slotted Head Proportions; Track Bolts and Nuts; Round Unslotted Head Bolts (Carriage, Step and Machine Bolts); and Plow Bolts.

Subcommittee No. 1 on Large and Small Rivets. J. E. Kiernan, Chairman. In March, 1931, the Chairman released a revision of the proposed American Standard for Large Rivets based on the criticisms received from the distribution of the draft dated March, 1929, to the members of the Subcommittee. The printer's type was accordingly corrected and revised page proofs were sent to the members of the Subcommittee in June, 1931, for approval, prior to submission to the Sectional Committee for approval by letter ballot. The Chairman expects to release this final draft shortly after October 1.

Subcommittee No. 2 on Wrench Head Bolts and Nuts. H. A. Spanagel, Chairman. The proposed revision of the American Tentative Standard, originally approved by the A.S.A. in February, 1927, has been completed during the year. Preliminary copies of the revised draft were distributed in May, 1931, to the members of this Subcommittee for their consideration and approval. With a few exceptions the members have indicated their approval, and this subject will be discussed at a meeting of the Subcommittee which is to be held at an early date. A joint subgroup, consisting of three members appointed by the Sectional Committee on Pipe Flanges and Flanged Fittings and three appointed by this Subcommittee, are studying the special requirements for bolt heads and nuts for use in flanged pipe connections. The personnel of this Committee includes Messrs. F. W. Martin, A. M. Houser, H. D. Summers, O. B. Zimmerman, J. H. Edmonds, E. Ward. No report has been received as yet from this Subgroup.

Subcommittee No. 9 on Socket Head Cap and Set Screws. H. Koester, Chairman. This Subcommittee held meetings in December, 1930, and April, 1931, to complete the preparation of its proposed Standard for Socket Head Cap and Set Screws, and since the April meeting several minor revisions have been made by correspondence. In September, 1931, this proposal was formally released by the Subcommittee and will be set in type at an early date for distribution to industry for criticism and comment.

Identification of Piping Systems. A. S. Hebble, Chairman. The present American Recommended Practice, "Scheme for Identification of Piping Systems," was developed by this Committee and approved by the American Standards Association in November, 1928. Since the publication of this "Recommended Practice" a need has been indicated for a detailed system of identification for powerhouse piping. The suggestion that such a system be developed was placed before the members of the Committee in March, 1931, for their consideration.

Small Tools and Machine Tool Elements. C. W. Spicer, Chairman. The Sectional Committee on the Standardization of Small Tools and Machine Tool Elements met in December, 1930, to review and discuss the problems of the 15 technical committees which are at work on the projects before the Sectional Committee. The standards of this group which have been approved by the A.S.A. and published by the A.S.M.E. are: T-Slots, Their Bolts, Nuts, Tongues, and Cutters; Tool Holder Shanks and Tool Post Openings; Milling Cutters; and Cut and Ground Thread Taps.

Technical Committee on T-Slots. E. Oberg, Chairman. The Cincinnati Milling Machine Company has submitted to the Tech-

nical Committee a suggestion that the depth of engagement of the present standard tongue be increased. Blueprints showing the maximum and minimum conditions and copies of the correspondence were sent to the members of the Committee. The Chairman now plans to hold a meeting in December for the purpose of discussing this question.

Technical Committee on Machine Tapers. E. F. DuBrul, Chairman. At the meeting held in October, 1930, it was decided that a layout for gages, including necessary drawings and dimensions for tapers based on present practice, be developed, and that consideration be given to the proposed tolerances on the large dimension shanks and holders. In April, 1931, the American Standards Association transmitted to the Committee, through its Chairman, a proposal from the International Standards Association concerning the advisability of postponing the development of an American Standard until an international standard had been evolved. The replies received reflected the opinion that it was difficult for European engineers to understand the vast expense which would be involved in a change in this country to the metric system of measurement. This made it impossible for the American Committee to consider any standardization on a metric basis.

Technical Committee on Spindle Noses and Collets for Machine Tools. E. F. DuBrul, Temporary Chairman. A revised draft of the proposed American Standard for Lathe Spindle Noses, dated March, 1931, was distributed by the Committee in April, 1931, for criticism and comment. This revision represents the work of the Subgroup on Correlation of Counter Proposals for Spindle Noses. A number of meetings of the Subgroup were held during the past twelve months to prepare this revised draft and to discuss the comments received. At the present time the testing of the new type of spindle nose under commercial conditions is in progress, and the Committee expects to report a definite recommendation within the next few months.

This activity has been correlated with the work of the Technical Committee on Chucks and Chuck Jaws in so far as it relates to the standardization of dimensions of the contact surfaces of the chuck and the spindle end.

Technical Committee on Designations and Working Ranges of Machine Tools. E. F. DuBrul, Temporary Chairman. The Committee held a meeting in New York in October, 1930, and decided first to direct its activities toward the standardization of the designations and working ranges of planers, engine lathes, cylindrical grinders, milling machines, and radial drills. The Committee's plans call for a canvass of the manufacturers of these machines for data on the designations they are now using and the working ranges of the machines of these types which they produce.

Technical Committee on Drill Bushings. C. J. Oxford, Chairman. In accordance with the action taken at the June, 1930, meeting of the Committee the recommendations contained in its Proposal "C" on Twist Drill Sizes and Lengths, distributed to industry in February, 1930, have been put into actual practice during the year by several large users of the product. Further action by the Committee on the proposed standard has been postponed, pending the results of this phase of the work which is still in progress. The criticisms and comments received from the distribution of Proposal "C" are being reviewed by the members of the Committee. Chairman Oxford states that in all probability the Committee will meet during the 1931 Annual Meeting of the A.S.M.E.

Technical Committee on Drill Jig Bushings. C. E. Rundorff, Chairman. Through correspondence during the year the Committee has reached a unanimous agreement on the dimensions and tolerances for headless and head press fit bushings, liner bushings, knurled head slip bushings, and nomenclature contained in its preliminary proposal, dated June, 1930. However, consideration is now being given to a suggestion for a change in the proposed hole size limits. As soon as this question is settled the proposed standard will be released for duplication and for distribution to a list of manufacturers, users, and interested individuals for criticism and comment.

Technical Committee on Punch and Die Holders. S. Diamant, Chairman. The Committee met in December, 1930, and in May, 1931. Reports on the two questionnaires which had been distributed by the Committee were made and a subgroup was appointed to study carefully the replies received and to draft a proposal based on them. The subgroup held one meeting in Chicago in January, 1931, and it is expected that a preliminary standard will be in form for distribution within a short time.

Technical Committee on Circular Forming Tools and Holders. W. C. Mueller, Chairman. The organization meeting of this Committee took place in March, 1931. At this meeting the Committee decided to prepare and distribute two questionnaires, one to cover the proportions of circular forming tools and the other to cover dovetailed circular forming tool blanks. These questionnaires were developed and submitted to the Committee in July, 1931, for review and comment prior to general distribution to the large users of circular forming tools and to the manufacturers of the machine tools in which circular forming tools are used.

Technical Committee on Chucks and Chuck Jaws. J. E. Lovely, Chairman. During the year this Committee held two meetings, the first in October, 1930, and the second in June, 1931. In March, 1931, two proposed standards, one on Rotating Air Cylinders and Adapters and the other on Chucks and Chuck Jaws, were distributed in proof form to industry for criticism and comment. A careful review of the comments was made and the proposed Standard for Rotating Air Cylinders and Adapters was approved, with minor revisions, for submission to the Sectional Committee. In September, 1931, the changes authorized were formally released by the Chairman for incorporation in a revised draft of the proposal. It was deemed advisable, however, to investigate further certain elements of the proposal for Chucks and Chuck Jaws, and a report of these studies will be made at the next meeting of this Committee.

Technical Committee on Cut and Ground Thread Taps. C. M. Pond, Chairman. Good progress has been made by this Committee during the past twelve months in the development of standards for special types of taps, such as ground thread pulley taps, cut and ground thread bent shank tapper taps, cut thread taper pipe taps, etc. These new standards when completed and approved will be combined with the present American Standard for Cut and Ground Thread Taps approved and published in April, 1930. One meeting of the Committee was held in December, 1930, and another has been scheduled for October, 1931. It is expected that following the coming meeting the Committee's proposal will be released for distribution to industry.

Technical Committee on Splines and Splined Shafts. C. W. Spicer, Chairman. In November, 1930, the Committee held a meeting in Detroit. Prior to the meeting copies of a number of manufacturers' data sheets on this subject were sent to the members for consideration. A review of this material and the discussion at the meeting indicated considerable divergence in present practices. Accordingly two subgroups were appointed to study the subject and to draft preliminary proposals; Subgroup No. 1 to cover splines as applied to machine tools and Subgroup No. 2 to cover splines as applied to automotive uses.

Technical Committee on Electric Welding Dies and Electrode Holders. J. A. Weiger, Chairman. This Committee held one meeting this year, in April, 1931. The attendance at the meeting was small and consequently little progress was made in the development of its preliminary proposal. The Chairman endeavored to hold a meeting during the Annual Meeting of the American Welding Society at Boston in September, 1931, but the time and place were not convenient to the majority of the members.

Technical Committee on Milling Machine Tables. G. A. Bouvier, Chairman. At its meeting in December, 1930, the Committee reached a tentative agreement on table widths for eight sizes from 3 in. up to and including 21 in. Agreement was reached also on the size and width of T-slots for these table widths and the slot spacing. Blueprints showing the proposed standard cross-sections of milling machine tables were distributed to the members of the Committee in March, 1931, and it is now actively at work on the further development of this proposed American Standard.

Scientific and Engineering Symbols and Abbreviations. J. F. Meyer, Chairman. During the year one additional standard was added to the list of standards previously completed and published by the Committee. Two more were passed through all but the final stages of the approval procedure. This year's accomplishments, therefore, practically complete the work originally assigned to this Sectional Committee. The eight standards so far approved by the A.S.A. and published are: Symbols for Hydraulics; Symbols for Heat and Thermodynamics; Symbols for Photometry and Illumination; Aeronautical Symbols; Mathematical Symbols; Letter Symbols for Electrical Quantities; Navigational and Topographical Symbols; Graphical Symbols for Telephone and Telegraph Use.

Subcommittee on Symbols for Mechanics, Structural Engineering and Testing Materials. J. T. Faig, Chairman. The Subcommittee completed its proposed list of symbols in October, 1930. Following approval by the Sectional Committee, the proposed standard was submitted to the five sponsor bodies, which are now voting on it.

Subcommittee on Symbols for Heat and Thermodynamics. S. A. Moss, Chairman. The American Tentative Standard, "Symbols for Heat and Thermodynamics," was approved and published in February, 1931.

Subcommittee on Abbreviations of Engineering and Scientific Terms. G. A. Stetson, Chairman. This Subcommittee's proposal is now with the sponsor organizations for approval. Three of the

five have given their approval of this proposal and final action by the other two is expected in the near future.

Plain and Lock Washers. C. W. Squier, Chairman. The Sectional Committee on the Standardization of Plain and Lock Washers has given some attention this year to the building up of its personnel. Since this was weak on the producer side the manufacturers of this product were requested to nominate persons employed in this industry who might be appointed to the Committee as "Members at Large." Satisfactory response was received and the resulting list of nominations will be acted upon at a meeting to be held in October. As soon as this personnel has been completed by the Sectional Committee it will be submitted to the American Standards Association for its approval.

Subcommittee on Lock Washers. C. H. Loutrel, Chairman. The Subcommittee on Lock Washers met in December, 1930, and again in April, 1931, to discuss tentative proposals. In September, 1931, another tentative proposal was distributed to the members of the Subcommittee for consideration prior to its next meeting, which has been scheduled for October. In developing these proposals the Subcommittee has confined its activity entirely to lock washers to fit the present American Standards for Wrench Head Bolts and Nuts (B18b-1927) and Slotted Head Proportions (B18c-1930).

Machine Pins. M. E. Steczynski, Chairman. The Sectional Committee on Machine Pins distributed a preliminary draft of its proposed standard to industry for criticism and comment in January, 1931. A large number of constructive comments were received. These are now in the hands of the Committee for consideration and incorporation in its revised proposal. As soon as this work has been completed a meeting of the Sectional Committee will be held for a thorough discussion of this entire subject.

Code for Pressure Piping. E. B. Ricketts, Chairman. The Sectional Committee on a Code for Pressure Piping met in December, 1930, and May, 1931, to discuss the several sections of the code proposed by its subcommittees. Another meeting is scheduled for October.

Subcommittee on Power Piping. John Lawrence, Chairman. At the meeting of this Subcommittee held in April, 1931, the revised proposal for Section 2, Power Piping Systems, which had been distributed to the members of the Committee earlier in the month, was discussed. The Subcommittee has decided to delay the final submission of Section 2 of the Code until a decision has been reached by the Manufacturers Standardization Society of the Valve and Fittings Industry on standard shell test pressures and by the Sectional Committee on the unification of the pipe thickness formulas used in the several sections.

Subcommittee on Gas and Air Piping. Alfred Iddles, Chairman. This Subcommittee reported at the May, 1931, meeting of the Sectional Committee that objections had been raised by the American Gas Association to certain paragraphs of its tentative proposal on Section 4, and that the A.G.A. had formulated a redraft of these paragraphs which exp ssed its views on the subject.

Subcommittee on Refrigerating Piping. A. H. Baer, Chairman. The first meeting of this Subcommittee was held in December, 1930, at which a preliminary outline of Section 5 distributed to the members in November was discussed, changed, and extended. It was pointed out that there are few if any precedents in the regulation of refrigerating piping. The Chairman submitted a tentative proposal for Section 5 to the members in April, 1931, for criticism and comment. This first rough draft has recognized the several requirements of the many different kinds of refrigerants, most of which make necessary differences in materials, joints, etc. However, the Committee plans to cover only the refrigerants in common use. It is expected that this proposal will be in tentative shape for distribution to the members of the Sectional Committee at an early date.

Subcommittee on Oil Piping. A. D. Sanderson, Chairman. This subcommittee has developed three tentative drafts of this proposal, dated January, March, and September, 1931. These drafts have been discussed by the members of the Subcommittee and at the meeting of the Sectional Committee held in May, 1931. The September draft will be discussed at the meeting of the Subcommittee to be held in October and reported at the Sectional Committee meeting the following day.

Subcommittee on Piping Materials and Identification. F. H. Morehead, Chairman. The meeting of this Subcommittee in October, 1930, dealt principally with schemes for marking and identification of pipe and fittings and with the plans of the Manufacturers Standardization Society of the Valve and Fittings Industry to present a recommendation on this subject. In September, 1931, the Chairman submitted a revised list of material specifications to the members for criticism. During the year the Subcommittee has also been conducting a study of the thermal expansion of pipe material.

Subcommittee on Fabrication Details. C. G. Spencer, Chairman. The five subgroups of this Subcommittee have been very active this year in developing questionnaires and draft proposals of the various sections on Hangers; Joints Other Than Welded; Pipe Bends; Power Piping; and Welding. The subgroup on Hangers, Supports and Anchors has distributed preliminary drafts of its proposal, dated January, April, June, and October, 1931, to the members of the Subcommittee for consideration. In January, 1931, the subgroup on Joints Other Than Welded distributed to the Chairmen of five Subcommittees of the Sectional Committee copies of the summary of the replies to the questionnaire which it had mailed to industry in June, 1930. Under date of August, 1931, the subgroup sent to the members of Subcommittee on Fabrication Details the subgroup's proposal, dated July, 1931. This proposal was based on the replies to a questionnaire sent out in June. The December, 1930, proposal of the subgroup on Welding was discussed at a joint meeting of the American Welding Society and the Sectional Committee held in New York in April, 1931. Copies of the discussion at this meeting were mailed to the members of the Subcommittee for their information in July, 1931. A meeting of this subgroup was held in Boston in September. This subgroup has developed a revised draft of its August proposal which will be discussed at the October meeting of the Sectional Committee.

Subcommittee on District Heating Piping. D. S. Boyden, Chairman. Two meetings of the Subcommittee on District Heating Piping were held during the year, one in December, 1930, and the other in April, 1931. This Subcommittee has prepared four drafts of its proposal during the past year. These were dated November, December, 1930; April and June, 1931. The last was distributed to the members of the Subcommittee in September for study prior to reporting at the meeting of the Sectional Committee in October.

Drawings and Drafting Room Practice. F. DeR. Furman, Chairman. During the past twelve months this Sectional Committee has not had occasion to hold a meeting. The Editing Committee made up of the Chairmen of the six Subcommittees, held an important meeting on January 2 and 3, 1931. At this meeting the progress made so far by the Subcommittees was thoroughly discussed and plans were formulated for the completion of the various sections of the Sectional Committee's report.

The Subcommittee on Specifications for Paper and Cloth developed a new series of proposed sizes in January, 1931, copies of which were sent to the members of the Subcommittee. The Subcommittee on Methods of Indicating Dimensions distributed a preliminary proposal in October, 1929. A summary of the replies are now in the hands of the Chairman of the Subcommittee, and it is expected that a tentative proposal will be ready for submission to the Sectional Committee for its consideration at its next meeting. The Subcommittee on Lettering developed a revised proposal in line with suggestions made at the January meeting of the Editing Committee, and in April, 1931, copies were mailed to the members of these two groups. The Subcommittee on Line Work distributed to industry in March and April, 1931, a questionnaire on the method of representing screw threads on drawings. The replies to this questionnaire are now in the hands of this Subcommittee for consideration.

Graphic Presentation. E. F. DuBrul, Chairman. The Sectional Committee on Graphic Presentation held no meeting during the year. Subcommittee No. 3 on Time Series Charts, No. 4 on Non-Time Series Charts, and No. 5 on Survey of Current Practice held a joint meeting in October, 1930, which resulted in the combining of the three committees into one Subcommittee on Preferred Practice in Graphic Presentation, with Mr. A. H. Richardson as Chairman. This new Subcommittee plans to divide its program of work among small subgroups.

Subcommittee on Scientific and Engineering Graphs. W. A. Shewhart, Chairman. The Subcommittee on Scientific and Engineering Graphs held an open meeting during the 1930 Annual Meeting of the A.S.M.E. Three papers were presented at this time under the following titles: (1) "Appearance Factors in Engineering Graphs;" (2) "Attention Value of Good Chart Design;" and (3) "Angle of View and Its Bearing on Layout, Proportions, Weights of Lines and Lettering of Lantern Slides." The first of these papers is to form one of a series of three monographs to be submitted by this Committee for publication during the next few months. The Subcommittee is now making plans for an open meeting to be held during the first week of December in New York, N. Y. At this time one of the subjects of discussion will be the principles of chart construction effecting the production of good lantern slides.

Transmission Chains and Sprockets. F. V. Hetzel, Chairman. This Sectional Committee on Transmission Chains and Sprockets

developed the American Standard for Roller Chains, Sprockets, and Cutters which was approved by the American Standards Association in July, 1930, and is now available in the pamphlet form, published by the Society of Automotive Engineers. The Committee now has under consideration the standardization of Silent Chains. The personnel of the Sectional Committee was approved by the A.S.A. in May, 1931.

Wire and Sheet Metal Gaging Systems. W. W. Hutchins, Temporary Chairman. While the Sectional Committee on Wire and Sheet Metal Gaging Systems has undertaken the development of standard gaging systems for (1) Flat Stocks, (2) Wires and Rods, and (3) Tubing, Piping, Casing, and Conduit, no progress has been reported during the past twelve months.

Electric Motor Frame Dimensions. W. F. Dixon, Chairman. In January, 1931, this Sectional Committee met for the purpose of clarifying the situation arising out of the request by the N.E.M.A. that the A.S.A. withhold approval of the proposed standard for mounting dimensions of general purpose motors which was submitted to the A.S.A. in April, 1930, by the A.S.M.E. and the N.E.M.A. as joint sponsors. At this meeting the N.E.M.A. proposed the adoption of a restricted scope for the Committee's activities, under which it would be willing to cooperate in the standardization of motor frame dimensions. This abridged scope was accepted tentatively by those present at the meeting subject, to a formal canvass of the Sectional Committee membership by letter ballot. The scope proposed by the N.E.M.A. consists of the following three items: (1) a series of dimensions for the height between the plane of base and the center line of shaft; (2) the spread, at right angles to the center line of the shaft, between the bolt holes, definitely co-related to each dimension in Item 1; (3) the distance, parallel with the line of the shaft, from the center line of the bolt holes next to the driving end of the motor to the shoulder of the shaft, also definitely co-related with each dimension in Item 1.

Pipe Threads. E. M. Herr, Chairman. Three Subcommittees have held meetings during the past year and the progress reported is recorded in the following paragraphs:

Subcommittee on Taper Pipe Threads. S. B. Terry, Chairman. In October, 1930, this Subcommittee met on two consecutive days and discussed in detail the proposed editorial revision of the Taper Pipe Thread section of the American Pipe Thread Standard published in 1919. A considerable proportion of the text was tentatively revised at that time and the remainder has been completed since through correspondence. Two informal conferences were also held during the year. The proposed revision of this section, in printer's proof form, is now in the hands of the committee members for consideration at the next meeting, which it is expected will be held before the end of the calendar year.

The possible consolidation of the American Standard Pipe Thread and A.P.I. Standard Line Pipe Thread has been given considerable attention, but no definite conclusion has been reached so far.

Subcommittee on Straight Pipe Threads. Alton S. Miller, Temporary Chairman. A meeting of this Subcommittee was held in October, 1930, and the character and scope of the Committee's work was discussed at length. As a result of this discussion it was decided to develop basic dimensions for gages on the basis of two pitch diameters, one for pressure joints and one for locknut threads. The development of "go" and "not go" gages for threadings not readily gaged by the present taper plug gages was also decided upon.

Subcommittee on Plumbers Threads. L. A. Cornelius, Chairman. This Subcommittee has held one meeting during the past twelve months, in December, 1930, at which the reports of its three subgroups were discussed. Two of these reports, on "threading for thin wall non-pressure tubing in sizes larger than 2 in." and "threading for 14 and 17 gage non-pressure tubing in sizes 2 in. and smaller" were accepted as complete. The third, on threading for corporation stops, was recommended for adoption, subject to certain modifications in thread sizes and the inclusion of tolerances. It is anticipated that this report in completed form will be available for distribution to the Subcommittee members at an early date.

Subcommittee on Screw Threads for Rigid Electrical Conduit. H. S. White, Temporary Chairman. While no meeting of this Subcommittee has been held during the past twelve months, a threading specification for rigid electrical conduit has been adopted recently as standard by the conduit manufacturers. This specification covers threading dimensions and tolerances for sizes $^1/_2$ in. to 6 in., inclusive, and copies are being forwarded to the Subcommittee members for their consideration.

Wrought Iron and Wrought Steel Pipe and Tubing. H. H. Morgan, Chairman. In June, 1931, a meeting of the Sectional Committee on Wrought Iron and Wrought Steel Pipe and Tubing was held in Chicago during the A.S.T.M. Convention, and a report of the progress made at a joint meeting of Subcommittee on Pipe and Tubing for Low Temperature Service and Subcommittee on Pipe and Tubing for High Temperature Service held the previous day was received. A report of the letter ballot of the Sectional Committee relative to the basic formula developed by the Subgroup for use in setting up schedules of pipe wall thicknesses, indicating approval, was made also at that time. This Sectional Committee, at the request of the Sectional Committee on Pipe Flanges and Fittings, had given consideration to the elimination of the $3^1/_2$ in. nominal pipe size from the standards developed by the latter committee. The Secretary reported at the June meeting that a ballot of the members of the Committee had been taken on this subject, and that the results indicated that they were somewhat divided in opinion. Following further discussion at this meeting, the Committee recommended the retention of the $3^1/_2$ in. pipe size in standard weight and extra strong pipe and its elimination from the heavier weight pipe, dimensions for which are to be set up.

Subcommittee on Plan, Scope, and Editing. H. H. Morgan, Chairman. The Subcommittee met in Pittsburgh in March, 1931, and discussed the program and progress of work of the several Subcommittees. It was agreed at that time that arrangements be made for meetings of the Sectional Committee and the two Subcommittees during the month of June in Chicago.

Subcommittee on Pipe and Tubing for Low Temperature Service J. J. Shuman, Chairman, and Subcommittee on Pipe and Tubing for High Temperature Service. J. R. Tanner, Chairman. The two Subcommittees held a joint meeting a day previous to the June, 1931, meeting of the Sectional Committee. Considerable work had been done by these Subcommittees in preparing schedules of pipe wall thicknesses conforming to the basic formula which had been adopted. Tables and graphs illustrating the general scheme of the application of the basic formula to commercial pipe sizes were distributed at the meeting. Both Subcommittees are now at work on the extension and completion of this study, which will include sizes up to 24 in. O.D. The subject of pipe nomenclature is also receiving the attention of these Committees.

Speeds of Machinery. A. E. Hall, Temporary Chairman. Progress in the work of the Sectional Committee on the Standardization of Speeds of Machinery has been dependent upon the results from the distribution of a questionnaire which was prepared by the Subcommittee on Questionnaire and Canvass to Industry. This Subcommittee held a meeting in December, 1930, and developed a preliminary draft of the questionnaire which was completed and distributed in March, 1931, to approximately 400 manufacturers and users of various types of machinery. The one hundred and sixty-one replies received to this distribution are now being compiled by a committee of one, preparatory to a meeting of the Subcommittee in October. The Subcommittee's report will be used as the basis of the Sectional Committee's first proposal.

Fire Hose Couplings. The American Standard for Fire Hose Coupling Screw Threads was approved and published in May, 1925. Each year since that date has marked a continuing increase in its adoption by cities and towns throughout the United States. The National Board of Fire Underwriters, the Chamber of Commerce of the United States, the A.S.M.E. Local Sections, and many local organizations have cooperated in the promulgation of this standard. A report compiled by the National Board of Fire Underwriters early in the year indicates that 3930 protected cities and towns located in 40 of the 48 states, including the District of Columbia, have adopted the American Standard. The States of Oregon, Massachusetts, Texas, Maryland, and California have passed legislative bills prohibiting the sale, purchase, or use of any fire-fighting apparatus which does not conform to this Standard. The adoption of the American Standard by the State of California, in June, 1931, is largely the result of the activities of the San Francisco Section of the A.S.M.E.

Screw Threads for Hose Couplings. H. W. Bearce, Chairman. The Sectional Committee on the Standardization of Screw Threads for Hose Couplings distributed its preliminary proposal for American Standard Screw Threads for Hose Couplings of $^1/_2$ to 2 in. in diameter in February, 1931, for criticism and comment. The comments received have been studied by the Committee and a revision based on these comments will shortly be submitted to the Sectional Committee for discussion and approval by letter ballot.

Plumbing Equipment. W. C. Groeniger, Chairman. The Sectional Committee on the Standardization of Plumbing Equipment has set up eight subcommittees which are at work developing standards in this field. In January, 1931, a new Subcommittee was appointed, that on Cast Iron Soil Pipe and Fittings. The personnels

of the first four subcommittees are identical with those of the Standing Committees on the same subject of the Division of Simplified Practice, U.S. Department of Commerce.

Subcommittee on Plumbing Code. W. C. Groeniger, Acting Chairman. This Subcommittee, the personnel of which is the same as that of the U.S. Department of Commerce's Subcommittee on Plumbing of the Building Code Committee, is revising its code which was published in 1929. As soon as revised copies are available they will be distributed to members of the Sectional Committee for consideration.

Subcommittee on Brass Plumbing Products. J. L. Murphy, Chairman. In March, 1931, the Chairman distributed to the members of the Subcommittee copies of the preliminary proposals of several of its subgroups. These were for Subgroup No. 1 on Compression Cocks, Washer Screws, Faucets, and Tail Piece Parts; Subgroup No. 3 on Fixture Connections and Fixture Valves; Waste Holes, Plugs, and Connections; Overflows; Subgroup No. 4 on Valves; and Subgroup No. 7 on Nomenclature. For the purpose of discussing these sections the Subcommittee now plans to hold a meeting during the third week in October.

Subcommittee on Copper Water Tube Fittings. F. L. Riggin, Chairman. This Subcommittee distributed a tentative proposal in February, 1931, to approximately 200 manufacturers and users. The comments received are to be considered at a meeting of the Subcommittee in October.

Subcommittee on Cast Iron Soil Pipe and Fittings. J. J. Crotty, Chairman. The Subcommittee was organized in January, 1931, and since then has held five meetings at which tentative proposals were considered by the members. In August a questionnaire was distributed to the manufacturers of soil pipe requesting that they submit to the committee a list of pipe fittings now included in the Soil Pipe Association's Specifications, but which, in their estimation, could be excluded from the proposed American Standard. An investigation is also being made to determine whether it is practical to improve upon the design of certain fittings.

Standardization and Unification of Screw Threads. R. E. Flanders, Chairman. In October, 1930, the Sectional Committee met and reviewed the progress already made by its six subcommittees. At this time also detailed arrangements were made for conducting the "Survey of Screw Thread Practice in the United States," and it was decided that this should be started as soon as practicable. The subject of wood screw standardization was considered, and subsequently a seventh Subcommittee was appointed to investigate this project. During the year three of the subcommittees have met and the following progress is reported.

Subcommittee on Scope, Arrangement, and Editing. R. E. Flanders, Chairman. This subcommittee met in April, 1931, for consideration of the various activities within its scope; one of the most important of these is the following:

Survey of Screw Thread Practice in the United States. At that time considerable progress was reported in the collection of samples for the Screw Thread Survey. Since then all the required samples of threaded product have been collected by the Committee's representative and shipped to the National Bureau of Standards, where the gaging is being done. In all, 161 firms were visited personally and of these 128 contributed samples. The total number of pieces collected and sent to the Bureau of Standards is 7579, comprising on the average 5 pieces each of 1539 individual samples. The expenses of this survey are being met jointly by the Bureau of Standards and the Sectional Committee. The Committee's part has been subscribed by Engineering Foundation and a number of firms which manufacture and use screw threading tools and threaded products. The carrying through of the detailed arrangements, including the financing of the survey, which had been worked out by Subcommittee No. 1 on Plan, Scope, and Editing, was entrusted to a subgroup of three appointed on the authorization of the Sectional Committee. The members of this Subgroup are: R. E. Flanders, C. W. Bettcher, and E. Buckingham.

Subcommittee on Special Threads and the Twelve Pitch Series Except Gages. R. E. Flanders, Temporary Chairman. In April, 1931, this Subcommittee held a meeting and decided to set up a 12-pitch series having the following characteristics: (1) tolerances adaptable to general purpose applications; (2) tolerances on the threads in the nuts limited to Class 2; (3) tolerances on the screw threads given in Classes 2 and 3. It also accepted as a part of its immediate work the setting up of an 8-pitch thread series in diameters from 1 to 3 in., inclusive. This latter series is intended primarily for the use of the piping industry.

Subcommittee on Wood Screws. A. Boor, Chairman. At its meeting held in April, 1931, this Subcommittee discussed the possibility of standardizing the minor diameters of wood screws. After reviewing certain data on the present-day practice, recommendations were made on the proportion of minor to major diameters, and it was decided to canvass the manufacturers on this point. It is expected that a definite conclusion will be reached, on the basis of the replies to the questionnaire, at the next meeting.

Rolled Threads for Screw Shells of Electric Sockets and Lamp Bases. R. E. Meyers, Chairman. During the year the Sectional Committee completed the proposed American Standard for Rolled Threads for Screw Shells of Electric Sockets and Lamp Bases. It was submitted to the A.S.A. in September, 1931, for final approval and designation as an "American Standard." Following the granting of this approval the standard will be printed in pamphlet form.

In February, 1931, the members of the Sectional Committee on Rolled Threads for Screw Shells of Electric Sockets and Lamp Bases were designated by the U.S. National Committee of the International Electrotechnical Commission as a Committee of Advisors on Lamp Holders and Bases to the U.S.N.C., to prepare the United States proposal for the September, 1931, meeting of I.E.C. Advisory Committee No. 6, Lamp Holders and Bases, in London. Two meetings of the Committee of Advisors were held, one in May and another in July, 1931. At the latter meeting the committee prepared a report recommending to the U.S.N.C. that the proposed American Standard be adopted as the International Standard for shallow threaded shells and that the European standard be adopted as the International Standard for deep threaded shells. This recommendation was made after careful consideration of American and European practice, which the Committee believed could not be unified at the present time.

Stock Sizes, Shapes, and Lengths for Hot and Cold Finished Iron and Steel Bars. F. H. Frankland, Temporary Chairman. At the time of the organization meeting of the Sectional Committee on the Standardization of Stock Sizes, Shapes and Lengths for Hot and Cold Finished Iron and Steel Bars, in June, 1930, it was recommended that several additional organizations be invited to appoint official representatives on the Committee. This took a longer time than usual, so that it was not until June, 1931, that the Chairman was able to complete the personnels of the three subcommittees on (1) Hot Rolled Steels; (2) Cold Finished Steels; and (3) Hot Rolled Iron.

Subcommittee on Cold Finished Steels. L. E. Creighton, Temporary Chairman. A meeting of this subcommittee has been set for October 20.

Pressure and Vacuum Gages. M. D. Engle, Temporary Chairman. Sponsorship was accepted by the A.S.M.E. for this project in June, 1930, and 55 associations and societies were requested to appoint representatives to this Sectional Committee. The organization meeting was held in December, 1930, at which time the scope as proposed by the American Standards Association was approved by the Committee without change. At a meeting of Subcommittee No. 1 on Plan and Scope in June three subcommittees were appointed to develop the several sections of the proposed standard.

Subcommittee No. 2. C. F. Schwep, Chairman. Subcommittee No. 2 was appointed to develop the proposed section on definitions, specifications, and rules for installation and use. In July, 1931, this committee held its first meeting, and in September the first preliminary proposal was sent to its members.

Subcommittee No. 3. H. B. Reynolds, Chairman. This Subcommittee plans to meet during the fall group of Technical Committee meetings in October. Its scope is to include gage sizes and mounting dimensions; (suggested list of preferred sizes); size and type of connections of gages and cocks; indicator hands (balance—interchangeability); stop pins (when and were to be used); bezel rings and methods of attachment; and safety features for high pressure gages.

Subcommittee No. 4. O. J. Hodge, Chairman. The section on dial ranges; recommended dial ranges for various working pressures; graduations, markings, and space for trade names; accuracy requirements for different services; method of expressing the degree of accuracy; and method of testing (laboratory and field), has been assigned to this subcommittee. No meetings have been held so far.

Foundry Equipment. E. S. Carman, Temporary Chairman. The sponsorship for this project was accepted by the American Foundrymen's Association and the A.S.M.E. in February, 1931. In the same month letters were addressed to 28 associations and societies requesting them to appoint official representatives to the Sectional Committee. The organization meeting was held in May, 1931, at which time the Committee voted to change its name to the Sectional Committee on the Standardization of Foundry Equipment and Supplies. The work of developing the proposed standard was divided at that meeting into three parts to be handled by subcommittees to be appointed by the Chairman. At this meeting the Committee also

voted to request six additional organizations to appoint representatives. As soon as these appointments are completed the Chairman will name the members of the subcommittees.

Leather Belting. J. C. Mattern, Temporary Chairman. Sole sponsorship for this project was accepted by the A.S.M.E. in January, 1931. The preliminary organization of the Sectional Committee on the Standardization of Specifications for Leather Belting was immediately begun, and in May, 1931, the Committee held its first meeting. A definite outline of procedure was developed at this meeting, and two Subcommittees were appointed, one on Standard Specifications and the other on Recommendations for Selection, Care, and Installation.

Subcommittee on Standard Specifications. R. C. Bowker, Chairman. The first meeting of this Subcommittee is scheduled to be held in November, 1931.

Subcommittee on Recommendations for Selection, Care, and Installation. G. A. Schieren, Chairman. On the same day that the Sectional Committee was organized the Subcommittee on Recommendations for Selection, Care, and Installation held its first meeting and elected permanent officers. As the first step in the development of its proposal the Subcommittee is endeavoring to secure copies of all specifications having a bearing on its work. This data will form the basis of its preliminary recommendations. The Chairman has planned for a meeting early in November.

A.S.M.E. Committee on Shaft Couplings. D. J. McCormack, Chairman. During the early part of the year an effort was made to correlate the work of the A.S.M.E. Committee on Shaft Couplings with that of one of the several Sectional Committees, organized under the American Standards Association procedure, whose scope and personnel might be appropriate to a review of a proposed standard on shaft couplings. This endeavor, however, proved to be impracticable, and it was decided that the Committee's proposal, after approval by the A.S.M.E., should be initiated into the A.S.A. procedure under the "Proprietary Standards" method. In July, 1931, printer's proof copies of the proposed standard for Integrally Forged Flange Type Shaft Couplings for Hydro-Electric Units, developed by this A.S.M.E. Committee, were distributed to the Committee members and to other interested persons for criticism and comment. The replies received as a result of this distribution are now in the hands of the Committee for consideration, and formal approval by the Committee of this proposal for submittal to the A.S.M.E. Council is expected at an early date. When it has been accepted by the A.S.M.E. it will be transmitted to the American Standards Association for approval and designation as an American Standard.

Standards Published. The following standards developed by these Committees have been published by the A.S.M.E.

Cold Finished Shafting, Diameters and Lengths (B17a-1924)
Square and Flat Stock Keys, Widths and Heights (B17b-1925)
Plain Taper Stock Keys, Square and Flat (B17d-1927)
Gib Head Taper Stock Keys, Square and Flat (B17e-1927)
Code for Design of Transmission Shafting (B17c-1927)
Woodruff Keys, Keyslots, and Cutters (B17f-1930)
Tolerances, Allowances, and Gages for Metal Fits (B4a-1925)
Spur Gear Tooth Form, 14½ Degree Composite System, 20 Degree Stub Involute System (B6b-1927)
American Standard Screw Threads for Bolts, Machine Screws, Nuts, and Commercially Tapped Holes (B1a-1924)
Cast Iron Pipe Flanges and Flanged Fittings; All Sizes for Maximum Working Saturated Steam Pressure of 25 Lb. per Sq. In. (Gage); All Sizes for a maximum gas working pressure of 25 Lb. per Sq. In.; Sizes 36 In. and Smaller for Maximum Non-Shock Working Hydraulic Pressure of 43 Lb. per Sq. In. (Gage) at or near the ordinary range of air temperatures. (B16b2-1931)
Cast Iron Pipe Flanges and Flanged Fittings; All Sizes for Maximum Working Saturated Steam Pressure of 125 Lb. per Sq. In. (Gage); Sizes 12 Inches and Smaller for Maximum Non-Shock Working Hydraulic Pressure of 175 Lb. per Sq. In. (Gage) at or Near the Ordinary Range of Air Temperatures (B16a-1928)
Cast Iron Pipe Flanges and Flanged Fittings; All Sizes for Maximum Working Saturated Steam Pressure of 250 Lb. per Sq. In. (Gage); Sizes 10 Inches and Smaller for Maximum Non-Shock Working Hydraulic Pressure of 400 Lb. per Sq. In. (Gage) at or near the ordinary range of air temperatures (B16b-1928)
Cast Iron Screwed Fittings, for Maximum Working Saturated Steam Pressure of 125 and 250 Lb. per Sq. In. (Gage) (B16d-1927)
Malleable Iron Screwed Fittings, for Maximum Working Saturated Steam Pressure of 150 Lb. per Sq. In. (Gage) (B16c-1927)
Cast Iron Long Turn Sprinkler Fittings (Screwed and Flanged) for Maximum Hydraulic Working Pressures of 150 and 250 Lb. per Sq. In. (Gage) (B16g-1929)
Steel Pipe Flanges and Flanged Fittings for Maximum Working Steam Pressures of 250, 400, 600, 900, and 1350 Lb. per Sq. In. (Gage) at a Temperature of 750 Deg. Fahr. (B16e-1927)
Small Rivets, 7/16 Inch Nominal Diameter and Under (B18a-1927)
Tinners', Coopers', and Belt Rivets (B18g-1929)
Wrench Head Bolts and Nuts and Wrench Openings (B18b-1927)
Round Unslotted Head Bolts (Carriage, Step, and Machine Bolts) (B18e-1928)
Slotted Head Proportions; Machine Screw, Cap Screws, and Wood Screws (B18c-1930)
Plow Bolts (B18f-1928)
Track Bolts and Nuts (B18d-1930)
Scheme for Identification of Piping Systems (A13-1928)
T-Slots, Their Bolts, Nuts, Tongues, and Cutters (B5a-1927)
Tool Holder Shanks and Tool Post Openings (B5b-1929)
Taps—Cut and Ground Thread (B5e-1930)
Milling Cutters—Nomenclature, Diameters, Thickness, and Other Important Dimensions (B5c-1930)
Symbols for Hydraulics (Z10b-1928)
Aeronautical Symbols (Z10e-1930)
Mathematical Symbols (Z10f-1928)
Symbols for Photometry and Illumination (Z10d-1930)
Symbols for Heat and Thermodynamics (Z10c-1931)
Fire Hose Coupling Screw Thread for All Connections Having Nominal Inside Diameters of 2½, 3, 3½, and 4½ Inches. (B26-1925)

Respectfully submitted,
LEWIS K. SILLCOX, *Chairman*
H. BIRCHARD TAYLOR
WILLIAM S. MONROE
EARLE BUCKINGHAM.

SAFETY COMMITTEE

The standing Committee on Safety has continued throughout the year its efforts in the promotion of industrial safety through the activities of the Society. A Symposium on Industrial Accident Prevention was held during the Annual Meeting of the Society in December, 1930. Half-day sessions on Safety were planned for the Birmingham, Hartford, and Kansas City meetings, as well as for this December's Annual Meeting of the Society; however, it was not found possible to carry out this program, except for the Birmingham Meeting. A joint session with the National Safety Council was held at the Birmingham Meeting in April, 1931. Papers were read on the economic aspects of accident prevention through engineering in the iron and steel industry. As a result of the efforts of the Committee an active part was taken by the Society in the New York Safety Conference, held in New York in February, 1931, in which 42 organizations cooperated. Chairman Mowery arranged for the five-minute talk on "Relation of Safety in the Printing Industry," given at the Printing Industries Session in December, 1930. Plans have been made also for a talk by Frances Perkins, Commissioner of Labor of the State of New York, on "State Labor Departments and Professional Engineers—Cooperation Essential to Progress in Accident Prevention" during the Annual Meeting, December, 1931.

The Committee has requested authors of papers to be given at the various A.S.M.E. meetings to deal with the safety aspects of their subjects. This plan is intended to take the place of the five-minute talks previously given at such times. It was further suggested that the economies of accident prevention should be stressed in articles in *Mechanical Engineering*, and that books on machine design should be reviewed from the safety point of view in *Mechanical Engineering*.

The Committee has proposed the appointment of a joint A.S.M.E. and N.S.C. committee to promote ways and means of increasing the interest of both organizations' membership in accident prevention, but after due consideration and discussion, it was decided that the existing set-up is best, whereby the A.S.S.E. Committee on Relations With Other Engineering Societies shall continue to function for the N.S.C. and the Standing Committee on Safety for the A.S.M.E. The Chairman also reports that the A.S.S.E. Committee on Cooperation With Other Engineering Societies, as a result of its recent experience, suggests the following activities as likely to create a greater interest within those national bodies in advancing the cause of accident prevention: (1) Endeavor to secure the appointment by each national engineering society of an accident prevention committee; (2) arrange for joint sessions on safety as an integral part of the annual, mid-year, or regional meetings of the national societies or their divisions; (3) contact the educational committees of the national societies (where such committees have been appointed)

to advance the project of inclusion of safety engineering in the curricula of technical schools and colleges.

Last spring the organization of a special committee to develop a code for aircraft safety and inspection, as suggested by the Professional Division on Aeronautics, was approved by the Committee, and in June, 1931, a committee of six was appointed by the President. The Code is to include construction, maintenance, and operation of aircraft from the viewpoint of safety in its relation to personnel, cargo, and the public—both on the ground and in the air.

During the year articles totaling ten pages of text have been published in *Mechanical Engineering*.

Of the 50 safety code committees now functioning under the procedure of the American Standards Association, five are sponsored solely or jointly by the A.S.M.E. These are:

Sectional Committee on a Safety Code for Mechanical Power Transmission Apparatus
Sectional Committee on a Safety Code for Elevators
Sectional Committee on a Safety Code for Machinery for Compressing Air
Sectional Committee on a Safety Code for Conveyors and Conveying Machinery
Sectional Committee on a Safety Code for Cranes, Derricks, and Hoists.

Personnel

The Sectional Committees on safety codes sponsored by the A.S.M.E. now have a membership of 220. Of this number 90 are members of the Society. These five Committees have subdivided their work among 23 subcommittees and subgroups, making a total of 28 committees at work on this safety program.

Activities of Sectional Committees

Mechanical Power Transmission Apparatus. C. B. Auel, Chairman. The Safety Code for Mechanical Power Transmission Apparatus was originally completed by the Sectional Committee in 1923, revised and raised to the status of an American Standard in 1927. Late in 1930 the State of New Jersey Department of Labor reprinted this code in its entirety. A revision of this Code, which is to include a chapter on mechanical power control, is now in progress.

Elevators. S. W. Jones, Chairman. The Sectional Committee completed and approved the second revision of the Safety Code for Elevators, Dumbwaiters, and Escalators in February, 1931, and after formal approval by the sponsor organizations the Bureau of Standards, the American Institute of Architects, and The American Society of Mechanical Engineers, it was approved by the American Standards Association in July.

The Subcommittee on Research, Interpretations, and Recommendations, M. H. Christopherson, Chairman, formed after the publication of the 1925 revision of the Code, has continued its monthly meetings.

Subcommittee on Elevator Inspectors' Handbook. K. A. Colahan, Chairman. In December, 1930, printer's proofs of Part 1, Routine Inspection of the Elevator Inspectors' Handbook, were sent to the Subcommittee for letter ballot approval. Before the ballot was completed, however, developments indicated that it would be necessary for the committee to await the results of certain cable tests before the paragraph on Cable Inspection could be approved. These tests are now being developed and will soon be submitted to the subcommittee for approval.

Conveyors and Conveying Machinery. C. H. Newman, Chairman. Two of the Subcommittees of the Sectional Committee on a Safety Code for Conveyors and Conveying Machinery submitted redrafts of their sections of the Code this spring. These sections deal with gravity conveyors, chutes and live roll conveyors, cable-dragline Conveyors, Part 1 and 2. Subcommittees are actively engaged in preparing reports on Belt Scraper and Screw Conveyors, also Chain Elevators and Conveyors, which are expected to be ready for presentation at the Annual meeting.

Cranes, Derricks, and Hoists. J. C. Wheat, Chairman. The Chairman reports that the three subcommittees on (1) overhead and gentry cranes, (2) locomotive and tractor cranes, and (3) jacks have completed their sections of the code. These are now in the hands of the editing committee. The two other subcommittees on (4) derricks and hoists, and (5) equipment for cranes and hoists, expect to release their tentative proposals about October first. The Editing Committee has been delayed in its work, due to the extra work put on its Chairman, Commander Church, of the U. S. Navy Bureau of Yards and Docks, by rush construction work planned to relieve the unemployment situation.

Other Safety Code Committees. In addition to these Committees, the Society has representation on 26 other safety code committees, thirteen of which have completed codes. These committees are as follows:

Safety Code Correlating Committee
Sectional Committee on Safety Code for Abrasive Wheels
Sectional Committee on Safety Code for Floor Wall Openings, Railings, and Toe Boards
Sectional Committee on Safety Code for Industrial Sanitation
Sectional Committee on Safety Code for Lighting Factories, Mills, and Other Work Places
Sectional Committee on Safety Code for Forging and Hot Metal Stamping
Sectional Committee on Safety Code for Ladders
Sectional Committee on Safety Code for Logging and Sawmill Machinery
Sectional Committee on Safety Code for Aeronautics
Sectional Committee on Safety Code for Textiles
Sectional Committee on Underground Power Transmission and Power Equipment for Metal Mines
Sectional Committee on Safety Code for Machine Tools
Sectional Committee on Safety Code for Paper and Pulp Mills
Sectional Committee on Safety Code for Power Presses, and Foot and Hand Presses
Sectional Committee on Safety Code for Rubber Machinery
Sectional Committee on Safety Code for Walkway Surfaces
Sectional Committee on Safety Code for Amusement Parks
Sectional Committee on Safety Code for Window Washing
Sectional Committee on Safety Code for Color for the Identification of Gas Mask Canisters
Sectional Committee on Safety Code for Construction Work
Sectional Committee on Safety Code for Protection of Industrial Workers in Foundries
Sectional Committee on Safety Code for Prevention of Dust Explosion
Sectional Committee on Safety Code for Ventilation
Sectional Committee on Safety Code for Mechanical Refrigeration
Special A.S.S.E. Committee on Low Voltage Electrical Hazard.

Two of the five sectional committees have approved codes which have been published by the Society; these are:

Safety Code for Mechanical Power Transmission Apparatus (B15-1927)
Elevator Safety Code (A17-1931).

Respectfully submitted,

Harold W. Mowery, *Chairman*
Lew R. Palmer
Lewis Price
Thomas A. Walsh, Jr.
Martin H. Christopherson.

POWER TEST CODES

The Committee on Power Test Codes presents the following report for the year ending September 30, 1931.

Test Codes Completed

During the year just closed the Test Code for Liquid Fuels and the Test Code for Atmospheric Water-Cooling Equipment were completed and printed in pamphlet form. In addition, several sections of Instruments and Apparatus were completed and issued in pamphlet form, namely, Part 3, Temperature Measurement—Chapter 1, General, Chapter 5 on Pyrometric Cones, Chapter 6 on Liquid-in-Glass Thermometers, and Chapter 7 on Bourdon Tube Thermometers; Part 11, Determination of Quality of Steam; Part 13, Speed Measurements; Part 16, Density Determinations; and Part 17, Determination of the Viscosity of Liquids.

Twenty of the proposed 24 test and supplementary codes and 12 of the proposed 34 sections of Instruments and Apparatus have been completed and published during the period 1922–1931, as follows:

Code on General Instructions
Code on Definitions and Values
Test Code for Solid Fuels
Test Code for Liquid Fuels
Test Code for Stationary Steam-Generating Units
Computation and Report Sheets—(Stationary Steam-Generating Units)
Test Code for Reciprocating Steam Engines
Test Code for Steam Turbines
Test Code for Reciprocating Steam-Driven Displacement Pumps
Test Code for Centrifugal and Rotary Pumps
Test Code for Displacement Compressors and Blowers
Test Code for Feedwater Heaters

Test Code for Condensing Apparatus
Test Code for Atmospheric Water-Cooling Equipment
Test Code for Refrigerating Systems
Test Code for Evaporating Apparatus
Test Code for Steam Locomotives
Test Code for Gas Producers
Test Code for Internal-Combustion Engines
Test Code for Hydraulic Power Plants and Their Equipment
Test Code for Speed-Responsive Governors

Instruments and Apparatus:
- Part 1, General Considerations
- Part 2, Pressure Measurements;
  - Chapter 1, Barometers
  - Chapter 2, Tables, Multipliers, and Standards for Barometers, Mercury, and Water Columns and Pressure Measurements
- Part 3, Temperature Measurement;
  - Chapter 1, General
  - Chapter 5, Pyrometric Cones
  - Chapter 6, Liquid-in-Glass Thermometers
  - Chapter 7, Bourdon Tube Thermometers
- Part 11, Determination of Quality of Steam
- Part 13, Speed Measurements
- Part 16, Density Determinations
- Part 17, Determination of the Viscosity of Liquids
- Part 21, Leakage Measurement;
  - Chapter 1, Condenser Leakage Tests

During the year two of the original codes in the series of test and supplementary codes were also completely revised in the light of current practice and reprinted in pamphlet form. These were the Code on Definitions and Values and the Test Code for Solid Fuels. The Computation and Report Sheets (1929) of the Test Code for Stationary Steam-Generating Units were also revised in line with the recently revised edition of the corresponding code.

Personnel Changes

The Committee records with deep regret the deaths during the year of three of its members who have been active in the work of the Individual Committees: Prof. S. W. Parr, who was a member of Committee No. 3 on Fuels, and Messrs. Charles Prentice Turner and J. T. Wilkin, both of whom were members of Committee No. 9 on Displacement Compressors and Blowers.

Additions have been made by appointment to the individual committees of Frank G. Philo on Committee No. 3 on Fuels and Prof. William A. Sloan on Committee No. 19 on Instruments and Apparatus.

Mr. Ely C. Hutchinson, a member of Committee No. 18 on Hydraulic Power Plants and Their Equipment, was appointed to succeed H. Birchard Taylor as Chairman of this Committee.

The 20 Individual Committees and the Main Committee on Power Test Codes now include in their personnel 119 members of the Society and 12 non-members.

Committee Activity

Committee meetings were held during the year by Committee No. 3 on Fuels, No. 10 on Centrifugal and Turbo-Compressors and Blowers, and No. 6 on Steam Turbines for the purpose of discussing matters pertinent to the revision and further development of their respective codes.

Individual Committee No. 10 on Centrifugal and Turbo-Compressors and Blowers has completed a fourth draft of its Code. The Committee, however, has learned that difficulties have been experienced with tests carried out in accordance with the section of the proposed Code dealing with the testing of fans. This section is based on a test code originally formulated by the American Society of Heating and Ventilating Engineers and the National Association of Fan Manufacturers.

When finally revised page proofs of the Test Code for Complete Steam-Electric Power Plants were presented for final approval, the Main Committee found a number of inconsistencies and omissions of a technical nature. The Code, therefore, was referred back to Individual Committee No. 11 for the purpose of giving consideration to the corresponding suggestions.

A résumé of the progress which has been made during the past year in the preparation of the various parts and chapters of Instruments and Apparatus is as follows: Part 2, Pressure Measurement—Chapter 2 on Static and Total Pressure, Static Holes and Tubes, Impact Tubes, and Chapter 3 on Pipes for Pressure Measurement appeared in the July, 1931, issue of *Mechanical Engineering;* Part 5 on Measurement of Quantity of Material appeared in the August, 1931, issue; Part 9, Heat of Combustion in the December, 1930, issue; Part 16, Density Determinations in the February, 1931, issue, and Part 20, Smoke-Density Determinations in the November, 1930, issue. The development of the following parts and chapters has progressed appreciably within the past year; Part 3, Temperature Measurement—Chapter 2 on Radiation Pyrometers, and Chapter 8 on Optical Pyrometers; Part 4 on Head Measuring Apparatus; Part 10 on Flue Gas Analysis; Part 12 on Measurement of Time; Part 14 on Measurement of Surface Areas; Part 18 on Humidity Determinations; and Part 21, Leakage Measurement—Chapter 2 on Boiler, Piping, and Engine Leakage.

The publication of abstracts of preliminary drafts of the Power Test Codes and other related material in *Mechanical Engineering* for the year has filled 16 pages.

Boiler Heating Surface

Since the publication of the 1929 edition of the Test Code for Stationary Steam-Generating Units some considerable differences of opinion have developed in industry regarding the determination of the heating surface of certain types of water-tube boilers. This discussion was brought to the attention of Individual Committee No. 4 on Stationary Steam-Generating Units. Before making further changes in their Code, the members of the Committee requested an expression of opinion by the Main Committee on the exact definition of boiler heating surface. The Main Committee voted that there is no deviation between the amount of the boiler heating surface as defined by the Code on Definitions and Values, the Test Code for Stationary Steam-Generating Units, and the Boiler Construction Code, and that the total boiler heating surface as thus defined is that surface which is between the fluid to be heated on one side and the hot gases or refractories on the other, the area measured on the heat receiving side, the fluid being part of the circulating system whatever the nature of the fluid and whether it be liquid or vapor.

Throttle and Exhaust Pressure

During the year considerable correspondence passed between the members of Individual Committee No. 5 on Reciprocating Steam Engines and the Main Committee relative to the addition of a section to the Test Code for Reciprocating Steam Engines dealing with the measurement of throttle pressure and exhaust pressure. Inasmuch as the Committee was not in a position to submit a unanimous report to the Main Committee the latter group undertook to set up certain definitions for admission or throttle pressure and back or exhaust pressure which it believes will meet the present need for reciprocating steam engines.

The Main Committee has requested Committee No. 5 to revise the Test Code for Reciprocating Steam Engines, bringing its context and form in line with the several codes in this field which have been formulated since this Code was completed and published. The present differences lie largely in definitions of load—particularly electrical loading—and in the rules for corrections which are being urged by the International Electrotechnical Commission.

Graphical Symbols

Sketches often play a large part in the negotiations for complex power-plant equipment. Consequently the use of conventionalized graphical representations of the various pieces of apparatus forming part of these installations has been made evident. A set of 20 standard graphical symbols for heat-power apparatus was adopted by Advisory Committee No. 5 on Steam Turbines of the International Electrotechnical Commission at its meeting in Stockholm in July, 1930. This list of symbols was, accordingly, given careful consideration by the Committee on Definitions and Values and has been adopted for use with the A.S.M.E. Power Test Codes.

Under the procedure of the American Standards Association, the Sectional Committee on Scientific and Engineering Symbols and Abbreviations developed standards for graphical symbols for electric power apparatus and wiring. A selection from this list was made and included in the June, 1931, revision of the Code on Definitions and Values.

Reproduction of A.S.M.E. Power Test Code

Frequent requests have been received by the Society for permission to reprint the A.S.M.E. Power Test Codes in whole or in part. The Main Committee recommended that the Society should grant permission for the reproduction of abstracts of these codes and supplementary material in textbooks and trade literature, provided that (1) attention is called to the fact that the material is reproduced in abstract and (2) that full reference to the originating source is given. This latter is important since all of these codes are in a constant state of flux and each reprinting is in reality a revision.

Principles Underlying the A.S.M.E. Power Test Codes

During the year the Committee has been giving considerable thought to a statement dealing with the principles which should underly the formulation and use of the A.S.M.E. Power Test Codes which was presented by Francis Hodgkinson. Copies of Mr. Hodgkinson's statement and recommendations were first sent to the entire membership of the Main Committee and the Chairman of the 20 individual committees for review and study preparatory to a full discussion at the May 25th meeting of the Committee. At this meeting the Main Committee reaffirmed the fundamental objectives and purposes of the codes adopted in 1918 and since, and endorsed certain modifications in their make-up, the same to apply to revisions of existing codes and the preparation of new codes.

A special subcommittee consisting of F. Hodgkinson, Chairman, C. Harold Berry, C. F. Hirshfeld, E. B. Ricketts, and W. M. White was appointed to study these principles in detail and to amend, if necessary, the policy to be followed henceforth in the preparation of new or revisions of existing power test codes. This special committee has completed its report, which will be submitted to the October, 1931, meeting of the Main Committee for final approval and subsequent distribution to the entire membership of the Power Test Codes Committee as to the new policy of the Committee.

National Contacts

The Committee this year has corresponded with the American Gas Association relative to the desire of the newly formed A.G.A. Subcommittee on Approval Requirements for Industrial Gas Boilers to secure information on an acceptable and recognized method of running efficiency tests and rating of gas-fired boilers.

Due to the fact that Individual Committee No. 3 on Fuels was best qualified to negotiate with the A.S.T.M. special committee which has been organized to give consideration to the development of definitions for net calorific value and gross calorific value of fuels, Prof. W. J. Wohlenberg, Chairman of the Committee, was appointed by the President as the A.S.M.E. representative, on the recommendation of the Main Committee. This project was brought to the attention of the A.S.T.M. through the American Committee of the World Power Conference.

International Electrotechnical Commission

Through membership on the U.S. National Committee of the I.E.C., the Society and the Main Committee on Power Test Codes has been actively engaged during the year in assisting to bring about international agreements on acceptance tests for steam turbines and internal-combustion engines. As previously reported, the Secretariat for the Advisory Committees on these types of prime movers is held by the United States. The Society was represented this year on the U. S. National Committee by Fred R. Low, William F. Durand, Francis Hodgkinson, C. Harold Berry, Harvey N. Davis, Ely C. Hutchinson, Irving E. Moultrop, and Paul Diserens.

Steam Turbines

In February, 1931, finally corrected proofs of Part I, "Specification" and Part II, "Rules for Acceptance Tests" of the I.E.C. Publication on Steam Turbines were released by the Secretariat for final publication. Since then, in July, 1931, the Central Office of the International Electrotechnical Commission completed the printing and distribution.

In July the Secretariat, of which Dr. F. R. Low is Director, mailed to the ten member countries of Advisory Committee No. 5 on Steam Turbines, printer's proofs of the Supplementary Notes to Section 4, Instruments and Methods of Measurement of Part II. This latter document consists of a revision of Appendix E, Supplementary Notes in accordance with agreements reached in Stockholm, July, 1930, and is now ready for further discussion by the members of the Advisory Committee.

While the two completed documents on Steam Turbines have been approved by the Advisory Committee and the I.E.C., a question was raised as to the desirability of increasing the scope of the Rules to include other types of steam turbines. In accordance with instructions, the Secretariat referred the matter to the member nations for recommendations. When these comments are received the Secretariat will prepare documents for further discussion by Advisory Committee No. 5 concerning proposals for increasing, or otherwise, the scope of the Rules upon which agreements already have been reached.

The French delegate at Stockholm introduced a paper on "A Rational Definition of Steam Turbine Efficiencies." A translation of this article was made by the Secretariat and circulated to the National Committees for their consideration and recommendation.

Inasmuch as the technique of flow measurement by means of tubes, nozzles, and orifices was somewhat in a state of flux, the material on flow measurement before the Advisory Committee was referred by the Secretariat to the International Standards Association with the request that they propose insertions or deletions which may be considered advisable.

Internal-Combustion Engines

At the Stockholm meeting, July, 1930, some of the documents before Advisory Committee No. 19 on Internal-Combustion Engines were discussed, but no decisions were reached. It was agreed that the Secretariat would transmit to the National Committees of the ten member nations a résumé of that discussion, requesting their recommendations concerning these proposals. This résumé was prepared by Francis Hodgkinson, Director of Secretariat, and distributed to the member nations under date of December 13, 1930. Since that time, the Secretariat has received the German Rules for Acceptance Tests of Gas and Internal-Combustion Engines. The German National Committee and the British National Committee have issued their comments in response to the Secretariat's letter.

On June 1, 1931, the Secretariat again requested the member nations to expedite the transmission of their recommendations. At the end of a reasonable time the Secretariat will proceed to prepare for future discussion a new document for Part I, "Specification," reconciling, as far as possible, the views of all the member nations. This will replace the corresponding documents now before the Advisory Committee.

Hydraulic Turbines

It will be recalled that the present I.E.C. Publication (No. 41) on the Testing of Hydraulic Turbines was completed by Advisory Committee No. 4 on Hydraulic Turbines at Bellagio, September, 1927, under Dr. Durand's able leadership and adopted by the Plenary Meetings of the I.E.C. during the same month.

Under date of February 15, 1930, the Secretariat addressed a letter to the delegates and secretaries of the 22 National Committees of the I.E.C. inquiring if, during the past two years, the hydraulic engineers of the several countries have put the provisions of this document into practice in national and international commerce, and if there was a desire to expand this I.E.C. publication. The replies received to the Secretariat's letter have been few and very non-committal. Under these circumstances no attempt was made to arrange for a meeting of Advisory Committee No. 4 on Hydraulic Turbines during the Scandinavian Meeting in June, 1930. Since that time, however, the Secretariat has decided to proceed with redrafting the Publication No. 41 along the lines suggested.

For some time past Dr. Durand has felt that he should retire from the directorship of this activity, leaving it to some one preferably in the East to carry on. After careful consideration on the recommendation of Dr. Durand, Dr. Sharp, President of the U. S. National Committee, appointed Ely C. Hutchinson in Dr. Durand's stead as Director of the Secretariat on Hydraulic Turbines. It is expected that the preliminary work of redrafting the publication on the Testing of Hydraulic Turbines will begin very shortly.

Respectfully submitted,

Fred R. Low, *Chairman*

C. Harold Berry	David S. Jacobus
Albert T. Brown	Charles E. Lucke
Edwin H. Brown	Lionel S. Marks
Alexander G. Christie	Lewis F. Moody
Harte Cooke	Irving E. Moultrop
Paul Diserens	Henry B. Oatley
Robert H. Fernald	Geo. A. Orrok
Edwards R. Fish	Reginald J. S. Pigott
C. F. Hirshfeld	Edwin B. Ricketts
Francis Hodgkinson	Edward N. Trump
Ozni P. Hood	William M. White
George A. Horne	Walter J. Wohlenberg.

BOILER CODE

The Boiler Code Committee submits the following report for the fiscal year ending September 30, 1931.

During this period the Committee held nine regular meetings, two of which were two-day meetings, which were devoted to interpretations of the Code and to the consideration of revisions and addenda to the various sections.

As stated in the 1930 Annual Report, the Committee has had under consideration rules for the fusion welding of drums or shells of power boilers and for unfired pressure vessels. These rules received widespread attention. As a result of the comments and criticisms received, the revised rules were published in the June issue of *Mechanical Engineering* and were adopted by Council as a standard of the Society on July 7, 1931.

The Rules for Fusion Welding of Drums or Shells of Power Boilers require the highest type of welding. The use of X-ray apparatus is specified for examining the soundness of all longitudinal and circumferential welded joints up to a limiting thickness, beyond which a manufacturer shall have first demonstrated his ability to produce sound welds in boiler drums or shells of a thickness that have been X-rayed. The limiting thickness specified represents the maximum practical capacity of X-ray equipment at the present time. It is stated in the Code that this limiting thickness will be increased when evidence is submitted to the Boiler Code Committee that X-ray or similar means are developed that can be commercially applied to greater thicknesses. All such drums and shells must be stress relieved by heating them to a specified temperature for a specified time. Searching physical tests are required of the weld metal, and every effort has been made to formulate a Code which will result in safety. It is an interesting fact that none of the many discussions received by the Committee has questioned the safety of welded drums or shells constructed in accordance with the Code.

The Rules for Fusion Welding of Unfired Pressure Vessels are the outcome of long and patient work by the Subcommittee on Unfired Pressure Vessels, which cooperated with the Subcommittee on Welding and a number of organizations and persons expert in the field. The rules divide pressure vessels into three classes based on the intended use of the vessel. Class 1, which represents the highest type of construction, is absolutely without limit as to size, working pressure, or the contents within the scope of the Code; Class 2 provides for vessels with a maximum plate thickness of $1^1/_2$ in., which are not to carry lethal gases or liquids, or to be subjected to as severe conditions of service as those built under Class 1. Class 3 provides for vessels with a maximum plate thickness of $^5/_8$ in., which are not to carry lethal gases or liquids, or to be used for the storage of gases or liquids at any temperature materially exceeding their boiling temperature at atmospheric pressure.

Included in the 1931 revisions are rules for the welding of miniature boilers which were incorporated in the Code for Miniature Boilers.

Hearings were granted to the Compressed Gas Manufacturers' Association relative to rules for the storage of liquefied petroleum gases. This Association had formulated "Tentative Regulations for the Design, Installation, and Operation of Liquefied Petroleum Gas Storage Containers and Appurtenant Equipment," and a copy of the proposed rules had been submitted to the Committee for consideration for incorporation in the Code for Unfired Pressure Vessels so that fusion welding for this particular type of container would be recognized.

Conferences followed with representatives of the American Welding Society, the Compressed Gas Manufacturers' Association, the American Boiler Manufacturers' Association, and the National Board of Boiler and Pressure Vessel Inspectors, who rendered valuable assistance and cooperated in the formulation of the rules. A number of others from different industries, such as the petroleum group, cooperated and made valuable contributions.

Another important addition to the Codes has been the incorporation of rules for seal welding of riveted seams of boilers and unfired pressure vessels.

As a result of continued cooperation with the American Society for Testing Materials, many revisions have been made in the Material Specifications Section of the Code. Changes were made in specifications which are identical with those of the A.S.T.M., so as to bring them into conformity with their latest form. During the year new Specifications have been adopted by the A.S.T.M. covering Open-Hearth Iron Plates for Flange Quality and have been accepted by the Boiler Code Committee for incorporation in the Code as Specification S-25.

The Committee has received reports on the tests being made by the Taylor Forge & Pipe Works, Chicago, Ill., in order to supply data for more logical rules on reinforcement of openings than now appear in the Code. The Committee has in the meantime appointed a special committee which is considering this subject with a view to rationalization of the requirements in the Code on such reinforcements.

The Honorary Secretary visited Tulsa and Oklahoma City during the early part of the year for the purpose of investigating the pressure vessel situation there and the application of the Code for Unfired Pressure Vessels to constructions employed in the petroleum industry. This led to a more harmonious understanding between the Committee and the group of manufacturers in that region.

The Committee has made numerous revisions and additions to the various sections of its Code as a result of requests from manufacturers, inspectors, users, and others. These revisions were formally adopted by the Council on July 7, 1931, and distributed in pink-colored addenda sheets.

At the request of the Boiler Code Committee the Council made the following appointments on Subcommittees: F. W. Herendeen, Raymond Newcomb, and W. E. Stark as members of the Subcommittee on Heating Boilers; Paul Diserens as a member of the Subcommittee on Unfired Pressure Vessels. Due to their inactivity in the boiler and pressure vessels industry, Leroy Mikels is no longer a member of the main Boiler Code Committee, and A. S. Kinsey is no longer a member of the Subcommittee on Welding. H. E. Aldrich was elected by the Boiler Code Committee as a member of the Executive Committee.

Respectfully submitted,

F. R. Low, *Chairman*
D. S. Jacobus, *Vice-Chairman*
C. W. Obert, *Hon. Secretary*
H. E. Aldrich
W. H. Boehm
F. S. Clark
F. W. Dean
W. F. Durand
E. R. Fish
V. M. Frost
C. E. Gorton
A. M. Greene, Jr.
F. B. Howell
J. O. Leech
E. F. Miller
M. F. Moore
I. E. Moultrop
C. O. Myers
H. B. Oatley
James Partington
C. L. Warwick
H. LeRoy Whitney.

PROFESSIONAL CONDUCT

During the fiscal year 1930–1931 no complaints on professional conduct were received.

The Committee has been engaged in correspondence on some details connected with the development of the Code of Ethics of the Society as a more profound influence upon the engineering profession.

Respectfully submitted,

John Hunter, *Chairman*
William B. Powell
A. G. Christie
H. S. Philbrick
D. Robert Yarnall.

Reports of Meetings

THE following pages present brief reports of meetings of the Society and of its Professional Divisions during 1931. More detailed accounts were published in *Mechanical Engineering* and the *A.S.M.E. News*. The meetings reported here are as follows:

MEETINGS OF THE SOCIETY

Semi-Annual, Birmingham, Ala., April 20–23, 1931
Hartford, Conn., June 1–3, 1931
Kansas City, Mo., September 7–9, 1931
Annual, New York, N.Y., November 30–December 4, 1931.

MEETINGS OF THE PROFESSIONAL DIVISIONS

Aeronautics, Baltimore, Md., May 12–14, 1931
Applied Mechanics, Lafayette, Ind., June 15–17, 1931
Fuels, Chicago, Ill., February 10–13, 1931
Iron and Steel and Machine Shop Practice, Boston, Mass., September 22, 1931
Materials Handling and Management, Cleveland, Ohio, April 13–17, 1931
Petroleum, Elizabeth, N.J., May 5–6, and State College, Pa., May 22, 1931
Printing Industries, Washington, D.C., March 16–17, 1931
Textiles, Providence, R.I., October 6, 1931
Wood Industries, Winston-Salem, N.C., October 15–16, 1931.

Semi-Annual, Birmingham, Ala.

APRIL 20–23, 1931

The Semi-Annual Meeting was reported in the *A.S.M.E. News* of May 7, 1931, with a special report of the conference of Student Branches in the issue of May 22. The attendance of 435 at the Birmingham meeting included 125 students representing twelve southern colleges. At the Council Meeting held on April 20 the establishment of a Student Branch at the University of Alabama was approved. An announcement that the 1932 Semi-Annual Meeting would be held at Bigwin Inn, Ontario, Canada, from June 27 to July 1, was made at the Business Meeting held on April 20. The Nominating Committee selected the following candidates for new officers of the Society in 1931–1932:

President, Conrad N. Lauer
Vice-Presidents, Robert M. Gates, Charles M. Allen, Fred H. Dorner, William B. Gregory
Managers, Alexander J. Dickie, Eugene W. O'Brien, Harry R. Westcott

The following delegates to the American Engineering Council were nominated:

Conrad N. Lauer, Paul Doty, Ralph E. Flanders, C. E. Ferris, L. P. Alford, J. H. Lawrence, R. C. Marshall, Jr., A. A. Potter, W. H. Trask, Jr., E. C. Hutchinson.

A conference of Local Section delegates was held on April 21. The excusions program included all-day trips to the Tennessee Coal, Iron & Railroad Co. and to the Jordan Dam of the Alabama Power Company on the Coosa River. A special program of automobile trips, luncheons, a flower show, and a garden party was arranged for the ladies. The chief social event of the meeting was the banquet on April 22 at which President Roy V. Wright delivered an address on "How About the Humanities?" and Dr. Harrison E. Howe, editor of *Industrial and Engineering Chemistry*, spoke on "Creating New Industries." The program of papers at technical sessions was as follows:

Monday Morning, April 20

HYDRAULICS

(Auspices of Hydraulic Division)

The Application of Hydraulic Laboratory Researches, I. A. WINTER
Calibration of a Large Nozzle, JOHN R. DUPRIEST and JAS. H. POLHEMUS
Present Status of Hydroelectric Development in the Southeastern United States, WM. P. HAMMOND

PROCESS

The Cottonseed-Oil Industry, E. L. CARPENTER and LEO HOLDREDGE

IRON AND STEEL

(Auspices of Birmingham Local Committee and Iron and Steel Division)

Cast Iron Pipe Produced Centrifugally by the DeLavaud Process, S. B. CLARK
New Mono-Cast Centrifugal Pipe Plant, S. D. MOXLEY

Monday Afternoon, April 20

FUELS AND PETROLEUM

(Auspices of Fuels and Petroleum Divisions)

Grindability of Coal, R. M. HARDGROVE
Design Considerations of a Natural-Gas Transmission System, CHAS. W. MERRIAM, JR.

RAILROAD

(Auspices of Railroad Division)

Practical Application of Alloy Steel to Locomotives, C. E. BARBA
Metallurgy in the Railroad Field, A. E. WHITE

SAFETY

Auspices of A.S.M.E. Safety Committee and American Society of Safety Engineers)

Economic Aspects of Accident Prevention in the Iron and Steel Industry, FREDERICK B. WINSLOW
Accident Prevention Through Safety Engineering in the Iron and Steel Industry, DAVID S. ANDERSON

Tuesday Morning, April 21

POWER

(Auspices of Power Division)

Heat Transfer From Condensing Steam to Flowing Water, A. T. BROWN
Design Features and Operating Results of Fairfield Blast-Furnace Power Plant, F. G. CUTLER

MATERIALS HANDLING AND MACHINE SHOP PRACTICE

(Auspices of Materials Handling and Machine Shop Practice Divisions)

Mold-Handling Methods in Foundries, W. L. HARTLEY
Materials Handling in the Automotive Machine Shop, NORMAN H. PREBLE
Preparation and Distribution of Molding Sands, H. L. MCKINNON

TEXTILES

(Auspices of Textile Division)

Control of Picker-Room Humidity, NORMAN E. ELSAS
Trends in the Cotton Textile Industry, C. L. EMERSON

Tuesday Evening, April 21

MANAGEMENT AND AERONAUTICS

(Auspices of Management and Aeronautic Divisions)

Principles of Industrial Plant Location, H. C. RYDING
Southern Air Transport, FREDERIC G. COBURN

Hartford, Conn.

JUNE 1–3, 1931

A report of the Hartford Meeting appeared in the *A.S.M.E. News* dated June 7, 1931. The attendance at the meeting was about 400. A Council Meeting was held on June 1 and a conference of Local Section delegates on the following day. Special excursions were made to plants of the Pratt & Whitney Aircraft Co., Chance Vought Corporation, Royal Typewriter Company, Colt's Patent Fire Arms Mfg. Co., and Pratt & Whitney Co. Several other plants were visited by the ladies, including that of the Fuller Brush Company and the Cheney Silk Mills. An evening program on June 1 included music by the Choral Club and soloists, and addresses by Senator Hiram Bingham on "The Engineer in Public Life" and by F. M. Ryan, of the Bell Telephone Laboratories on "Airplane Telephony." Speakers at the banquet on June 2 were President Wright and E. E. Wilson, president of the Chance Vought Corporation and Sikorsky Aviation Corporation, whose subject was "Aeronautics." The program of technical sessions follows:

Monday Afternoon, June 1

FUELS

(Auspices of Fuels Division)

How Coal Market Values Are Related to Steam-Plant Design and Operation, G. B. GOULD

APPLIED MECHANICS

(Joint auspices of Applied Mechanics Division and Special Research Committee on Strength of Vessels Under External Pressure)

The Collapsing Strength of Steel Tubes, T. MCLEAN JASPER and JOHN W. W. SULLIVAN
The Strength of Thin Cylindrical Shells Under External Pressure, HAROLD E. SAUNDERS and DWIGHT W. WINDENBURG

EDUCATION AND TRAINING

(Auspices of Committee on Education and Training for the Industries)

Training Foremen to Deal With Human Problems in Industry, JOHN A. MCCARTHY
Training Apprentices and Foremen, FRANCES PERKINS

Tuesday Morning, June 2

POWER

(Auspices of Power Division)

Radiant-Superheater Developments, M. K. DREWRY
Load-Deflection Relations for Large Plain, Corrugated, and Creased Pipe Bends, E. T. COPE and E. A. WERT
The Decomposition of Sodium Carbonate in Steam Boilers (by title), REINHOLD F. LARSON

AERONAUTICS

(Auspices of Aeronautic Division)

Recent Developments in Airplane Design, C. J. MCCARTHY
Recent and Future Developments in Aircraft Engines, A. V. D. WILLGOOS
Recent 6000-Mile Flight to and Above Central America, I. I. SIKORSKY
Motion-Picture of Pratt & Whitney Aircraft Corp. and Chance Vought Corporation Factories

MACHINE SHOP PRACTICE (I)

(Joint auspices of Machine Shop Practice Division and Special Research Committee on Cutting of Metals)

Performance of Cutting Fluids, O. W. BOSTON and C. J. OXFORD
The Influence of Chemical Composition and Heat Treatment of Steel Forgings on Machinability With Shallow Lathe Cuts T. G. DIGGES

PETROLEUM AND APPLIED MECHANICS

(Joint auspices of Petroleum and Applied Mechanics Divisions)

Stresses in Dished Heads of Pressure Vessels, CYRIL O. RHYS
Analysis of Stresses in Circular Plates and Rings, E. O. HOLMBERG and KARL AXELSON

Wednesday Morning, June 3

MANAGEMENT

(Auspices of Management Division)

Discussion of Stabilization of Employment, led by HOWELL CHENEY, RALPH E. FLANDERS, W. R. WEBSTER, H. V. COES, HOWARD TILSON, W. H. RASTALL, FAIRFIELD E. RAYMOND

MACHINE SHOP PRACTICE (II)

(Auspices of Machine Shop Practice Division)

Small Tools and Gaging for Interchangeable Manufacture, E. J. BRYANT
Conventional Gages and Their Application to Duplicate Production, THOMAS F. MACLAREN

TEXTILES

(Auspices of Textile Division)

Use of the Butted Seam by Textile Manufacturers, JOHN M. WASHBURN
Use of Counters in the Textile Industry, HARVEY L. SPAUNBURG
Modern Textile Machinery the Result of Research and Comprehensive Engineering, ALBERT PALMER

MECHANICAL SPRINGS

(Joint auspices of Applied Mechanics Division and Special Research Committee on Mechanical Springs)

Progress Report on Comparative Tensile and Shearing Behavior in Springs, Presented by M. F. SAYRE
Stresses in Curved Bars, H. C. PERKINS

Kansas City, Mo.

SEPTEMBER 7–9, 1931

The A.S.M.E. was one of thirteen organizations which participated in the Southwest Power Conference at Kansas City, September 7–11. More than 400 members and guests of the Society attended the Conference, a report of which will be found in the *A.S.M.E. News* for September 22, 1931. The Southwest Power and Mechanical Exposition paralleled the Conference. The A.S.M.E. was represented at the formal opening of the Conference on September 7 by President Wright. Joint luncheons of the participating organizations were held at which the speakers were Captain Theo. Wyman, Jr., U. S. Engineers Corps, on "The Missouri River," and Marshall E. Sampsell, first vice-president of the N.E.L.A. At a joint banquet a lecture and demonstration on "Adventures in Science" was given by E. L. Manning of the General Electric Research Laboratories. Conferences of the Local Section and Student Branch delegates were held, and the afternoons were largely devoted to sightseeing and inspection

trips. The A.S.M.E. program included six technical sessions, as follows:

Tuesday Morning, September 8

POWER

Flour Milling and Power Requirements, GROVER C. MEYER
1400-Lb. Pressure Boiler and Turbine Operation at Northeast Station, J. A. KEETH
Diphenyloxide, JOHN J. GREBE

OIL AND GAS POWER

A Speed Characteristic for Oil Engines, V. L. MALEEV
Diesel Engines as Peak Load Units, S. A. HADLEY
Internal-Combustion Engines for the Oil and Gas Industries, H. F. SHEPHERD

AERONAUTICS (I)

The Design of Practical Light Airplanes, KARL H. WHITE
Aircraft Diesel Engines of High Output, H. C. EDWARDS
Necessary Trends in Airplane Design, WILLIAM B. STOUT

Wednesday Morning, September 9

FUELS

Improvements in Mining and Preparing Strip Coal in the Southwest, L. RUSSELL KELCE
Coal as Basic Fuel, H. KREISINGER
Some General Chemical and Physical Properties of Boiler Room Refractories, J. H. KRUSON

PETROLEUM

Recent Developments in the Liquefied Petroleum Gas Industry, O. M. SETRUM
Building an Empire (film), GEORGE H. BAIRD
Economic Analysis of Pipe-Line Pumping, Round-Table Discussion, led by W. G. HELTZEL

AERONAUTICS (II)

Aviation Insurance, J. B. GUINOTTE
The Maintenance and Overhaul of Aircraft and Engines, H. L. BREDOUW
Airplane Landing Gears, FREDERICK KNACK

Annual Meeting, New York, N.Y.

NOVEMBER 30–DECEMBER 4, 1931

The Fifty-Second Annual Meeting of the A.S.M.E. was reported in the January, 1932, issue of *Mechanical Engineering* and in the *A.S.M.E. News* of December 22, 1931. The attendance at the meeting was approximately 2000. The closing meeting of the 1931 Council was held on November 30, and the organization of the 1932 Council took place on December 4, when new officers as nominated at the Semi-Annual Meeting were installed. The conferences of Local Section delegates began on November 29 and continued on November 30 and December 1. December 2 was "Student Branch Day." A contest in extemporaneous public speaking was held in conjunction with the formal opening of the second exhibition of art for engineers, on Monday evening, November 30.

The series of lectures on "Talking With an Audience," by Dr. S. Marion Tucker, initiated at the 1930 meeting, were continued. The fourth Henry Robinson Towne Lecture was delivered by W. B. Donham, dean of the Graduate School of Business Administration, Harvard University, on the subject, "The Temporary Emergency and Twenty-Year Planning," and the fourth Robert Henry Thurston lecture by Edward L. Thorndike, professor of education, Teachers College, Columbia University, on "Psychology and Engineering." Another outstanding address was made by Frances Perkins, industrial commissioner of the State of New York, on "State Labor Departments and Professional Engineers—Cooperation Essential to Progress in Accident Prevention."

The Annual Dinner was distinguished by the conferring of honorary membership upon Calvin W. Rice, secretary of the Society since 1906. Dr. Karl T. Compton, president of the Massachusetts Institute of Technology, was the chief speaker of the evening. Honorary membership was also conferred upon Dr. Palmer C. Ricketts, president of the Rensselaer Polytechnic Institute, during the Presidents' Night program, and other awards, including three John Scott medals, were made at that time. The presidential address of Roy V. Wright, entitled "The Engineer Militant," will be found elsewhere in this issue of the Record and Index. Many important committee meetings and conferences were held during the week, and the program of excursions included the observatories of the new Empire State Building, the new plant of the Ford Motor Company at Edgewater, N.J., the S.S. *Pennland*, S.S. *Minnetonka*, and the *Aquitania*, as well as several generating stations, and other inspection trips of technical interest. Reports on the Economic Status of the Engineer were presented at the Business Meeting held on December 1 and at the Local Section conference on the preceding afternoon. The Woman's Auxiliary to the A.S.M.E. arranged a program of special events for the ladies.

The program of papers presented at the technical sessions and conferences follows:

Monday Morning, November 30

MACHINE SHOP PRACTICE AND APPLIED MECHANICS

(Auspices of Machine Shop Practice and Applied Mechanics Divisions)

Improved Device for Recording Instantaneous Tool Pressures in Machinability Studies, O. F. GECHTER and H. R. LAIRD
Photoelastic Developments
(*a*) Kinematography in Photoelasticity, MAX M. FROCHT
(*b*) Photoelastic Study of Shearing Stresses in Keys and Keyways, ARSHAG G. SOLAKIAN and GEORGE B. KARELITZ
(*c*) Celluloid as a Medium for Photoelastic Investigation, R. H. G. EDMONDS and B. T. MCMINN

MATERIALS HANDLING

(Auspices of Materials Handling Division)

Rustless-Steel Conveyors—Their Availability, Application, and Use, NIXON W. ELMER
Application and Design of the Floating Drive for Conveyors, N. H. PREBLE
Improved Material-Handling Methods in a Container Plant, H. N. MACDONALD
Progress Report of Materials Handling Division (presented by title)

PROCESS INFORMAL CONFERENCE

(Auspices of Process Committee)

Discussion on a Rational Approach to the Classification of Data of Unit Operations and Industrial Processes, led by H. J. MASSON

Monday Afternoon, November 30

METAL CUTTING

(Auspices of Machine Shop Practice Division and Special Research Committee on Cutting of Metals)

The Elements of Milling, O. W. BOSTON and CHARLES E. KRAUS
Tungsten Carbide and Other Hard Cutting Materials (Progress Report No. 3 of Subcommittee on Metal-Cutting Materials), COLEMAN SELLERS, 3D
Motion Picture: Latest Development in German Machine Tools as Exhibited in Operation at Leipzig Trade Fair

Fluid Meters

(Auspices of Special Research Committee on Fluid Meters)

A Study of Primary Metering Elements in 3-In. Pipe, S. R. Beitler, Paul Bucher, and T. C. Barnes, Jr.
Flow of Steam Through Square-Edge Orifices in 4-In. Line, W. W. Frymoyer and A. Herschel
Reports of Subcommittees on Revision of Part I, Description of Flow Meters and Water Meters, and Influence of Installation

Applied Mechanics

(Auspices of Applied Mechanics Division)

Design of Spring Gears for Exhaust-Turbine Installations, J. Ormondroyd and T. C. Kuchler
Compression of Rectangular Blocks, and the Bending of Beams by Non-Linear Distributions of Bending Forces, J. N. Goodier
Forced Vibrations With Non-Linear Spring Constants, J. P. Den Hartog and Stanley J. Mikina
Progress Report of Applied Mechanics Division, presented by A. L. Kimball

Tuesday Morning, December 1

Industrial Power

(Auspices of Power Division)

Ford Motor Company's 1200-Lb. Steam-Plant Installation, A. R. Smith and H. B. Hanson
Characteristics of a High-Pressure Series Steam Generator (*Contributed by Special Research Committee on Critical Pressure Steam Boilers*), A. A. Potter, H. L. Solberg, and G. A. Hawkins
Once-Through Series Boiler for 1500 to 5000 Lb. Pressure (*Contributed by Special Research Committee on Critical Pressure Steam Boilers*), H. J. Kerr
Progress Report of Power Division

Casting and Machine Design

(Auspices of Machine Shop Practice and Management Divisions)

Machine-Design Management, J. L. Alden
Correlation of Casting Design and Foundry Practice, Alex Taub
Progress Report of Machine Shop Practice Division, presented by F. C. Spencer

Applied Mechanics and Railroads

(Auspices of Applied Mechanics and Railroad Divisions)

Air Resistance of High-Speed Trains and Interurban Cars, O. G. Tietjens and Kenneth C. Ripley
Stresses in Railroad Track, S. Timoshenko and B. F. Langer
Presentation of Resolutions on the Deaths of Prof. Arthur J. Wood and George Hillyer, Jr.
Progress Report of Railroad Division (presented by title)

Lubrication Research

(Auspices of Special Research Committee on Lubrication)

Optimum Conditions in Journal Bearings, Albert Kingsbury
Pressure Distribution in the Oil Films of Journal Bearings, S. A. and T. R. McKee
Oil-Film Pressures in an End-Lubricated Sleeve Bearing, Louis J. Bradford

Wednesday Morning, December 2

Stabilization Symposium

(Auspices of Management Division with cooperation of American Management Association)

Business Stabilization
(*a*) Financial: Some Thoughts on the Solution of the Depression Problem, Paul M. Mazur
(*b*) Industrial: The Unemployed—What Shall We Do With Them? James W. Hook
(*c*) Economic: The Economic Aspects of Stabilization, Virgil Jordan
Progress Report of Management Division (presented by title)

Fuels

(Auspices of Fuels Division and Special Research Committee on Removal of Ash as Molten Slag from Powdered-Coal Furnaces)

Fluxing of Ashes and Slags as Related to the Slagging-Type Furnace, P. Nicholls and W. T. Reid
Progress Report of Fuels Division

Hydraulics (I)

(Auspices of Hydraulic Division)

Progress Report of Hydraulic Division (presented by title)
Hydraulic Butterfly Valves, Ross L. Mahon
Piezometer Investigation, Charles M. Allen and Leslie J. Hooper

Textiles

(Auspices of Textile Division)

Sanforizing Process, Sanford L. Cluett
Cause and Effect of Recent Advances in Textile Machinery, Edwin D. Fowle
Progress Report of Textile Division (presented by title)

Wednesday Afternoon, December 2

Engineering Economics

(Auspices of Management Division with cooperation of American Management Association)

Dividend Programs Related to Depreciation, E. G. Field
Economic Life of Equipment (Contributed by Materials Handling Division and Special Research Committee on Measures of Management), H. O. Vorlander and F. E. Raymond

Lubrication Engineering

(Auspices of Petroleum Division)

Modern Oil-Lubrication Methods as Applied to Rolling-Mill Equipment, Joseph A. Merrill and Edwin M. May
Oil Requirements for Modern Rolling-Mill Oil Circulating Systems, C. M. Larson

Hydraulics (II)

(Auspices of Hydraulic Division)

Leakage Loss and Axial Thrust in Centrifugal Pumps, Alexey J. Stepanoff
Turbulence in Centrifugal Pumps, J. W. Macmeeken

Steam Tables

(Auspices of Special Research Committee on Thermal Properties of Steam)

Report of the Progress at the Massachusetts Institute of Technology, L. B. Smith and F. G. Keyes
Report of the Progress in Steam Research at the Bureau of Standards, N. S. Osborne
Progress in Steam Table Experiments, Harvey N. Davis
Thermodynamic Availability in Power Plants, J. H. Keenan

Thursday Morning, December 3

Central Station Power

(Auspices of Power Division)

Solubility of Calcium Salts in Boiler Water, Frederick G. Straub
Performance of Modern Steam-Generating Units, C. F. Hirshfeld and G. U. Moran
Influence of Inlet Boxes on the Performance of Induced-Draft Fans, Lionel S. Marks and E. A. Winzenburger

Management Research

(Auspices of Management Division with cooperation of American Management Association)

Coordination of Research and Engineering With Production and Sales, Carl L. Bausch

The Coordination of Research, Sales, and Production, RICHARD F. WILDER

SYMPOSIUM ON STEEL MILL BEARINGS

(Auspices of Iron and Steel Division and Special Research Committee on Heavy Duty Anti-Friction Bearings)

Straight Copper-Lead Alloys Versus Leaded Solid-Solution Bronzes for Heavy-Duty Bearings, F. R. HENSEL and L. M. TICHVINSKY
Millwright Bearings for Roll Necks of Medium-Sized Rolling Mills, K. W. ATWATER
Water-Lubricated Soft-Rubber Bearings, W. F. BUSSE and W. H. DENTON
Application of Celoron Bearings to Roll Necks, ARTHUR J. SCHMITT
Application of Tapered Roller Bearings in Merchant-Bar Mills, S. M. WECKSTEIN

AERONAUTICS

(Auspices of Aeronautic Division with cooperation of American Institute of Physics)

A Study of Light Signals in Aviation and Navigation, I. LANGMUIR and W. F. WESTENDORP
Design Problems of Controllable-Pitch Propellers, W. B. HEINZ
Discussion of Aeronautic Progress
Progress Report of Aeronautic Division, presented by ELMER A. SPERRY, Jr.

Thursday Afternoon, December 3

BOILER-FEEDWATER STUDIES

(Auspices of Joint Research Committee on Boiler-Feedwater Studies)

Determination of Carbonate Hydroxide and Phosphate in Boiler Waters (Progress Report of Subcommittee No. 8), W. C. SCHROEDER and C. H. FELLOWS
Foaming and Priming of Boiler Water, C. W. FOULK
The SO_4/CO_3 Ratio for the Prevention of Sulphate Boiler Scale, EVERETT P. PARTRIDGE, W. C. SCHROEDER, and R. C. ADAMS, JR.

OIL AND GAS POWER

(Auspices of Oil and Gas Power Division)

Progress Report of Oil and Gas Power Division, L. R. FORD
Presentation of the Rudolph Diesel Award for Best Paper Delivered at Oil and Gas Power National Meeting, 1931, to K. J. DE JUHASZ
Small Marine-Diesel Installations, L. B. JACKSON
United States Navy Diesel-Engine Requirements, HOLBROOK GIBSON

METAL ROLLING

(Auspices of Iron and Steel Division)

Recent Developments in Merchant-Bar Mills, G. L. FISK
Progress Report of Iron and Steel Division, presented by LEON CAMMEN
Foreign Supplement to Progress Report, presented by W. TRINKS

EDUCATION AND TRAINING

(Auspices of Committee on Education and Training for the Industries and Management Division)

Opening Address, ERIK OBERG, representing PRESIDENT ROY V WRIGHT
Foremanship Training, U. J. LUPIEN
Modern Industrial Training, GEORGE A. SEYLER

Meetings of the Professional Divisions

Aeronautic Division

BALTIMORE, MD., MAY 12–14, 1931

The Fifth National Technical Meeting of the Aeronautic Division was reported in the *A.S.M.E. News* of May 22, 1931. This meeting was held under the auspices of the Baltimore Section of the A.S.M.E., the Engineers Club of Baltimore, and local sections of the American Chemical Society, American Institute of Architects, American Institute of Chemical Engineers, American Society for Steel Treating, Illuminating Engineering Society, A.I.E.E., and A.S.C.E. Structural parts of different planes, models, and other exhibits were arranged by the A.S.M.E. Student Branch of Johns Hopkins University under the direction of W. H. Miller. The Merrill self-landing airplane was demonstrated at the Curtiss-Wright Airport, where the Wednesday afternoon sessions were held. Wednesday morning was devoted to inspection trips, which included the plants of the Glenn Martin Company, B-J Aircraft Company, Friez Instrument Company, and others. A harbor boat trip was arranged for Tuesday afternoon, and the ladies program included other sightseeing trips. The principal speakers at the banquet were Senator Hiram Bingham, president of the National Aeronautic Association, and Admiral W. A. Moffett, head of the Bureau of Aeronautics. The following technical sessions were held.

Tuesday Morning, May 12

OPERATION

An Analysis of Various Airplane Types for Transport Uses, F. T. COURTNEY
Maintenance of Equipment in Air Transport Operation, LESTER D. SEYMOUR
Aircraft Servicing Problems, CHARLES FROESCH

DESIGN

Design and Possibilities of the Light Airplane, J. HENRY REISNER
Airplane Design in Relation to Tactical Requirements, TEMPLE N. JOYCE
Weight Control in Airplane Design, WM. P. MCKINNIE

ENGINES (1)

Analysis of Aircraft Engine Design, HAROLD CAMINEZ
Development of Engines for Light Airplanes, HENRY LOWE BROWNBACK
Light Aircraft Engines and Their Progress, OTTO E. SZEKELY
Two Cycle Aircraft Engine, J. H. GEISSE

Tuesday Afternoon, May 12

AIRPORTS AND MANAGEMENT

Airports for Small Cities, E. L. WHELESS
Engineering Essentials of Airport Enclosures, FREDERICK SCHAUSS
Architectural Principles Applied to Airport Planning, H. V. WALSH
Actual Cost of Operating a Business Airplane, JEROME LEDERER

CONSTRUCTION AND MATERIALS

Special Methods of Testing Aircraft Materials, D. M. WARNER
Plywood in Aircraft, GEO. L. FISCHER
Types of Design and Their Effect on the Structural Design of Airplanes, F. S. HUBBARD

ENGINES (2)

Altitude Laboratory Tests of Aircraft Engines, H. K. CUMMINGS and E. A. GARLOCK
Aircraft Engine Testing and Testing Laboratories, R. N. DUBOIS
Junkers Aero Diesel Engine Type FO-4, by Junkers Research Institute, Germany, presented by CAPTAIN KENNEDY

Tuesday Evening, May 12

The Aviation Lessons of the Byrd Antarctic Expedition, BERNT BALCHEN

Wednesday Afternoon, May 13

METAL

Construction of the Metal Airplane, GEO. H. PRUDDEN
Some Effects of Corrosion-Fatigue on Streamline Wire of Aircraft, H. E. HAVEN
The Cause and Prevention of Heat Cracks in Aircraft Welding, H. S. GEORGE

PROPELLERS

Racing Propellers for Light Airplanes, FRED. E. WEICK
Magnesium and Hollow Steel Propellers, G. T. LAMPTON
The Value of Controllable Pitch Propellers, W. B. HEINZ
Controllable Pitch Propellers, FRANK W. CALDWELL

TRAINING AND NAVIGATION

Vocational Training in Aviation, FRANK CUSHMAN
Modern Flight Training, W. D. TIPTON
The Aviation Course at the U. S. Naval Academy, D. C. RAMSEY
The Ice Hazard on Airplanes, WILLIAM C. GEER

Thursday Morning, May 14

AERODYNAMIC

Control at Low Speed, CARL HARPER
A Study of Stabilizer Beam Loadings in a Dive and During a Sudden Pull-Up, W. H. MILLER
Tail Spinning of Aircraft, G. G. BUDWIG
Decrease of Disturbed Motion in the Flight of an Airplane, LOUIS BREGUET, presented by A. KLEMIN

ENGINES (3)

Wing Radiator Development, G. A. LUBURG
Engine Ring Cowls, W. H. EVERS
Engine Ring Cowls, PAUL M. BOYD
Observations on the Development of the English High Speed Oil Engines From Submarine to Airship R-101, ALAN E. L. CHORLTON, presented by E. T. VINCENT

INSTRUMENT

Instrument Flying and Its Teaching, W. C. OCKER and CARL J. CRANE
Performance of Aneroid Altimeters for Landing and for Level Flying, W. G. BROMBACHER
The Sonic Altimeter, C. W. RICE
The Directional Gyro, E. A. SPERRY, JR.

Thursday Afternoon, May 14

COMMUNICATION

Radio Methods for Flying and Landing Airplanes in Fog, H. DIAMOND and F. W. DUNMORE
Transmission of Visible Infra-Red and Ultraviolet Radiations Through Fog With Reference to Developing Aids and Fog Flying, S. HERBERT ANDERSON
Effect of Lightning on Airplanes, A. O. AUSTIN
Discussion on Airways Communication, EUGENE SIBLEY, GEO. G. BREED, H. A. ELLIOTT, H. N. WILLETS

AIRSHIPS

The Mooring and Ground Handling of a Rigid Airship, C. E. ROSENDAHL
Water Recovery Apparatus for Airships, CHARLES P. BURGESS
Stresses in Rigid Airships; The Effect of Indeterminateness on Their True Value, W. WATTERS PAGON
Pressure in Airships, W. A. KLIKOFF

GENERAL DESIGN

The S-Rotor, S. SAVONIUS
Single Blade Windmill for Electric Generation, E. N. FALES
The Autogiro Rotor, HERACLIO ALFARO
Supercharging the Engine and Increasing the Speed With Altitude, ROBERT J. NEBESAR
Aerodynamics and Structural Aspects of the Merrill Biplane, JOHN BALLOU

Applied Mechanics Division

LAFAYETTE, IND., JUNE 15–17, 1931

The *A.S.M.E. News* for July 7, 1931, contains an account of the First National Meeting of the Applied Mechanics Division, held at Purdue University, preceding the annual meeting of the Society for the Promotion of Engineering Education. About 170 were registered for the meeting. An exhibit of special research apparatus was held in conjunction with the meeting and included sound-deadening devices, vibration-recording apparatus, small fatigue machines, tensile-testing machines, etc. At a dinner meeting on June 15 speeches were made by J. M. Lessells on "The Growth of the Division," by C. R. Soderberg on "The Engineer of Today," and by Dr. S. Timoshenko on "Teaching of Mechanics at the University of Michigan," and Dr. Bakhmeteff, of Columbia University, spoke briefly on "Hydraulics." There were four technical sessions, the papers for which were as follows:

Monday Morning, June 15

STRESS ANALYSIS

Spiral Springs, J. A. VAN DER BROEK
Recent Advances in Photo-Elasticity and an Investigation of the Stress Distribution in Square Blocks Subjected to Diagonal Compression, MAX M. FROCHT
The Flexibility of Corrugated Pipes Under Longitudinal Forces and Bending, L. H. DONNELL
The Strength of Thin Plates in Compression, T. VON KARMAN, ERNEST E. SECHLER, and L. H. DONNELL

Monday Afternoon, June 15

STRESS ANALYSIS AND PLASTICITY

Thermal Stresses in Thin-Walled Cylinders, CLARENCE H. KENT
Strength of Materials Subjected to Shear at High Temperatures, FRANKLIN L. EVERETT
Flow of Steels at Elevated Temperatures, F. P. COFFIN and T. H. SWISHER
Concerning the Yield Point of Mild Steel, C. W. MACGREGOR
The Center of Torsion for Angle and Channel Selections, W. L. SCHWALBE (presented by title)

Tuesday Morning, June 16

VIBRATION

A New Type of Dynamic Balancing Machine, ERNEST L. THEARLE
Noise Prevention Through Elastic Suspension, E. H. HULL
The Sub-Critical Speeds of the Rotating Shaft, C. RICHARD SODERBERG
Forced Vibration With Combined Coulomb and Viscous Friction, J. P. DEN HARTOG

Tuesday Afternoon, June 16

HYDRODYNAMICS

Propeller Type Fan, O. G. TIETJENS
The Hydraulic Jump and Related Phenomena, BORIS A. BAKHMETEFF
The Boundary Layer and Skin Friction for a Figure of Revolution, CLARK B. MILLIKAN
Jets From Manifold Tubes, JAKOB KUNZ

Fuels Division

CHICAGO, ILL., FEBRUARY 10–13, 1931

The Fourth National Meeting of the Fuels Division was held in conjunction with the Fifth Midwest Power Engineering Conference and the Fifth Midwest Power Exposition. It was reported in the February 22, 1931, issue of the *A.S.M.E. News*.

More than 1000 engineers attended these events. At a joint banquet on February 12 O. H. Caldwell, editor of *Electronics*, delivered an address on "The Electron, Midget Ruler of the Universe," and H. B. Stevens, of the Westinghouse Elec. & Mfg. Co., demonstrated a number of electron tube devices.

Wednesday Afternoon, February 11

Use of Refinery Waste Fuels in Modern Steam Plants, H. J. KLOTZ
The Economics of Fuel Utilization, A. L. BROWN
New Processes of Dehydrating Low-Grade Coal, MAX TOLTZ

Wednesday Evening, February 11

Symposium on Small Stokers, T. A. MARSH
Utilization of Anthracite Coal, C. A. CONNELL
Conversion of Coal-Fired Boilers and Furnaces to Gas Firing, W. D. EDWARDS

Thursday Morning, February 12

Slag Tap Furnaces, A. L. BAKER
Developments in Pulverized-Fuel Firing, H. KREISINGER

Thursday Noon, February 12

LUNCHEON MEETING

Recent Developments in Bituminous-Coal By-Products, A. C. FIELDNER

Thursday Afternoon, February 12

Effect of Fineness of Pulverization on Boiler Efficiencies, E. H. TENNEY
Radiation in Boiler Furnaces, N. ARTSAY
Reduction of Unaccounted for Losses in Boiler Tests, W. F. DAVIDSON

Friday Morning, February 13

Selection of Steam Generating Equipment, F. H. ROSENCRANTS
The Pioneering Element in the Boiler Plant of the Future, GEO. A. ORROK
Changes in Character of Mid-West Coals and Stoker Performance, E. L. MCDONALD

Friday Afternoon, February 13

The Fly Ash Problem, H. N. BUBAR
Round Table Discussion of "Air Pollution Problems," led by ELLIOTT H. WHITLOCK and ELY C. HUTCHINSON

Iron and Steel and Machine Shop Practice Divisions

BOSTON, MASS., SEPTEMBER 22, 1931

A joint meeting of the Iron and Steel and Machine Shop Practice Divisions was held under the auspices of the Boston Section during the National Metal Congress, in which the A.I.M.E., A.S.S.T., S.A.E., and A.W.S. also cooperated. The excursion to the Watertown Arsenal on the morning of September 22 was participated in by about 200. Simultaneously the Iron and Steel Division joined with the A.S.S.T. in a session on sheet steel, at which the following papers were presented.

The Evaluation of the Drawing Quality of Extra Deep Drawing Sheets, THOMAS DOCKRAY
Thin Strip Steel for Deep Drawing, H. T. MORTON and I. A. RUMMLER
Some Effects of the Amount and Rate of Deformation on a Low Carbon Strip Steel, JOSEPH WINLOCK and A. E. LAVERGNE
Effect of Normalizing Upon the Grain Structure and Physical Properties of Automobile Sheet Steel, WM. F. MCGARRITY
Plastic Drawing of Sheet Steel Into Shapes, E. V. CRANE
Lubricants Used in Deep Drawing of Sheet Steel, MAURICE REDWICK
Comparison of Sheet and Strip Steels for Difficult Stampings, EDW. S. LAWRENCE

At a session of the Machine Shop Practice Division in the afternoon three papers were presented, as follows:

Belt Drives With Cast-Iron Pulleys and Paper Pulleys, C. A. NORMAN and G. N. MOFFATT
Positive Drive Equipment, C. R. WEISS
Radiographic Inspection of Steel Castings and Welded Structures, HERBERT R. ISENBURGER

At this session two films were shown, one on the latest developments in German machine tools as exhibited in operation at the Leipzig Trade Fair and the other from the Research Laboratories of the Westinghouse Elec. & Mfg. Co. on gear research studies. A group conference on coordination of research on wear of metal resulted in a unanimous vote to ask the A.S.M.E. to consider the establishment of a committee to act as a clearing house for the research work being done on wear of metals with the possibility of establishing some general principles.

M-M-M Congress

CLEVELAND, OHIO, APRIL 13–17, 1931

The Management and Materials Handling Divisions and the Elimination of Waste Committee of the A.S.M.E. participated in the National Industrial Congress, the local arrangements for which were under the auspices of the Cleveland sections of the A.S.M.E. and S.I.E., the Cleveland Engineering Society, and the Cleveland Handling Equipment Manufacturers Committee and Chamber of Commerce. Other participants were the Society of Industrial Engineers, American Management Association, and Cleveland Electrical Maintenance Engineers Association. An account of the congress will be found in the *A.S.M.E. News* for May 7, 1931.

In addition to the program of technical sessions given herewith, there was a joint banquet at which Newton D. Baker introduced one of the principal speakers of the evening, General G. G Harbord, who spoke on "What the Future Can Expect From Radio." Dr. Lillian Gilbreth was the second speaker; her address dealt with the machine age and its relation to the development of skill

Inspection trips were made to the Westinghouse Elec. & Mfg. Co., the Willard Storage Battery Co., the Hupp Motor Car Co., Lamson & Sessions Co., the Midland Steel Company, American Steel & Wire Co., and the White Motor Company.

The Second National Industrial Exposition paralleled the congress and included a number of exhibits by the Elimination of Waste Committee.

The maintenance engineering sessions were arranged by the Management Division of the A.S.M.E. and the Materials Handling sessions were arranged by the Materials Handling Division.

Monday Afternoon, April 13

SCRAP HANDLING

(Auspices Elimination of Waste Committee, A.S.M.E. and A.M.A.)

Handling of Thin Sheet Scrap in Tin Plate and Sheet Steel Mills, H. H. GILES
Handling of Automobile Body Scrap, R. R. REES
Briquetting of Cast-Iron Borings and Steel Turnings, A. K. NOWAK
Discussion of Handling and Disposition of Ferrous Scrap

Monday Evening, April 13

WASTE ELIMINATION

(Auspices Elimination of Waste Committee, A.S.M.E. and A.M.A.)

Segregation and Control of Waste, C. H. MCKNIGHT
Waste Campaign Experiences and Suggestion Systems. (Short talks giving experiences from several companies.)
A Plant Museum and Its Relation to Plant Control in the Ceramic Industry, W. A. WELDON

Tuesday Afternoon, April 14

MANAGEMENT

(Auspices Management Division, A.S.M.E.—A.M.A. cooperating)

Developments in Manufacture of Lead-Covered Paper-Insulated Telephone Cables, J. R. SHEA
Planning a Lead-Covered Telephone Cable Manufacturing Plant, L. C. HANLEY

HOISTS AND MONORAIL HANDLING

Monorail Handling of Materials, C. W. PRESTON
Engineering Factors in the Use of Electric Hoists, H. J. FULLER
Use of Monorails, A. A. BUREAU and J. R. HOOPER

Tuesday Evening, April 14

DRAFTING ROOM MANAGEMENT

(Auspices Management Division, A.S.M.E.—A.M.A. cooperating)

Production Management Applied to the Drafting Department, W. J. KUNZ
Improved Drafting Room Management, F. D. NEWBURY

Wednesday Afternoon, April 15

CONVEYOR HANDLING

Synchronization of Production Control With Conveyorization, E. M. OLIN and W. C. BEATTIE
Materials Handling in Mass Production at General Electric Plant, H. C. RUNDLE

OUTDOOR HANDLING

Remote Controlled A.C. Haulage at the Plant of the Trinity Portland Cement Company, R. F. EMERSON
Automatic Electric Transfer Cars, E. W. SHELLENTRAGER
Industrial Type of Internal-Combustion Engine Locomotive, S. B. SCHENCK

Wednesday Evening, April 15

DESIGN HANDLING EQUIPMENT

Application of Two-Speed Motors to Monorail Electric Hoists, C. E. SCHIRMER
Variable-Speed Transmission for Materials Handling, D. W. CLEM
Types of Materials Used in Materials Handling Equipment, HERVEY SKINNER

MANAGEMENT

(Auspices Society of Industrial Engineers)

Skill and Satisfaction, LILLIAN GILBRETH

TRUCKING AND SHIPPING

Skids and Skid Shipments, R. L. LOCKWOOD
Use in Marine Terminals, H. E. STOCKER
Use in Railroad Terminals and Stores, J. B. MILLER
Materials Handling of Platforms, C. C. STUBER
The Use of Gas Trucks and Tractors in Materials Handling, GEO. E. HAGEMANN

Thursday Morning, April 16

ORGANIZATION SESSION FOR MAINTENANCE ENGINEERS

Variations in Maintenance Costs and Procedure, E. B. STOODY and G. I. ROSS
Departmental Analyses and Wage Incentives, W. G. UHLIR
The Control of Non-Productive Costs, HARRY W. BENTON
Organization and Operation of the Electrical Maintenance, SIDNEY WATKINS

MATERIALS HANDLING

Automatic Electric Control of Materials Handling, J. E. WOOD
Pneumatic Handling of Vegetable Potash, J. F. BARNARD
Handling Small Packages in the Shop, C. A. FIKE

Thursday Afternoon, April 16

OPERATING SESSION FOR MAINTENANCE ENGINEERS

Maintenance of Electric Cranes and Hoists, ROBT. W. WALTON
Tools for the Maintenance Engineering Department, T. A. KEEFER
Inspection for Preventative Maintenance, A. HECKMAN

MATERIALS HANDLING

Use of Rubber Products for Materials Handling, CHAS W. STAACKE
Weighing and Counting Devices Used With Material Handling Equipment, A. B. JACOBUS

Friday Afternoon, April 17

HOUSEKEEPING SESSION FOR MAINTENANCE ENGINEERS

Maintenance of an Industrial Illumination System, R. C. SMITH
Maintaining the Lighting System, WALTER STURROCK
Maintenance Painting, T. J. MALONEY

MATERIALS HANDLING

Handling Bulk Materials Mechanically, W. W. SAYERS
Uses and Application of Portable Conveyors, J. B. BRAY

Oil and Gas Power Division

MADISON, WIS., JUNE 23–26, 1931

The Fourth National Meeting of the Oil and Gas Power Division held at the University of Wisconsin with a registration of more than 250, was reported in the *A.S.M.E. News* for July 7, 1931. The laboratory of the new mechanical engineering building, which was dedicated on June 22, provided excellent facilities for the exhibit of Diesel engines and equipment. An informal gathering, with entertainment and dancing, on the first evening of the meeting, and a banquet at which Professor Louis Kahlenberg, of the Chemistry Department of the university, was the principal speaker, on Thursday evening, were the chief social events. Thursday was largely devoted to an inspection trip to the works of Fairbanks, Morse & Co. A number of special events for the ladies were provided. The program of technical papers follows:

Tuesday Afternoon, June 23

Dispersion of Oil Sprays, K. J. DEJUHASZ
A Quiescent Combustion Chamber, JOHN A. SPANOGLE
Experiments on the Flow of Air Through Engine Valves, E. S. DENNISON and T. C. KERCHLEE

Wednesday Morning, June 24

Report of Diesel Fuel Research Committee, presented by WILEY BUTLER
Report of Diesel Power Committee, presented by M. J. REED
How to Select a Diesel Engine, OTTO NONNENBRUCH
Development of a High-Speed L-Head Diesel Engine, MAX HOFMANN

Wednesday Afternoon, June 24

Development of Double-Acting Diesels, LOUIS R. FORD
Present Position of the Automatic Diesel-Electric Plant, EDGAR J. KATES
Diesel Municipal Light Plants in the Middle West, C. F. LAMBERT

Wednesday Evening, June 24

Observation of European Diesel Developments, JOHN O. HUSE
European Trends in Diesel Design, O. F. ALLEN
Diesels for Motor Trucks, contributed by Verein deutscher Ingenieure

Thursday Morning, June 25

Application of Oil Engines to Ship Propulsion, A. J. C. ROBERTSON
Light Alloy Pistons and Rods in High-Speed Industrial Engines, B. J. ISIDIN

Friday Morning, June 26

Piston Packing Problems, C. K. SENCEBAUGH
An Outstanding Municipal Diesel Power Plant, CLEMENT RUEGG
Heat Recovery From Internal-Combustion Engines, THOMAS CLARKSON and WILLIAM BRADFORD

Petroleum Refining Meeting

ELIZABETH, N.J., MAY 5–6, 1931

The meeting of the Refining Committee of the Petroleum Division under the auspices of the Plainfield Section was attended by nearly 300 members and guests. An account of it will be found in the *A.S.M.E. News* for May 22, 1931. At a dinner on the evening of May 5 Dr. P. K. Frolich, of the Standard Oil Development Company, spoke on "Chemistry and Mechanical Engineering," and there was an illustrated talk on oil developments in Russia. The discussion at the Tuesday afternoon session resulted in a unanimous vote to request the Refining Committee to appoint a committee made up of representatives of oil refiners and heat-exchange-equipment manufacturers to study the question of a test code. The meeting closed on Wednesday afternoon with an inspection trip to the Bayonne plant of the Standard Oil Company of New Jersey. Papers presented at the technical sessions were as follows:

Tuesday Morning, May 5

Centrifugal Dewaxing of Vacuum Distilled Overhead Lubricating Stocks, MAX B. MILLER
Distillation of Topped Crude Under Vacuum for the Recovery of Lubrication Oil Distillates, C. E. MCCULLOUGH and JOHN PRIMROSE

Tuesday Afternoon, May 5

Suggested Specifications for Testing Tubular Heat Transfer Equipment for Oil Refineries, prepared by Atlantic Refining Company, presented by WALTER SAMANS
Discussion of Problems of High Pressure and Temperature Oil Refining

Wednesday Morning, May 6

The Evolution of Pressure Vessels to Meet Present Day Refining Pressures and Temperatures, H. LEROY WHITNEY
Forged Drums for Oil Refining, JOHN L. COX
Welded Vessels for Oil Refining, T. MCLEAN JASPER
New Development of Chromium Tungsten Steel for Oil Refining Service, V. T. MALCOLM
Discussion on Metal Problems in Oil Refining

Lubrication Engineering Meeting

STATE COLLEGE, PA., MAY 22, 1931

About 125 attended the joint meeting of the Lubrication Engineering Committee of the Petroleum Division with The Pennsylvania State College, during the Fifth Oil Power Conference of the College. At a lubrication research session in the morning the following papers were presented:

Lubricating Testing, E. G. BODEN and O. L. MAAG
Bearing Design in the Light of Oil Film Pressure Investigations, L. J. BRADFORD
Fluid Film Lubrication as Applied to Railroad Journal Bearings, F. O. WILLHOFFT
Oral Discussion of Lubrication of Railway Bearings, M. E. MCDONNELL

A symposium on internal-combustion engine lubrication was held in the afternoon, with papers discussing lubricants from coastal crudes by A. Ludlow Clayden, Pennsylvania crudes by C. M. Larson, and Midcontinent crudes by E. R. Lederer. The meeting was reported in the *A.S.M.E. News* for June 7, 1931.

Printing Industries Division

WASHINGTON, D.C., MARCH 16–17, 1931

The Second Conference of Technical Experts in the Printing Industry was attended by about 500 delegates. The Government Printing Office and the United Typothetae of America were joint hosts for the meeting. The *A.S.M.E. News* for April 7, 1931, presents a report of the conference. In addition to the program of technical papers presented here, there were group conferences on air-conditioning data, establishing color standards for printing, and the standardization of paper, thickness of plates, patent bases, and cylinder diameters. The various departments of the Government Printing Office and its new research laboratory were open for inspection, and there were special demonstrations of printing machinery and other exhibits. Takeo Ohara, connected with the two Japanese newspapers, "Osaka Mainichi" and "Tokyo Nichi-Nichi," was the principal speaker at a dinner meeting, his subject being "The Great Newspapers of Japan: Their Circulations and Equipment." There were four technical sessions, as follows:

Monday Morning, March 16

Air Conditioning for Printing, WILLIS H. CARRIER
Paper Pulp From Hybrid Poplars, a Reforestation Remedy, RALPH H. MCKEE

Monday Afternoon, March 16

Premakeready Methods and Machines, J. W. ROCKEFELLER, JR.
Paper and Ink Testing—Some Simple Tests and Their Application; Some Troubles and the Remedy, SUMMERFIELD ENEY, JR.
Ink Drying by Ozone and the Ultraviolet Ray, ROBERT A. BROWN

Tuesday Morning, March 17

Electrotype Versus Stereotype Printing Plates, WILLIAM T. TIMMONS
The Mechanics of Photoengraving, WILLIAM GAMBLE
Chromium Plating of Plates and Slugs: Their Life and the Advantages, GEORGE B. DRACH and A. WEISBURG

Tuesday Afternoon, March 17

Measuring Color and the Use of the Spectrophotometer in Determining and Matching Shades, P. J. MULDER and JOSEPH RAZEK
The Renck Process, FRIEDRICH SACHS
Ink Misting, ARTHUR THOMPSON
The Effect of Rubber Rollers, BERT REED

Textile Division

PROVIDENCE, R.I., OCTOBER 6, 1931

The New England Meeting of the Textile Division was held jointly with the Providence Section and is reported in the October 22, 1931, issue of the *A.S.M.E. News*. Inspection trips to the Cranston Point Works, the Universal Winding Company, and the Crompton & Knowles Loom Works were arranged for the afternoon. An informal dinner was followed by an evening session at which the following papers were presented.

United Quality and Economy in the Fabrication of Textiles, A. I. HARVEY
Development in Gaseous Fuels for Textiles and a New Singeing Application, R. S. VAN NOTE

The meeting was attended by about 80 members and guests.

Wood Industries Division

WINSTON-SALEM, N.C., OCTOBER 15–16, 1931

The Sixth National Wood Industries Meeting, held under the auspices of the Charlotte and Raleigh Sections, with the cooperation of the Southern Furniture Manufacturers' Association, was

reported in the *A.S.M.E. News* for October 22, 1931. Inspection trips to the B. F. Huntley Furniture plant and the Winston-Salem Chair Company, which opened the meeting, were followed by a luncheon meeting at which Fred Tate, president of the Continental Furniture Company, High Point, N.C., talked on the "Growth of the Furniture and Panel Industry in the South." The second morning was also devoted to inspection trips, visits being made to plants of the Myrtle Desk Company, High Point Furniture Company, Thomasville Chair Company, K. & L. Panel Co., and Kennedy Plywood Company. A feature of the meeting was an exhibit to which educational charts and panels were contributed by the National Committee on Wood Utilization of the Department of Commerce, the Government Forest Products Laboratory of Madison, Wis., Iowa State College, the Bureau of Standards, and a number of companies. The program of the technical sessions follows.

Thursday Afternoon, October 15

MANUFACTURING

The Essentials of Securing Maximum Kiln Efficiency, KENNETH REDMAN
Dimension Stock Manufacture, Its Theory and Economy, D. B. MORGAN
Manufacture of Plywoods and Panels Made With Veneer Cores, N. M. LEWIS
Developing a New Wood Product Through Research, THOMAS D. PERRY

Thursday Evening, October 15

MACHINE AND EQUIPMENT

Types of Hydraulic Presses in Wood Gluing Processes, H. G. FRANCIS
Double End Tenoner and Its Use, JUDSON H. MANSFIELD
Modern Methods Used in Dust-Collecting Systems, H. A. COLVIN
Studies in Design of Local Exhaust Hoods, J. DALLAVILLE and T. HATCH

Friday Afternoon, October 16

SYMPOSIUM ON GLUES FOR WOOD PRODUCTS

The Development of Wood Adhesives and Gluing Technic, T. R. TRUAX
Vegetable and Starch Glues, A. B. MAINE
Oil Seed Residue Glues, I. F. LAUCKS and GLENN DAVIDSON
Casein Glues, W. F. LEICESTER
Animal Glues, WILBUR L. JONES
Blood Albumen Glues, C. P. SMITH

Friday Evening, October 16

DRYING AND FINISHING

How Dry Should Wood Be for Various Industrial Uses? H. L. HENDERSON
Some Observations on Raised Grain, ARTHUR KOEHLER
Comparative Tests on Aluminum Oxide Versus Garnet Paper as Wood Abrasive, J. F. TRAENDLY
Study in Present-Day Wood Finishing, H. A. DERR

Necrology

EDWARD GOODRICH ACHESON

Edward Goodrich Acheson, whose death occurred on July 6, 1931, was born at Washington, Pa., on March 9, 1856, the son of William and Sarah D. (Rupl6) Acheson. At the age of sixteen he was obliged to terminate his studies at the academy at Bellefonte, Pa., because of financial losses suffered by his father. He became time-keeper at his father's blast furnace at Monticello, Pa., and devoted his leisure time to mechanical pursuits which resulted in the invention of a rock-boring machine, for which he filed his first application for a United States patent at the age of seventeen. His father died in 1873 and the furnace was closed the following year. Mr. Acheson was variously employed during the next six years as a dry-goods clerk, surveyor, ticket clerk, and tank gager in the oil fields at Bradford, where he developed a set of tables for easily determining the contents of any tank.

In 1880 Mr. Acheson became interested in the electrical industry and secured employment as an assistant draftsman for Thomas A. Edison at Menlo Park. The following year he received a prize of $100 offered by Mr. Edison for the preparation of a graphite loop or filament, and was appointed first assistant to the chief engineer in charge of the Edison exhibit at the International Exposition in Paris. At the close of the exposition Mr. Acheson aided in the construction of power units and lamp factories at Ivry-sur-Seine for the Société Edison Continentale of Paris, and later installed the first electric lighting systems at Milan, Antwerp, Amsterdam, and Brussels.

After two and one-half years in Europe, Mr. Acheson returned to New York. He left the Edison interests shortly afterward to take a position as superintendent of the Consolidated Lamp Company, Brooklyn, a lamp factory working under the Sawyer-Man patents. He also conducted experiments on the problem of direct conversion of heat into electrical energy and the production of insulated wire. He invented an anti-induction telephone cable in 1886 and attempted to reduce iron ore with natural gas and to produce synthetic rubber. He was electrical engineer for the Standard Underground Cable Company of Pittsburgh from 1886 to 1889.

In 1890 Mr. Acheson founded the Monongahela Electric Light Company. With the high temperatures obtainable electrically there he conducted experiments upon clay-carbon mixtures which resulted in the production of silicon carbide ("carborundum"). The Carborundum Company was formed at Monongahela in 1891, moving four years later to Niagara Falls, where cheaper power was available.

Mr. Acheson next experimented with silicon carbide and found that when subjected to high temperatures it decomposed, leaving a residue of graphitic carbon of high purity. This first product was put on the market in 1899 by the Acheson Graphite Company, which was later known as the International Graphite Company and then the Acheson Graphite Corporation of Niagara Falls.

Graphite, in its turn, led Mr. Acheson to another development. In 1901, while endeavoring to use graphite in crucible manufacture, he tried clay as a binder. He discovered a method of improving the plasticity of clays which, when applied to graphite five years later, led to the production of colloidal-graphited oil and colloidal-graphited water. The Acheson Oildag Company, Port Huron, Mich., was organized in 1908 to manufacture these products under their trade names of "oildag" and "aquadag," respectively.

Another product developed by Mr. Acheson, in 1902, was a chemical compound by electrothermic means which he named siloxicon, a refractory material.

His later years were devoted to finding new uses for his products. The firm of E. G. Acheson, Ltd., was established in England in 1911 to cover the demand for colloidal graphite products in Europe. The Acheson Ink Company was incorporated in 1915 for the production of high quality inks by the deflocculating process. The Acheson Corporation and the Goodrich Corporation were formed in 1915 and 1925, respectively, to coordinate Mr. Acheson's various interests and activities.

Mr. Acheson was the recipient of many honors, including the John Scott Medal by the City of Philadelphia, in 1894 for the production of silicon carbide and in 1901 for the production of artificial graphite; the Count Rumford Medal by the American Academy of Arts and Sciences; the Perkin Medal by the American Section of the Society of Chemical Industry; and the first award under the Edward Goodrich Acheson Fund which he established in 1928 under the control of the American Electrochemical Society. The degree of Doctor of Science was conferred upon him in 1909 by the University of Pittsburgh.

Among the engineering and scientific societies to which Mr. Acheson belonged were the American Institute of Electrical Engineers, Franklin Institute, Society of Arts (London), American Institute of Chemical Engineers, of which he had been a vice-president, American Chemical Society, Society of Chemical Industry, American Ceramic Society, and the A.S.M.E., which he joined in 1918. He was a fellow of the American Association for the Advancement of Science, past-president of the American Electrochemical Society, and an honorary member of the Russian Imperial Technical Society and the Swedish Technical Society.

Mr. Acheson is survived by his widow, Margaret (Maher) Acheson, whom he married in 1884, and by four sons and three married daughters.

EDWIN HUGH AHARA

Edwin Hugh Ahara, who retired from active business in 1923, died of heart trouble in London on June 22, 1931. He is survived by his widow, Emma Josephine (Bigelow) Ahara of Mishawaka, Ind., whom he married in 1897, and by one married daughter. Mr. Ahara was born on January 6, 1865, at Evansville, Wis., the son of Hugh and Mary Jane (Leighton) Ahara. He attended the seminary in Evansville and then entered the University of Wisconsin, from which he received a B.S. degree in civil engineering in 1892 and an M.E. degree four years later. His first position was with the Eclipse Manufacturing Company, Beloit, Wis., where he did general machine-shop work before the completion of his college education. He was also superintendent in 1893–1894 for the Esterly Harvester Company, Minneapolis, Minn., and for a time transit operator on the Upper Mississippi for the United States Government. From 1895 to 1902 he was in charge of tools and mechanical equipment for the Deering Harvester Company, Chicago. From then until his retirement he was associated with the Dodge Manufacturing Company, later known as the Dodge Manufacturing Corporation, of Mishawaka, Ind., serving successively as manager of the manufacturing department, general superintendent, and manager of manufacturing for the corporation.

Mr. Ahara became a member of the A.S.M.E. in 1906. He served as president of the Indiana Engineering Society in 1915 and was president in 1931 of the St. Joseph Valley Society of Engineers. He was an active member of the Masonic fraternity and a member of several clubs in South Bend and Mishawaka. He was captain of the Wisconsin football team while at the University and retained a keen interest in football and other sports. He was also active in civic affairs, being a member of the Mishawaka Chamber of Commerce and of the Mishawaka School Board, on which he had served as secretary and president. In 1915 he was a member of the Mishawaka Public Library Commission. He held United States patents for driving clutches and guards and for sheet metal hangers. As a result of extensive tests extending over many months at the Dodge plant, he presented a paper at the 1913 Annual Meeting of the A.S.M.E. on "Efficiency of Rope Driving as a Means of Power Transmission."

CHARLES DAVIES ALLAN

Charles Davies Allan, of Chicago, Ill., died in that city on January 25, 1930. He was born at Fishkill-on-the-Hudson (now Beacon), N.Y., on March 29, 1868, the son of Dr. George Allan. His technical education was secured at Cornell University, where he took a three-year course in mechanical engineering. After leaving Cornell in the spring of 1893, Mr. Allan spent about twenty years in construction and sales work for different companies, for the most part in New England and New York, and during the remainder of his life engaged in consulting work in the design of heating, ventilating, and power plants.

The companies with which Mr. Allan was connected included the Buffalo (N.Y.) Forge Company, American Stoker Company, New York, N.Y. (no longer in business), B. F. Sturtevant Co., Boston, Mass., American District Steam Company, North Tonawanda, N.Y., Almirall & Co., New York, and General Fire Extinguisher Company, Providence, R.I., for whom he was western manager of the heating and power piping department.

Mr. Allan became a member of the A.S.M.E. in 1917. He also belonged to the American Society of Heating and Ventilating Engineers and served as president of its Illinois chapter in 1923 and 1924. He is survived by his widow, Catherine (Harriman) Allan.

ETHAN EDWARD ALLEN

Ethan Edward Allen, whose death from heart trouble occurred at St. Lukes Hospital, Manila, P.I., on September 10, 1930, was born at Milton, Vt., on July 9, 1867. He was the son of Ethan Edward and Grace Ruth Pearl Allen. He obtained his education at Brigham Academy, Bakersfield, Vt. He served an apprenticeship as a machinist at the Mystic Iron Works, Stanbridge, Quebec, Can., from 1884 to 1889. During the next two years he was toolmaker for the Hamden Watch Company, Springfield, Mass., and Canton, Ohio, and then foreman of the machine department of the Danner Manufacturing Company, Canton. He went to Cleveland in 1892 to work as foreman and designer for the St. John's Typeobar Company. Following that he was employed in 1894 and 1895 by the Warner & Swasey Co. of Cleveland. In 1895 and 1896 he was engaged in the manufacture of bicycles as foreman toolmaker for the Eclipse Bicycle Company, Elmira, N.Y., and the Worcester (Mass.) Cycle Company. He next designed dies for the Worcester Sheet Metal Company for a year. In 1897 he went to Salt Lake City, Utah, to accept a position as superintendent of the Keller Balance Manufacturing Company.

From 1898 to 1900 Mr. Allen saw service in the Philippine Islands as a first sergeant, Battery A, Utah Volunteer Artillery. He was chief engineer of a United States gunboat, at Laguna Bay, P.I., for the next two years. He returned to private business as construction engineer for the Manila & Dagupan R.R. Co., Philippine Islands, in 1902. The following year he became chief engineer for the Coast and Geodetic Survey in the Philippine Islands. In 1905 and 1906 he was assistant superintendent and master mechanic of the San Nicholas Iron Works in Manila, and for the next eight years superintendent for Manuel Earnshaw & Co. of that city. He was engineer for the Abaca Products Company of Manila from 1914 to 1917 and for the next year was connected with the Mambulao Placer Company, Mambulao, P.I. From 1918 to 1920 he was employed by the Cooper Company and during the next five years by the North Negros Sugar Company as chief engineer. After a year's vacation he became president and manager of the Moro Improvement & Trading Co., Inc., at Davao, Mindanao, P.I., where he remained until his death.

Mr. Allen became a member of the A.S.M.E. in 1915. He was a life member of the National Rifle Association and belonged to the Manila Rifle Club. He is survived by his widow, Elizabeth (Rahe) Allen, whom he married in 1913.

HARRY BARTLETT ALVERSON

Harry Bartlett Alverson, electrical engineer for the Buffalo General Electric Company, Buffalo, N.Y., died on January 25, 1931. Ill health for about a year had made it impossible for him to do more than serve the company as a consultant.

Mr. Alverson was born at Portage, Wis., on March 27, 1872, the son of Miles Talcott and Melissa (Low) Alverson. He married Corrine Van Buskirk in 1904. Two daughters survive him.

The University of Wisconsin conferred a B.S. degree in electrical engineering upon Mr. Alverson in 1893. During the following year, he engaged in graduate study at Cornell University. The next two years were spent with the White-Crosby Company (later J. G. White Company) of Baltimore, building a conduit system at Baltimore and street railways at New York and in New Jersey, and with the Buffalo & Niagara Falls Electric Ry. in construction work and as shop and repair foreman.

In 1896, Mr. Alverson became superintendent of The Cataract Power & Conduit Co. This company, which was founded in 1896, was a pioneer in central-station work. It handled power transmitted from Niagara Falls, and Mr. Alverson's position required a thorough knowledge of the design of the plants and their equipment. He continued in the capacity of superintendent and electrical engineer for the company until it was merged with the Buffalo General Electric Company in 1915, when he became electrical engineer of the latter company. He was connected with the construction of the first transmission lines between Niagara Falls and Buffalo, and with the underground conduit system in Buffalo. From the Terminal Station near the Buffalo city line, Mr. Alverson supervised the cable system to the then existing stations of the Buffalo General Electric Company, The International Railway Company, and The Cataract Power & Conduit Co., including the design and construction of the terminal stations and substations of the latter company.

Much of the success of the spectacular electrical display and other features of the Pan-American Exposition in 1901 may be attributed to his ability in the handling of the power involved.

In 1908 he also became treasurer of the Hual Construction Company, Inc., builders of underground conduits, sewers, etc.

Mr. Alverson became a member of the A.S.M.E. in 1915. He also belonged to the American Institute of Electrical Engineers and had served for one year as chairman of its Niagara Frontier Section. He was a member of the University Club, the Buffalo Athletic Club, and the Masonic fraternity.

NATHAN BABCOCK AMBLER

Nathan Babcock Ambler, superintendent of the hydroelectric plant at the Wilson Dam, Florence, Ala., died on April 11, 1931.

Mr. Ambler was born at Chatham, N.Y., on May 29, 1876, and attended the Academy there. He was graduated from the School of Applied Science of Columbia University in 1900 with an electrical engineering degree. During the next three years he was employed by the Metropolitan Street Railway Company, New York, N.Y., as wireman and electrical operator and foreman in its railway substation. From 1904 to 1907 he was engaged in the construction and operation of the railway substation for the Westinghouse, Church, Kerr Company, and the Long Island Railway Company. During the next year, he was superintendent in charge of electrical construction work for the Aluminum Company of America and the Westinghouse, Church, Kerr Company. Acting for the Box Board Manufacturing Company, he took charge in 1909 of the operation and development of an oil field in West Virginia. He returned to New York to accept a position as chief draftsman for electrical traction and station construction for the Pennsylvania Tunnel & Terminal Ry. Two years later, in 1912, Messrs. Gibbs & Hill, consulting engineers, New York, employed him as chief draftsman.

Mr. Ambler entered the employ of the Toronto Power Company, Ltd., of Niagara Falls, in 1913 as superintendent of operation and took charge of the construction and operation of the generating and transformer station for the company. Following his resignation from the company in 1924, Mr. Ambler was connected for a short time with McClellan & Junkersfeld, Inc., New York. He had been with the United States Engineering Department at Florence since 1927. Mr. Ambler's membership in the A.S.M.E. began in 1915

PELLE ANDERSSON

Pelle Andersson, who died on January 10, 1931, was born at Kellstorp, Sweden, on October 9, 1866. He was educated at the Malmo Technical Institute, Sweden, and did engineering work in that country until 1892. In that year he came to the United States and secured employment as a draftsman for the Chase Elevator & Manton Windlass Co. at Warren, R.I. After two years with this company, he went to Philadelphia, Pa., to accept a similar position with the Link-Belt Engineering Company. In 1896 and 1897 he was draftsman for the Corliss Engine Company, Providence, R.I., and the following year for the Marine Machine & Conveyor Co. in New York. From 1898 to 1901 he was associated with the Filer & Stowell Co., Milwaukee, Wis., as chief draftsman and engineer. He then spent four years as mechanical engineer for the Brown Corliss Engine Company, Corliss, Wis., and a year in the position of chief draftsman for the Power & Mining Machinery Co., Cudahy, Wis. For three years, beginning in 1906, he was connected with the Wisconsin Engine Company of Corliss. Since 1909 he had been mechanical engineer for the Allis-Chalmers Manufacturing Company of West Allis and Milwaukee, Wis.

Mr. Andersson became a member of the A.S.M.E. in 1913. He is survived by his widow, Linnea (Hoyer) Andersson, whom he married in 1905, and by one son, Malcolm A. Andersson.

GILBERT L. H. ARNOLD

Gilbert L. H. Arnold, a member of the A.S.M.E. since 1899, died at his home in Stamford, Conn., on February 1, 1931. Mr. Arnold was born in that city on October 12, 1865, and secured his early education there. He was graduated from the Sheffield Scientific School of Yale University in 1886 with a Ph.B. degree in mechanical Engineering. His early experience was secured as a draftsman for the Yale & Towne Co., Stamford, and with F. C. and A. E. Rowland, New Haven. In 1893 and 1894 he assisted in the design, construction, and installation of Belleville boilers on the S.S. *North West* for the Northern Steamship Company, and later had charge of repairs to the boilers on this vessel and on the *Northland* for the same company. For a time he was connected with the Atlas Portland Cement Company, Northampton, Pa., as chief engineer.

About 1904 Mr. Arnold became associated with Frederick A. Waldron, consulting engineer, New York, with whom he remained until his retirement from business in 1925. Concerning his work during these years Mr. Waldron wrote as follows:

"Mr. Arnold's modesty and retiring disposition have hidden from publicity his engineering accomplishments and foresight in the de-

sign of reinforced-concrete and structural steel. As far back as 1906 he developed and used the mathematical data for reinforced concrete now demanded by law. Among his outstanding achievements were the woodworking building (reinforced-concrete) for the National Cash Register Company (1906), service plant (reinforced-concrete and structural steel) of the Edison Electric & Illuminating Co. of Boston (1911–1912), and the John Hancock Mutual Life Insurance Building (reinforced-concrete and structural steel), Boston (1920–1921).

"His versatility and adaptability to engineering problems were well developed, and on data research and special machine design his ability was far above the average. These accomplishments, combined with diplomacy and excellent judgment, will be a loss to the engineering profession."

Surviving Mr. Arnold are his widow, Carrie Elizabeth (Adams) Arnold and three sons and two daughters, John A. Arnold, Englewood, N.J., Edmund S. Arnold, New York, N.Y., and Alfred, Nancy, and Elizabeth Arnold, Stamford, Conn.

JOHN CORLUS WHITE BAKER

John Corlus White Baker, who died at his home in Naugatuck, Conn., on July 15, 1931, was born on January 20, 1865, at Long Branch, N.J. His technical education was secured through the International Correspondence Schools, from which he was graduated in 1902 as a mechanical engineer. He also served an apprenticeship as a machinist with the Wharton Railroad Switch Company of Jenkintown, Pa., and was subsequently employed as foreman by that company.

He secured his early shop experience with the Naugatuck Manufacturing Company and the Excelsior Needle Company, with which he was connected in 1895 and 1896. Later experience was obtained in the Engineering and Repair Departments of Plant No. 4 of the Goodyear Metallic Rubber Shoe Company, a subsidiary of the U. S. Rubber Co. He served successively as foreman, assistant superintendent of construction, and master mechanic at this plant, which later became the Rubber Regenerating Company. From 1920 until 1931, when he retired from active business, he was plant engineer and consulting engineer for that company. Mr. Baker was responsible for the design of special machines for use in rubber mills.

A sister, Kate S. Baker, of Naugatuck, survives Mr. Baker. He had been a member of the A.S.M.E. since 1904.

THOMAS DENT BANKS

Thomas Dent Banks was born in Newcomerstown, Ohio, on December 24, 1885, and attended the high school there. He was graduated from Ohio State University in 1910 with an M.E. degree. His first technical experience was obtained during summer vacations with the Mesta Machine Company of Pittsburgh and the Hocking Valley Railway of Columbus. Following his graduation he secured a position as assistant superintendent of the Garbage Disposal Plant of the City of Columbus. The following year he was made assistant engineer of the Construction Division, and subsequently superintendent of the Division of Garbage and Refuse Disposal. He went to Marion, Ohio, in 1920 to become chief engineer for the Marion Tire & Rubber Co., having charge of the construction of buildings and machinery, factory maintenance, and power plant operation. About a year later he returned to Columbus as superintendent of the power plant at the Ohio State University.

He was stricken with paralysis on July 5, 1930, and had been subject to heart attacks since then. His death occurred on September 14, 1931. His father, Thomas A. Banks, died about seven weeks later. He is survived by his mother and by his widow, Ethel Banks, and three children.

In 1918 Mr. Banks enlisted for overseas reclamation work and was commissioned a first lieutenant. The unit was not sent, however, and after the armistice he was transferred to the Reserve Corps and subsequently was commissioned a captain in the Quartermaster Corps, Fifth Corps area, Fort Hayes, Columbus, Ohio. He was a member of the Faculty Club of the Ohio State University, and was affiliated with the Odd Fellows. He became a junior member of the A.S.M.E. in 1910 and was promoted to full membership in 1921. He was a member of the Columbus Symphony Orchestra.

WILLIAM F. BARNES

William F. Barnes, who died at Rockford, Ill., on December 28, 1930, was born at Mt. Morris, N.Y., on August 6, 1841. After completing his education, he became president in 1874 of the W. F. & J. Barnes Co. of Rockford, Ill., and served continuously in that capacity until his retirement in 1906. He had been a member of the A.S.M.E. since 1886.

REINOLD BERRENBERG

Reinold Berrenberg, owner and general manager of the American Rotary Pump Company, Roxbury, Mass. (now known as the R. Berrenberg Co.), died on March 25, 1931. Mr. Berrenberg was born near Cologne, Germany, on November 1, 1856, the son of Wilhelm and Maria Berrenberg. He served four years as an apprentice in a shop owned by his father and took charge of the shop for two years following that. For three years he was engaged in military service in Germany as first mechanic. He came to the United States in 1882 and was employed for three years by the Osborne Machine Company, builders of marine and stationary engines, as machinist and foreman. He then established his own business in this field, taking up also the manufacture of incandescent electric lamps in 1895. From 1900 to 1903 he was employed as a foreman in a shop manufacturing switchboard and woodworking machinery. During the next ten years he was connected with the Trimount Rotary Power Company of Boston in the design and construction of rotary compressors and pumps for all purposes. Since that date he had been general manager of the American Rotary Pump Company.

Mr. Berrenberg had been a member of the A.S.M.E. since 1913. He married Mary J. Berghaus, in Boston, in 1897. Five daughters and a son survive him.

ARTHUR OLIN BERRY

Arthur Olin Berry, since 1914 senior mechanical engineer for the Division of Valuation, Southern District, Interstate Commerce Commission, died at his home in Chattanooga, Tenn., on May 18, 1931. He is survived by his widow, Helen (Converse) Berry, whom he married in 1904, and by one daughter, Doris (Berry) Rountree, of Chattanooga.

Mr. Berry was born in Holyoke, Mass., on April 29, 1878, the son of Isaac and Maria Fanning (Stewart) Berry. He attended the public schools of that city and was graduated from Sibley College, Cornell University, in 1901, with an M.E. degree.

During the three years after his graduation Mr. Berry served a special apprenticeship and worked with the Boston & Albany Railroad at Springfield, Mass. His work included road tests of locomotives. In 1903 he was appointed division master mechanic at the Rensselaer, N.Y., shops. From 1905 to 1911 he was in the employ of the Lake Shore & Michigan Southern Ry., for two years at Collinwood, Ohio, as general foreman of the locomotive department, and during the remainder of the time at Elkhart, Ind., as superintendent of shops. At Collinwood he had charge of the blacksmith shop and brass foundry, as well as repairing and building locomotives, and at Elkhart he also supervised the foundry which supplied the entire road with its gray iron castings, and made changes which increased the capacity of the foundry about thirty-three per cent.

In 1911 Mr. Berry became assistant master mechanic of the Lake Erie & Western R.R. at Lima, Ohio. During his three years with this road he had charge of the car and locomotive departments, conducted road tests of different classes of bituminous coal, and redesigned caboose cars and the braking equipment of passenger cars to conform to the state laws of Indiana and the requirements of the Interstate Commerce Commission.

In 1914 he was appointed senior mechanical engineer for the Southern District of the Interstate Commerce Commission (Division of Valuation), at Chattanooga, Tenn. In 1921 he took up consulting engineering in that city and three years later was appointed director of smoke abatement, which position he retained until his death.

Mr. Berry became a member of the A.S.M.E. in 1914. He belonged to the Masonic fraternity.

CHARLES SPAFFORD BLAKE

Born at Windsor Locks, Conn., on October 25, 1860, Charles Spafford Blake was educated in the public schools of Springfield, Mass., and Jersey City, N.J., where for a time he engaged in newspaper work as a metropolitan district reporter for the United Press Association. His aptitude for things mechanical led him to abandon the journalistic field and to serve an apprenticeship at the Central Iron Works, Jersey City, a firm that built engines and boilers. Before he had reached his twenty-first birthday he was licensed as an engineer for small vessels and shortly afterward he was certified as chief engineer of ocean-going steamers. Shortly before the close of the Chile-Peruvian war he was commissioned as chief engineer in the Peruvian Navy and was awaiting orders when hostilities ended.

After considerable experience in marine engineering, Mr. Blake entered the boiler insurance field in 1884 as an inspector for the American Steam Boiler Insurance Company. Within three years he had become chief inspector at that company's Philadelphia office and subsequent to that was made adjustor for its Chicago agency. During the last four years which he spent in Chicago he was manager of a brass manufacturing works and consulting engineer in the boiler insurance field.

Mr. Blake became identified with The Hartford Steam Boiler Inspection and Insurance Company on June 1, 1898, as its general agent in Hartford, and six years later he was advanced to the position of supervising general agent. In 1907 he was elected second vice-president and, a year later, was made secretary. He served in that capacity until in 1916, when he was made a director and at the same time elected to the presidency. He was made chairman of the board of directors on February 8, 1927.

Evidence of the confidence his competitors had in him was his annual election for 13 years as president of the Steam Boiler and Flywheel Service and Information Bureau. At various times he was a member of the administrative council of the American Uniform Boiler Law Society, vice-president of the International Association of Casualty and Surety Underwriters, president of the Insurance Institute of Hartford, and director of The Boiler Inspection and Insurance Company of Canada. At the time of his death he was a director of the Phoenix State Bank and Trust Company.

He was prominently identified with church and civic enterprises, and as a member of Asylum Hill Congregational Church served as deacon for several terms, besides being a member of the Ecclesiastical Society's Committee. At one time he was president of the City Missionary Society and was thus actively engaged in the work of the society at its Village Street Mission. He served for a term as president of the City Club and was a member of the Hartford Club and the Hartford Golf Club. He became a member of the A.S.M.E. in 1912.

Mr. Blake was the son of John W. and Lucy Chapin (Hitchcock) Blake. He married Miss Elizabeth Reeves of Newark, N.J., in 1882, and is survived by her and by two sons, Charles Edgar of Windsor and Stanley Reeves of Hartford. He died on March 31, 1931.

ROBERT HARBISON BOOTH

Robert Harbison Booth, who retired in December, 1930, on account of ill health, died at his home in Providence, R.I., on March 22, 1931. He is survived by his widow, Edith Gardner (Andrews) Booth, whom he married in 1907, and by one son.

Mr. Booth was born at Chester, Pa., on November 11, 1883, the son of John W. and Ella F. (Ellis) Booth. He studied mechanical engineering for three years at the Massachusetts Institute of Technology, leaving in 1905 to study steel rolling mill management and practice at the Moline, Ill., plant of the Republic Iron & Steel Co. After three years there he went to Bettendorf, Iowa, as assistant general inspector for the Bettendorf Steel Car Company.

In 1910 Mr. Booth became a member of the staff of Fred A. Kerry, consulting industrial engineer and accountant, Chicago, Ill. This organization was dissolved in 1915 and Mr. Booth continued independently in the same line of work during the next three years. His experience during these eight years covered wholesale and retail stores, foundries, forge shops, bridge and structural shops, metal working plants, etc. In 1918 he accepted a position as superintendent, later becoming assistant to the president, of the Bridgeport Brass Company, Bridgeport, Conn. His duties related particularly to wage rates and incentives and personnel problems. He resigned his position in 1925 to become vice-president and general manager of Whitney-MacDonald, Inc., Philadelphia, and in 1927 was made president of that company, which was reorganized as the MacDonald Corporation. During 1930 he was general manager for the U.S. Bobbin & Shuttle Co., Manchester, N.H.

Mr. Booth became a member of the A.S.M.E. in 1922.

HARRISON SEBA BOWEN

Harrison Seba Bowen, for more than thirty years associated with the Robert W. Hunt Company, Chicago, died of coronary thrombosis on February 14, 1931, following a brief illness.

Mr. Bowen was born at Des Moines, Iowa, on July 13, 1872, the son of Cromwell and Martha (Fletcher) Bowen. He attended high school in Des Moines and then entered Iowa State College, Ames, from which he received the degree of B.M.E. in 1894. A year later he took his M.E. at Cornell University. Prior to his connection with the Hunt company he was employed for about six months by the Missouri, Kansas, and Texas Railroad, Parsons, Kan., and for the same length of time by Booth, Garret & Blair, in inspecting and testing work in Pittsburgh and Philadelphia.

As mechanical engineer for the Hunt company Mr. Bowen was engaged in the investigation and efficiency testing of all kinds of power units. This work covered such important installations as the Chicago water works pumping stations and drainage pumping station units. The mechanical efficiency testing work on heating and power plants under the direction of Mr. Bowen was a pioneer work in the Chicago area. Many installations were reviewed and tested and brought to the highest state of mechanical and fuel consumption uses. Mr. Bowen was engaged in research with complicated and difficult mechanical installations both from the practical operating standpoint and from the standpoint of changes and advance in the art of power use.

Mr. Bowen became a junior member of the A.S.M.E. in 1895 and was promoted to full membership in 1908. He also belonged to the Western Society of Engineers.

Surviving Mr. Bowen are his widow, Katherine (Christman) Bowen, whom he married in 1898, and two sons, Cromwell and Llewellyn Bowen.

JOSEPH BOYER

Joseph Boyer, chairman of the board of directors of the Burroughs Adding Machine Company, died of pneumonia on October 24, 1930, at his home in Detroit, Mich.

Mr. Boyer was a Canadian by birth. His parents, David and Modlany (Brown) Boyer, lived on a farm near Pickering, Ontario, about thirty miles from Toronto, and he was born there on September 19, 1848. At the age of eighteen he persuaded his father to take him to Oshawa where some friends of his were working in a machine shop known as the Hall Works. Three years later, with the opening of the transcontinental railway, he was invited to join some friends who had decided to go to California. His father had died and his mother mortgaged the farm to secure the necessary fare for the trip. Eventually he arrived at San Francisco, where he found work at the Risdon Iron Works for about six weeks. There being nothing further available, he took an opportunity to secure a return ticket to St. Louis, Mo., where he worked for a year or more at the Fulton Iron Works. He then worked at Leavenworth for about a year, after which he returned to St. Louis and secured employment at the St. Louis Type Foundry. Next he was engaged by the Victor Sewing Machine Company to adjust machines.

About 1876 Mr. Boyer left the Victor company to help with the manufacture of a new sewing machine. The company failed and Mr. Boyer and another employee took over the tools and started a small machine shop under the firm name of Boyer & Swaine. Not long afterward he bought out Mr. Swaine's interest and established larger quarters elsewhere in the city, where he remained until 1900. During these years Mr. Burroughs came to his shop for an assistant to work with him on the development of his adding machine. Mr. Boyer's shop built the first few hundred of these machines for the American Arithmometer Company, which later took over their manufacture, though still sharing the same quarters. Mr. Boyer was elected vice-president and a director of this company in 1898. During this period Mr. Boyer invented the pneumatic hammer, the air drill, and the railway speed recorder, and eventually it became necessary to enlarge the shop to provide space necessary for the manufacture of both the adding machine and the pneumatic hammer. Toward 1900 still more space was required, and Mr. Boyer decided to move his business, the Boyer Machine Company, to Detroit.

After a year in his new location Mr. Boyer sold out to the Chicago Pneumatic Tool Company. The following year he was made president of the American Arithmometer Company, which relocated in Detroit in 1904 and was reorganized as the Burroughs Adding Machine Company the following year. Mr. Boyer resigned from the presidency and was made chairman of the board in 1920, at the age of 72, but never lost his active interest in the work, and spent much of his time in the experimental and engineering departments.

Mr. Boyer became a member of the A.S.M.E. in 1913 and also belonged to a number of clubs in Detroit. Surviving him are a son and four married daughters. His wife died in 1913.

HARRY BROADBENT

Harry Broadbent, for more than thirty years associated with the Philadelphia Electric Company, died on March 17, 1931. Mr. Broadbent was born at Oldham, Lancastershire, England, on September 13, 1866. The following year his parents came to Providence, R.I., where he attended the public schools and served an apprenticeship as journeyman machinist with the Harris-Corliss Engine Works.

The family moved to Philadelphia in 1887, and he secured a position

as machinist with Robert Wetherill & Co., Chester, Pa., where he remained only a few months. During the next six years, he was similarly employed in the Maintenance Department for cable roads and motive power of the Philadelphia Traction Company; by Chambers Bros. & Co., manufacturers of brick and paper-folding machinery in Philadelphia; and by the John Lang Paper Company of that city. In 1894 he became engineer at the power house of the Jenkintown Light Company, Jenkintown, Pa. He entered the employ of the Philadelphia Electric Company in a similar capacity in 1899 and was made inspector of the Mechanical Design and Construction Department in 1917. As chief engineer, Mr. Broadbent took charge of reconstruction and extension work at several stations of the company in Philadelphia. He supervised the installation of new equipment and the maintenance of machinery in operation. He was directly connected with the design and construction of the Chester and Delaware Stations, which were built after he was appointed to the position of inspector.

Mr. Broadbent became a member of the A.S.M.E. in 1922. He was very fond of music and sang in the Episcopal choir in West Philadelphia for many years. He is survived by a sister, Mary Alice Broadbent, of Philadelphia, with whom he had made his home since the death of his parents in 1911.

WILLIAM LEROY BRYANT

William LeRoy Bryant, president and general manager of the Bryant Chucking Grinder Company, Springfield, Vt., died there on April 25, 1931, of heart disease. His widow, Blanche (Brown) Bryant, whom he married in 1900, and two sons survive him.

Mr. Bryant was born at Northfield, Vt., on April 29, 1875, the son of William Nelson and Sally Angelia (Holbrook) Bryant. He attended the Black River Academy at Ludlow, Vt., and entered the University of Vermont with the class of 1898. He was obliged to leave college at the end of his junior year but was presented his degree in 1921 by the college "as of the class of 1898."

All of Mr. Bryant's business experience was obtained in Springfield. In 1897–1898 he was employed by the Jones & Lamson Machine Co. in general shop work, and subsequently he was draftsman and chief engineer until 1909 when he organized the Bryant Chucking Grinder Company.

Mr. Bryant had a number of inventions to his credit, holding patents for single slide control for internal and cylindrical grinders, hole and face grinders, multiple spindle automatic internal grinders, grinding wheel spindles, and fluid gages for precision measurements. He became a member of the A.S.M.E. in 1910 and also belonged to the Society of Automotive Engineers and the Vermont Society of Engineers. He was a director of the Associated Industries of Vermont and of the First National Bank of Springfield, a member of the Masonic fraternity and of the Machinery Club in New York. Mr. Bryant belonged to the glee club and band while in college, and played in local bands later.

GEORGE BUCHANAN

George Buchanan, president of the Long Island Foundry Company, Long Island City, N.Y., was a native of Pollokshaws, Scotland, where he was born on February 6, 1867. He came to the United States about the year 1890, and until 1918 was superintendent of the McGee Foundry Company, Long Island City. He was then made vice-president of the Long Island Foundry Company, later succeeding to the office of president. He had been an associate of the A.S.M.E. since 1921. His death occurred on May 22, 1931.

J. NORMAN BULKLEY

J. Norman Bulkley, consulting engineer, Johannesburg, South Africa, died on October 6, 1931. Mr. Bulkley was born at Memphis, Tenn., on September 17, 1867. He received an S.B. degree from the Massachusetts Institute of Technology in 1889, and during the remainder of that year was draftsman for the Wright Engineering Company of Boston. The next year he was employed as chief engineer by the Pennsylvania General Electric Company at Scranton; from 1894 to 1897 he was assistant engineer and in charge of mining machinery, designing machinery and supervising the erection of electric mining plants.

Mr. Bulkley went to Johannesburg in 1902 as chief engineer for the United Engineering Company, Ltd., a position which he held for seven years. He had engaged in general consulting practice in the mechanical and electrical fields since 1909. For five years he was consulting electrical engineer for the General Mining & Finance Corp. of Johannesburg. He spent several years in London and New York in consulting work and returned to South Africa in 1922. Among his clients were the African and European Investment Company, Guest Sykes & Chapman Ltd., Manganese Corporation, and other Johannesburg firms. Mr. Bulkley became a member of the A.S.M.E. in 1902.

STANLEY HALE BULLARD

Stanley Hale Bullard, vice-president and works manager of The Bullard Company, formerly The Bullard Machine Tool Company, of Bridgeport, Conn., died on March 22, 1931, at his home in Fairfield, Conn., of heart disease. A heart attack about a year earlier had forced him to give up his activities for some time but he had begun to renew them when he suffered another attack.

Mr. Bullard was the son of Edward Payson Bullard, the founder of The Bullard Company, and Alice Martha (Camp) Bullard, and was born on July 4, 1877, in Hoboken, N.J. After his graduation in 1895 from the New York Military Academy he became an apprentice machinist at the Bullard works and subsequently entered upon the sales work of the company. In 1901 he managed the Bullard exhibit at the Pan-American Exposition in Buffalo, N.Y. He served successively as manager of the New York office, manager of the Eastern Sales District, and general sales manager. He took up the duties of general works manager in 1915 and was made vice-president of the company three years later.

In 1917 Mr. Bullard served as vice-president and member of the management council of The Bullard Engineering Works, organized to manufacture 155 mm. G.P.F.'s. He developed the equipment and operating plans for this company and cooperated in the development of the Bullard boring mills, vertical turret lathes, and the Bullard Mult-Au-Matic. He contributed articles on machine tool manufacture and use to the technical press.

E. Kent Hubbard, president of the Connecticut Manufacturers' Association, issued the following statement on the death of Mr. Bullard:

"The industrial life of the state and the nation suffers in the death of Stanley H. Bullard. Identified as he was with a company which has made notable contributions to the advancement of modern machinery design and construction, his name is probably most closely associated with that trade. But his active interest in cooperative enterprises which characterize the business picture of today made him a recognized and respected figure wherever business men gathered. As a citizen, as a manufacturer, and as a cooperative spirit, his loss will be keenly felt."

The "cooperative enterprises" to which Mr. Hubbard refers were numerous and varied. Mr. Bullard was director of the United States Chamber of Commerce for two terms; first president of the National Association of State Chamber of Commerce Officials; president for four years of the Connecticut Chamber of Commerce; and vice-president of the Bridgeport Chamber of Commerce. He was an original member of the Connecticut Manufacturers' Association and at the time of his death was a member of its executive board and its special advisory committee. He was also an original member of the New England Council, formed about five years before his death, and had served on its executive committee and external relations committee. He had been a member of the Connecticut Division of the New England Governors Railway Advisory Commission, known as the Storrow Committee, which was appointed to make a survey of railway transportation in New England. He was also a member of the New England Shippers Advisory Board of the American Railway Association. Among the engineering societies to which he belonged were the A.S.M.E., which he joined in 1920, the Society of Automotive Engineers, the National Machine Tool Builders Association, and the Atlantic Deeper Waterways Association, of which he had been vice-president. He was vice-president of the Bridgeport Manufacturers' Association and a member of the Bridgeport Engineering Institute. He also participated in the civic affairs of Fairfield, being secretary of the Board of Education there and serving on the building committee of the board, and also being director of the Fairfield Bank & Trust Co. and the Home Building and Loan Association of Fairfield. His clubs included the Engineers' in New York, the Brooklawn and Algonquin in Bridgeport, and the Black Rock Yacht Club, Fairfield Beach Club, and Fairfield County Hunt Club, and he was a member of the Masonic fraternity.

Surviving Mr. Bullard are his widow, formerly Grace Cleaveland, of Bridgeport, whom he married in 1901, a son, Stanley Hale, Jr., and two daughters, Mrs. John Budd, of St. Paul, Minn., and Miss Katherine Bullard of Fairfield.

LOUIS STELLE BURBANK

Louis Stelle Burbank, who was injured in an automobile accident on October 12, 1930, died at the Milford (Mass.) hospital on January

6, 1931. He is survived by his widow, Grace (Wilson) Burbank, whom he married in 1905. She was also injured but was able to leave the hospital shortly before her husband's death. Also surviving him are two sons and a daughter, Davis E. Burbank, Worcester, Mass., Thurlow G. Burbank, New York, N.Y., and Mrs. Kester S. Lindsay, St. Louis, Mo.

Mr. Burbank was born at Lexington, Ky., on May 22, 1866, the son of Davis and Martha (Cox) Burbank. In addition to a year of study at Macalaster College, St. Paul, Minn., he had private instruction in technical subjects, and served a two-year apprenticeship as a machinist with C. M. Breuer, Fergus Falls, Minn. His early shop experience was secured with the St. Paul, Minneapolis & Milwaukee Ry. and Union Iron Works in Fergus Falls, Barnesville, and Minneapolis, Minn. He did some early work on the design of sugar and mining machinery and during 1899 was located in Boston, engaged on patent drawings and designing. A machine which he designed at this time for putting a string through the uppers of shoes and tying a knot in it was patented in 1893.

His first connection of importance was with the Cereal Machine Company, Worcester. For a number of years he designed all of the special machinery for this company, being located first in Boston and later in Worcester, where he was superintendent of machinery. Subsequent to this connection he was mechanical engineer for the Natural Food Company, Niagara Falls, N.Y. These two companies were forerunners of the present Shredded Wheat Company, "shredded wheat" being the name given to the product by Mr. Burbank.

From 1902 to 1908 Mr. Burbank engaged in mechanical engineering private practice in Niagara Falls, Worcester, Mass., and Phoenix, Md. In 1908 he served as president of the Corn Crystal Company, Worcester, and for the next nine years was mechanical engineer and assistant chief draftsman for the Draper Company, Hopedale, Mass. He resigned this position in 1916 to become mechanical engineer for the Ludlow Manufacturing Associates, Ludlow, Mass. In 1919 he was engaged as chief engineer of the Manville Manufacturing Company, Waterbury, Conn., and since that year had been with the Reed & Prince Mfg. Co., Worcester, as chief engineer since 1921.

Mr. Burbank became a member of the A.S.M.E. in 1898. He also belonged to the Engineering Society of Western Massachusetts and the Masonic fraternity.

WALTER SAFFORD BURKE

Walter Safford Burke, for some years inspector of the grounds and buildings of Harvard University, died on August 5, 1931. He is survived by his widow, Frances B. Burke, whom he married in 1893, and by a married daughter, Mrs. Ralph H. Wales.

Mr. Burke was born at Babcock Hill, Oneida County, N.Y., on December 30, 1866, the son of Albert G. and Clarissa (Booth) Burke. He was educated at the United States Naval Academy, from which he was graduated in 1887. He served in the Navy for ten years and then went to Harvard as instructor in the Mechanical Engineering Department. He was made an assistant professor of mechanical engineering in 1894. During the World War, he was stationed at the United States Navy Yard as Lieutenant-Commander, United States Navy.

In addition to his duties at Harvard, Mr. Burke was connected with the consulting firm of Burke & Hughes from 1906 to 1912.

Mr. Burke became a member of the A.S.M.E. in 1913.

NOBLE CHASE BUTLER, JR.

Noble Chase Butler, Jr., whose death occurred on June 7, 1931, was born at New Albany, Ind., on July 27, 1874, the son of Noble Chase Butler and Annie (Browning) Butler. After completing his work at the Indianapolis High School, he attended Purdue University and then entered Rose Polytechnic Institute, from which he was graduated with a B.S. degree in 1899. From then until 1916, he worked in the Testing Department of the Henry R. Worthington Hydraulic Works in Brooklyn, N.Y. He made graphical analyses of water-works engines to determine their most suitable parts, and also studied the arrangement of engines and accessories for plants. He made plans for the layout of water works and condenser installations and superintended the erection and testing of such equipment.

In 1916, Mr. Butler resigned from the Worthington company to handle New York sales for the Wilson-Snyder Company, manufacturers of all kinds of pumping equipment. In 1918, he superintended the erection of an oil line pumping station at the wells and a sea loading station at the coast in the Southern Oil Field of Mexico, for the Cortez Oil Company, as representative of the Wilson-Snyder Company. After completing this work, Mr. Butler was engaged as engineer in the Technical Department of the Vacuum Oil Company, New York. He analyzed lubrication requirements in central power stations and mills in various industries and made a study of machinery to determine the proper selection and application of lubricants. His work for the Standard Oil Company of New York, for which he became sales engineer in 1922, resulted in his appointment as chairman of its Committee of Industrial Lubricating Engineers, in which capacity he acted as chief engineer and consultant on all major lubricating problems both as to lubricants to be used and methods of application in all types of industries.

Mr. Butler first became a member of the A.S.M.E. in 1900. He resigned two years later but joined the Society again in 1927. He is survived by his widow, Ruth G. (Sears) Butler, whom he married in 1925.

WILLIAM ALFRED BUTLER

William Alfred Butler, who died on April 17, 1930, was born on January 19, 1868, at Cincinnati, Ohio. After completing his public school education, he went to work in 1886 as assistant to the general repairman of the C. H. Kilgour Narrow Gage Railway in Cincinnati. After four years in this work, he secured a position as chief engineer of the plant of Benjamin Hey & Co. in that city. From 1893 to 1895 he was night chief engineer of the Cincinnati Edison Electric Company, and for the next six years engineer for the Cincinnati Street Railway Company. In 1901, when the latter company was merged into the Cincinnati Traction Company, he was appointed superintendent of steam machinery repairs and later became superintendent of construction and maintenance of the five stations belonging to the company. In 1906 he was made chief engineer of the Pendleton Power Station of the company, where he remained for seven years. He was next connected in 1913 with the Duquesne Light Company of Pittsburgh as assistant superintendent of power stations. At the time of his death he was chief engineer of the Braun Brothers Baking Company of Pittsburgh.

He had been a member of the A.S.M.E. since 1915.

LEONARD CAHOON

Leonard Cahoon died of cancer at his home on April 1, 1931, after an illness of six months. Surviving him are his widow, Matilda (Watts) Cahoon, whom he married in 1909, and two daughters, Virginia and Janice. Mrs. Cahoon was a member of the Utah State Legislature in the session preceding her husband's death.

Mr. Cahoon was the son of Mary Erekson Cahoon, who also survives him, and Andrew Alonzo Cahoon, who died in 1924. He was born in Salt Lake City on December 10, 1881, and received his early education there. He studied mining and mechanical engineering at the University of Utah, which conferred a B.S. degree upon him in 1905. He was secretary of the board of directors of the University Club in Salt Lake City at the time of his death.

Following his graduation Mr. Cahoon gave five years to design and construction work for the U.S. Smelting, Refining & Mining Co. of Salt Lake City and Boston. Among the plants built under his supervision were rock crushing and loading plants in Utah, a smelter and bag houses at Kennett, Calif., smelter for the Needles Mining & Smelting Co. at Needles, Calif., as well as mining equipment for the company at its Kingman, Ariz., mines, and electrostatic concentrator baghouse and arsenic plants at Midvale, Utah.

In 1910 and 1911 Mr. Cahoon was engaged in the design and construction of a complete new hoisting plant for the Chief Consolidated Mining Company at Eureka, Utah, following which he superintended the construction of reinforced concrete and steel buildings for the James Stewart Company, including a theater building in Salt Lake City. During 1912 and 1913 he was placed on special design of haulage equipment and ore storage equipment for the International Smelting Company, Tooele City, Utah. Next, as a representative of the P. J. Moran Contracting Company, Salt Lake City, he superintended the construction of the Administration Building for the University of Utah. When this job was completed he took a position as engineer and superintendent of shops with the Galigher Machinery Company, Salt Lake City, with which he remained for about six years. He then went into business as sales manager and vice-president of the Stratton-Cahoon Company in Salt Lake City, western representative for the Weller Manufacturing Company, Republic Flow Meters Company, Erie City Iron Works, and Powers Specialty Company, handling equipment used in mines, factories, smelters, etc. He was later appointed western representative for the Foster Wheeler Corporation of New York, with headquarters at Salt Lake City, the position which he held at the time of his death.

Mr. Cahoon became a member of the A.S.M.E. in 1921. He was past-president of the Utah Society of Engineers and prominent in

politics, being a member of the Republican County Committee at the time of his death.

JAMES HENRY CALDWELL

James Henry Caldwell, chairman of the Board of Directors of the Ludlow Valve Manufacturing Corporation, Troy, N.Y., was born at Mobile, Ala., on March 21, 1865. His parents were Edward Holland and Carolina A. (Shields) Caldwell. He prepared for college at schools in Mobile, Ala., Salisbury, Md., and New York, N.Y. He was graduated from Rensselaer Polytechnic Institute, Troy, N.Y., in 1886 with a B.A. degree.

For eighteen years following his graduation, Mr. Caldwell was associated with the Mobile Gas & Electric Co. He engaged in consulting work and served as vice-president of the company. From then until his death on November 18, 1931, he was identified with the Ludlow company. He was made vice-president and general manager in 1894 and president in 1908.

Mr. Caldwell became a member of the A.S.M.E. in 1912. He also belonged to the Society of Professional Engineers of the State of New York, the American Foundrymen's Association, American Gas Institute, American Water Works Association, and the National Gas and Petroleum Association of Canada. He was an associate member of the American Museum of Natural History. Mr. Caldwell was elected president of the Troy Trust Company when it was organized in 1902. He resigned that position in 1913 and was serving as chairman of the board at the time of his death. He was treasurer of the Rensselaer Improvement Company, trustee and vice-president of the Troy Savings Bank, vice-president of the National State Bank of Troy, and of a number of railroads, including the Rensselaer & Saratoga, Albany & Vermont, and the Troy & Schenectady. He was vice-president and trustee of Rensselaer Polytechnic Institute and a dormitory of the Rensselaer campus was named for him. He had been active in the community affairs in Troy, serving as president of its Samaritan Hospital, of the Public Library, and of the Troy Chamber of Commerce. He married Margery Christie of Troy, in 1887, and is survived by two daughters.

ARTHUR LATHAM CHURCH

Arthur Latham Church, secretary and assistant treasurer of the Baldwin Locomotive Works, Philadelphia, Pa., died on June 25, 1931. Mr. Church was born in Philadelphia, on October 11, 1858, the son of William A. and Elizabeth (Barker) Church. His wife, Louise (Brant) Church, whom he married in 1888, died in 1929. He is survived by one son and by three grandchildren.

Mr. Church was graduated from the Towne Scientific School of the University of Pennsylvania in 1878, with a B.S. degree in mechanical engineering. The next four years were spent in the shops and drafting room of the William Cramp & Sons Ship & Engine Building Co. of Philadelphia. From 1882 to 1884, he served as electrician and oiler on the S.S. *Queen of the Pacific* and as assistant engineer on the S.S. *City of Peking* and S.S. *Granada*. He also did some work as designer for the Union Iron Works of San Francisco. During the next two years, Mr. Church was chief engineer of the Electrical and Novelties Exhibitions of The Franklin Institute and superintendent of The Spring Garden Institute. In 1886, he began his long connection with the Baldwin Locomotive Works as manager of the Department of Repairs. He was appointed assistant to the president in 1915, and had been secretary and assistant treasurer since 1920. He was always interested in shipbuilding, and from 1912 until it was dissolved in 1920, was manager of the Essington Shipbuilding Company, Ltd.

Mr. Church joined the A.S.M.E. in 1917. He was also a member of The Franklin Institute, the Pennsylvania Academy of Fine Arts, the Historical Society of Pennsylvania, and a number of clubs. He was a trustee of the University of Pennsylvania and a manager of the Preston Retreat, The Spring Garden Institute, and the University Hospital of Philadelphia. He was the author of "The Training of a Secretary."

BERTRAND JOSEPH CLERGUE

Bertrand Joseph Clergue, president of The Waterbury Tool Company, Waterbury, Conn., died on March 28, 1930. He was born in 1873 at Bangor, Me., where he received his early training in the repair shop of the Bangor Street Railway Company. He spent three years at the Massachusetts Institute of Technology, and in 1895 accepted a position with the Penobscot P. & P. Co., at Veazie, Me. The following year he became assistant manager of the Lake Superior Power Company, where he remained for three years. From 1899 to 1902 he was manager of the Algoma Central Railway, Michigan Lake Superior Power Company, and Tagona Water & Light Co. Subsequently he did consulting work at Sault Ste. Marie, Ontario, Canada, and at Montreal, and for a time was vice-president and general manager of Universal Transmission, Ltd. of Montreal.

He became an associate member of the A.S.M.E. in 1907.

CLARENCE E. CLEVELAND

Clarence E. Cleveland, president of the Intervale Mills, Inc., Webster, Mass., died at the home of his sister in Worcester, on May 8, 1931. Mr. Cleveland was born in that city on August 31, 1865, the son of Edwin Clarence and Anna (Goddard) Cleveland. He was graduated from the Worcester Polytechnic Institute with a B.S. degree in 1886. By 1892 he had become a designer at the Intervale Mills at Quinebaug, Conn., and subsequently was made a partner in this company. An additional larger mill was built in 1904 under Mr. Cleveland's supervision and a great deal of new machinery installed during the next few years. Mr. Cleveland had charge of practically all of the engineering work for the two plants of the company. He was appointed treasurer in 1918 and in 1924 went to Webster, Mass., when the firm was relocated in that city. He was appointed president the following year.

Mr. Cleveland became a member of the A.S.M.E. in 1916. He is survived by his widow, Gertrude (Stevens) Cleveland, whom he married in 1894, and by two sons and a daughter, as well as by his sister, Mrs. J. Isabel C. Wood.

BENJAMIN DENVER COPPAGE

Benjamin Denver Coppage, who died on October 6, 1931, was born at Sudlersville, Queen Anne County, Md., on April 13, 1872. He was the son of Edward Keene and Georgia E. (Crouch) Coppage. He supplemented his public school education with correspondence courses and night classes at the Y.M.C.A. His first position was with the Baldwin Locomotive Works as a helper in the erecting shop. In 1890 and 1891, he was draftsman for the Edge Moor Iron Company and the Newport News Ship Building & Dry Dock Co. He returned to the employ of the Edge Moor Iron Company in 1892 and remained with that company until 1907, serving as chief draftsman and superintendent. He designed improved forms for a number of types of boilers and, while in charge of the shops, designed and put into operation many improvements in equipment.

For ten years, beginning in 1907, Mr. Coppage was connected with the Pusey & Jones Co. During his first year with the company, he reorganized the engineering department. For the next six years, he was works manager in charge of the engineering and production departments. He was made chief engineer in 1914, and his duties in that capacity included supervision of all patent matters and consulting work for the Patents Owning Company, Inc. While Mr. Coppage was associated with the company it built paper-making machines for producing sheets from 170 in. to 204 in. Mr. Coppage was intimately connected with the design and construction of these machines. He also prepared plans for building machine shops and improving the equipment of the plant.

From 1917 to 1925, Mr. Coppage was connected successively with the E. I. du Pont de Nemours & Co., Wilmington, as experimental engineer, with the District of Columbia Paper Manufacturing Company, Washington, D.C., as chief engineer, and with the Levgar Structural Company, N.Y., as production engineer. Since 1925, he had been draftsman for the Viscose Company, Marcus Hook, Pa.

Mr. Coppage became a member of the A.S.M.E. in 1915. He also belonged to The Franklin Institute, which presented him with a "Certificate of Merit" in 1925 for his invention of a testing instrument known as the plastometer.

Surviving Mr. Coppage are his widow, Elsie B. (Johnson) Coppage, whom he married in 1898, and one daughter, Mabel Coppage.

WILLIAM VALDEMAR DAM

William Valdemar Dam, managing director of Thompsons Engineering & Pipe Co., Ltd., Castlemaine, Victoria, Australia, died at Castlemaine on February 12, 1931. He was born at Otahuhu, near Auckland, New Zealand, on November 18, 1864, the son of Edward Thomas Köller and Elizabeth (Corliss) Dam. After preparatory work in private and state schools, he attended the Workingmens College at Melbourne. In November, 1881, he began a six-year apprenticeship with the Bay of Islands Coal Company at Auckland, working particularly on the manufacture and repair of mining equipment.

His first position after completing his apprenticeship was with the Austral Otis Engineering Company, Ltd., of Melbourne, as an

engine fitter and erector. During the thirteen years of his connection with this company he held the positions of outside foreman, machine-shop foreman, draftsman, quantity surveyor, estimator, and assistant manager. These titles indicate the general character of his work, which dealt with all classes of hydraulic elevators and auxiliary machinery, as well as other hydraulic and mining equipment. For about a year and a half he was engaged in treating different ores mechanically and chemically and experimenting with new mining machinery, and for two years he inspected elevators in Melbourne, Sydney, Adelaide, and Brisbane.

In July, 1900, Mr. Dam was appointed works manager for Thompson & Co., Castlemaine, where he was promoted successively to designing engineer, assistant general manager, general manager, and managing director. He devoted special attention to the subject of mining and dredging machinery and was responsible for the design and construction of highly successful large steam winding engines and numerous important sluicing plants.

Mr. Dam visited America and Europe in 1919, and while in England completed arrangements for the manufacture by his firm in Australia of large steam turbines and their condensing plant under license to the Metroplitan Vickers Electrical Company, of Manchester, England.

Mr. Dam possessed a great inherent mechanical aptitude and an extraordinarily retentive memory and was regarded as one of the foremost mechanical engineers of Australia. He was elected a member of the A.S.M.E. in 1906, and for many years belonged to the Victorian Institute of Engineers.

Mr. Dam married Mary Findlay in 1914, and is survived by her and by one daughter by a former marriage.

CHARLES DAY

Charles Day, chairman of the board of Day & Zimmermann, Inc., Philadelphia, Pa., died at the University Hospital in that city on May 10, 1931, following an illness of ten days. An eye disorder, which had developed a few days after his return from a trip to North Africa and which was at first attributed to the effects of the tropical sun, proved to be an organic affliction.

Mr. Day was born in Philadelphia on May 15, 1879, the son of Richard H. and Frances (Corson) Day. He attended the Germantown Academy and was graduated from the University of Pennsylvania with a B.S. degree in 1899. An M.E. degree was conferred upon him in 1903. He was superintendent of installation of power plant equipment and transmission machinery for the National Export Exposition at Philadelphia during the latter part of 1899 and then spent about two years as assistant superintendent and engineer of works for the Link Belt Engineering Company at Nicetown, Philadelphia.

In 1903 the engineering firm of Dodge & Day, specializing in management and construction work, was established. It became Day & Zimmermann, Inc., in 1911, with Mr. Day as president from 1916 until 1926, and chairman of the board since that time. Through his work with this firm Mr. Day became widely known as an authority on the construction and efficient equipment of industrial plants and the management of public service corporations. He lectured on these subjects before the Graduate School of Business Administration at Harvard, and at Columbia and Lehigh universities. He served as president, vice-president, or director of a very large number of public service companies, including railway, light, heat, and power. He was the author of many articles on industrial management, and in 1911 published a book on "Industrial Plants."

Mr. Day was called upon by the Government to serve in a number of important posts. He was made a trustee of the Emergency Fleet Corporation at the beginning of the World War and was appointed a member of a civilian board to investigate the efficiency of the navy yards. During the War he was a member of the storage committee of the General Munitions Board and conferred in France with General Pershing on means of expediting the transportation of troops and supplies to Europe. He was appointed to the Depot Board by the Secretary of War and was present at the War Council of the Allies in the winter of 1917. He was also engineering adviser to Colonel Edward M. House's commission.

Mr. Day became a junior member of the A.S.M.E. in 1902 and a member six years later. He was a member of the Bureau of Engineering Standards in 1913 and 1914 and of the Standardization Committee of the Society during the next two years. He also served from 1913 to 1917 as a member of the Subcommittee (of the Committee on Meetings) on Industrial Buildings.

Among the other societies to which he belonged were the American Institute of Electrical Engineers, the American Philosophical Society, the Taylor Society, and the Society for the Promotion of Engineering Education. He was a member of the board of managers of The Franklin Institute, a trustee of the University of Pennsylvania, and a director of the Pennsylvania Railroad and Fidelity-Philadelphia Trust Company. His clubs included the Racquet, the Engineers' Club of Philadelphia, the Engineers' Club of New York, the Philadelphia Cricket Club, Sunnybrook Golf, University and Rittenhouse Clubs.

Surviving Mr. Day are his widow, Margaret (Dunning) Day, whom he married in 1904, two sons, William L. and Charles C., and a daughter, Margaret D. Day.

VAN NESS DeLAMATER

Van Ness DeLamater, since 1915 associated with the Hyatt Roller Bearing Company, died on August 29, 1931. It is believed that he and his son, Van Ness, Jr., were drowned when their canoe overturned during a sudden storm on Green Lake, Algonquin Provincial Park, Ontario.

Mr. DeLamater was born at Hudson, N.Y., on July 3, 1878, a son of Ezra Doane and Anna (Van Ness) DeLamater. He attended high school in Hudson and Ithaca, N.Y., and was graduated from Cornell University in 1900 with an M.E. degree. He continued his studies in engineering and law at Cornell for a year and then served an apprenticeship at the Mt. Clare shops of the Baltimore & Ohio R.R., Baltimore, Md. During the next two years he was connected with the department of tests of that railroad, inspecting material and new equipment and conducting laboratory investigations. He was next employed for about six months by the American Steel Foundries Company, testing and reorganizing power plants in foundries. In 1905 and 1906, he was salesman in the New York district for the Star Brass Manufacturing Company. During the next two years he was employed by the Central Railroad of New Jersey, first as inspector of the motive power department, testing locomotives and power plants and designing new car shops, and later as acting assistant chief draftsman in the mechanical department in Jersey City. From 1908 to 1912 he was chief draftsman and mechanical engineer for the Rushmore Dynamo Works, Plainfield, N.J. He designed brass goods and tools, installed and operated producer gas and engine plants, supervised the construction and installation of equipment of concrete factory buildings, and developed direct-current and storage-battery lighting systems for automobiles.

Mr. DeLamater went to Winnipeg in 1912 as sales manager for the Canadian Fairbanks Morse Company, resigning that position in 1915 to become assistant sales manager in the motor bearings division at Detroit of the Hyatt Roller Bearing Company. He was transferred to Harrison, N.J., as assistant to the vice-president of that company in 1920 and since 1924 had been located at the Newark office, specializing in railroad bearing applications.

Mr. DeLamater became a member of the A.S.M.E. in 1913. He is survived by his widow, Jacqueline (Newton) DeLamater, formerly of California, whom he married in 1904, and by two sons and two daughters.

CONRAD G. DIETZ

Conrad G. Dietz, vice-president and general manager of the Aeronautical Corporation of America, Lunken Airport, Cincinnati, Ohio, died on September 12, 1931, as the result of an aeroplane accident. Mr. Dietz was born in Cincinnati on May 13, 1900, the son of Conrad and Margaretta Dietz. He attended the Asheville Preparatory School, Asheville, N.C., and was graduated from the Ohio Mechanics Institute in 1914.

The next six years were spent in traveling throughout the western part of the United States and operating an automobile sales and service agency in Cincinnati. For eight years, beginning in 1920, he had complete charge of all work at the Dietz Experimental Laboratories, Dayton, Ohio, of which he was president. He had been connected with the Aeronautical Corporation of America since 1928. His specialty was the design of aircraft and aircraft motors. Among the planes which he built at the Dietz Experimental Laboratories were a three-place biplane and a four-place biplane, both powered with a model E Hispano-Suiza motor, and a single-place racing biplane, powered with a Wright J5 motor. The Aeronautical Corporation of America, which was organized by Mr. Dietz, has developed and constructed a number of monoplanes, seaplanes, and motors.

Mr. Dietz became an associate member of the A.S.M.E. only a few months before his death. Surviving him are his widow, Harriet (Otte) Dietz, and two sons, Robert P. and James C. Dietz.

FRANK BUTLER DILTS

Frank Butler Dilts, lifelong resident of Fulton, N.Y., died at his home on September 17, 1931, after an illness of several months. Mr. Dilts was president of the Dilts' Machine Works, Fulton, manu-

facturers of paper mill machinery. He was also vice-president and director of the Citizen's National Bank & Trust Co., vice-president of the Draycott Paper Mills Co., Inc., and affiliated in an official capacity with the Arrowhead mills, all of Fulton.

Mr. Dilts was born on February 14, 1874. He received his early education in the Fulton schools and then took up the study of mechanical engineering at Cornell University. He left college in his junior year following the death of his father, Frank Dilts, then head of the Dilts Machine Works, Inc., and assumed active charge of the foundry and machine works. He was made president and treasurer in 1913. Mr. Dilts supervised the design work of the company, which included plans for new buildings and equipment, and water and electric power development. He laid out and arranged for the installation of machinery in paper mills, and was the first to conceive the idea of tempering paper mill beater roll bars and of using heat-treated alloy steels for this purpose. His company was also the first to market a bandless beater roll and a beater tub designed for a concrete base.

Mr. Dilts was a Republican and took an active interest in civic affairs in Fulton, serving as alderman of the Fourth ward from 1904 to 1907. He was also several years as officer of the Mount Adnah cemetery association. Mr. Dilts became a member of the A.S.M.E. in 1918.

His preference in sports was toward golf and he was a past-president of the Oswego country club and a member of the Onondaga golf and country club. During the World War, Mr. Dilts served as a Captain in the Ordnance Department and was stationed at the Watervliet Arsenal, Watervliet, N.Y. He served for some years as a member of the Fulton Hydraulic Race Commission, appointed by the Supreme Court, in connection with which his knowledge of engineering was applied to problems involved in the division of water among the various users of water power.

He is survived by his widow, Mrs. Bessie (Smith) Dilts.

ALVA CLYMER DINKEY

Alva Clymer Dinkey, president of The Midvale Company, died at his home in Wynnewood, Pa., on August 11, 1931.

He was born at Weatherly, Pa., on February 20, 1866, the son of Reuben and Elizabeth (Hamm) Dinkey, and attended the public schools of Weatherly and Braddock, Pa. In 1879, at the age of thirteen, he entered the Carnegie service as a water boy at the Edgar Thomson Steel Works, later serving as telegraph operator. In 1885 he became machinist for the Pittsburgh Locomotive Works and in 1888 expert machinist for the McTighe Electric Company, Pittsburgh.

He was appointed secretary to the superintendent of the Homestead Steel Works of the Carnegie Steel Company in 1889 and in 1893 was made electrician of the Homestead plant; in 1898 he became superintendent of the electric light and power plant. In 1899 he was appointed assistant general superintendent of the Homestead Steel Works, and in 1901, general superintendent.

In 1903, at the age of thirty-seven, Mr. Dinkey was elected president of the Carnegie Steel Company, retaining that position until 1915, when he resigned to assume the presidency of the newly formed Midvale Steel & Ordnance Co. When this latter company was sold, in 1923, to the Bethlehem Steel Corporation, he was elected president of The Midvale Company. He was a director of the Midvale Company, The Baldwin Locomotive Works, and the American Iron and Steel Institute.

Mr. Dinkey became a member of the A.S.M.E. in 1902. He also belonged to the American Institute of Electrical Engineers, the Society of Naval Architects and Marine Engineers, the Engineers Club of Philadelphia, the Engineers Society of Western Pennsylvania, the American Association for the Advancement of Science, and the Academy of Natural Sciences, as well as to many clubs in New York, Philadelphia, and Pittsburgh.

Surviving Mr. Dinkey are his widow, Margaret (Stewart) Dinkey, whom he married in 1891, two sons, and a daughter.

JOSEPH DOVE-SMITH

Joseph Dove-Smith, who died on March 25, 1931, was born at Newcastle-on-Tyne, England, on January 19, 1855. His parents were Joseph and Mary Ann (Asquith) Dove-Smith. He had a public school education and served an apprenticeship with Bolckow Vaughan & Co., iron and steel manufacturers. After experience as a fitter, turner, and draftsman with various engineering works, he became general foreman in 1885 of the Vulcan Iron Works, Winchester, England. He went to Manchester five years later as general foreman of Samuel Fisher & Co. In 1892 he accepted a position as general manager for John Holroyd & Co. in Manchester, where he remained for four years. He then went to Birmingham as works manager for Heenan and Froude. From 1898 to 1903 he was mechanical engineer for the London Rubber Company.

Mr. Dove-Smith left England for Canada in 1903 and for five years worked as a machinist, part of the time in charge of outside mechanical construction, for the Allis Chalmers Bullock Company, Montreal. He founded the firm of Dove-Smith & Son in 1908 at Niagara Falls to design and build special machinery and tools. During the World War the entire shop was devoted to the manufacture of master inspection gages for the Canadian and United States governments. He held patents in England for a rubber heel and in the United States and Canada for all-in-mesh transmission.

Mr. Dove-Smith became a member of the A.S.M.E. in 1918. He also belonged to the Society of American Military Engineers and was a director in 1924 of the Rotary Club. He was interested in all church work, especially small missions, and was an enthusiastic choral director of children. He is survived by his widow, Phoebe (Bell) Dove-Smith, whom he married in 1879, and by five daughters and a son.

ERVIN THAYER DRAKE, JR.

Ervin Thayer Drake, Jr., plant engineer for the Lever Bros. Co., Hammond, Ind., died on July 27, 1931. Mr. Drake was born on November 3, 1894, the son of Dr. E. T. Drake and Mary Aiken Drake, of Franklin, N.H. He attended the Cambridge (Mass.) Latin School and was graduated from Harvard University in 1916, with a B.S. degree.

Immediately following his graduation, he went to France and joined the Norton Hayes Ambulance Corps. He was discharged from the Army in the fall of 1919, with the rank of captain, and entered the soap plant of the Jones Bros. Tea Co. in Brooklyn, N.Y., for a six month's period of training in that buineass. He then secured a position as engineer with the Houchin-Aiken Company, Brooklyn, working on the design, construction, and installation of mechanical equipment for the production of soaps and allied materials. He also developed chemical processes for manufacturing soap, and equipment for the recovery of the by-products, especially crude and refined glycerine. In 1922 he laid out and supervised the construction of a complete soap plant and double-effect evaporators in Madrid, Spain. In 1923 he installed large crude glycerine evaporators in Havana, Cuba. During the next two years, he installed soap and glycerine plants at Middletown, Conn., Woonsocket, R.I., and Baltimore, Md. Later he was located at the plant of the company at Hawthorne, N.J.

When the Houchin-Aiken partnership was dissolved in 1926, Mr. Drake entered the employ of R. A. Keashey, New York, N.Y., selling commercial asbestos. He had been associated with the Lever Bros. Co. since February, 1928. He acted as construction engineer on the Hammond plant of the company, and during the last year of his life had served as plant engineer.

Mr. Drake became a junior member of the A.S.M.E. in 1923 and an associate of the Society two years later. He is survived by his widow, formerly Elizabeth K. Lum, of Chatham, N.H., whom he married in 1921, and by three sons, E. T. Drake, III, and Charles Lum and Herrick Aiken Drake.

THOMAS ALVA EDISON

Thomas Alva Edison was born at Milan, Ohio, on February 11, 1847. On the paternal side he was descended from John Edison, who came to New York from Holland in 1728. His great-grandfather, Thomas Edison, was a banker, and signed the Continental currency in 1778. John Edison went to Canada in 1783 and his son, Samuel, father of Thomas Alva Edison, was born at Digby, Nova Scotia, in 1804. He married Nancy Elliott in 1828 and they lived in Canada until 1838, when he became involved in the Papineau rebellion and found it necessary to leave the country. They came to the United States and in 1842 settled at Milan.

Thomas Edison's mother was a native of New England and had taught school in Canada prior to her marriage. Largely to her he owed his early education. She took him out of school, where he was considered stupid and stood at the foot of his class, and not only succeeded in developing a love of study and reading but also encouraged him in the experimental work which then, as always, was his chief interest.

These early experiments of Edison's were for the most part of a chemical nature, and throughout his life he retained his love for chemical research, though circumstances directed so large a part of his effort into the electrical field.

The Edison family, it is pointed out in the biography of his life by Dyer and Martin, one of the most authoritative works, while not

rich, were in comfortable circumstances. It was not to support himself that Edison became a newsboy at the age of twelve, but to earn additional pocket-money with which to purchase materials for his experiments. The family had moved to Port Huron, Mich., in 1854, and finally securing the consent of his parents, Edison applied, in 1859, for the privilege of selling newspapers on the trains of the Grand Trunk Railroad, between Port Huron and Detroit.

The details of these early years need not be recorded here. Selling newspapers, books, candy, and fruit gave Edison what money he required for experimenting. He set up a laboratory in the baggage car of the train, and all of his leisure time was spent there and in the Detroit public library, where he read many books and magazines otherwise inaccessible to him. During the early days of the Civil War he further increased his revenues and experience by issuing a small newspaper, for which he wrote the copy and set the type, and which was printed on a handpress in the baggage car.

During this period Edison had also become interested in telegraphy and to this he turned when a fire in his train laboratory one day led to his abrupt ejection on the Mount Clemens platform, with his entire outfit. The station agent there, whose child Edison had rescued from the path of a train, had already taught him train telegraphy, and this experience enabled him to secure the post of operator at Port Huron. Subsequently, he applied for a job as night operator for the Grand Trunk at Stratford Junction, Canada, and in the cellar below that office he spent nearly all his time in chemical experiments and in reading. It was at Stratford, in 1863, that he made his first invention—a device to send automatically the hourly signals which he was required to make to the central office to show that he was on duty. This device, until detected, permitted him to remain in his workshop uninterruptedly!

Records of the next five years show Edison to have been a roving operator in various western and southern cities, including Toledo, Indianapolis, Cincinnati, Louisville, Memphis, and New Orleans. He worked on night shifts whenever possible, as that arrangement gave him more time and freedom for experiments and reading. He developed unusual speed and accuracy as an operator and frequently made improvements in the equipment which he used. One of his first important inventions was a repeating instrument which permitted a message to be transmitted automatically on a second line without the presence of an operator.

Edison's restlessness led him East in 1868. A friend in Boston helped him to secure a night line with the Western Union, where his speed and general ability were soon recognized. He established a small workshop and spent the most of his salary on books and apparatus. He applied for his first patent in October of that year, to cover a device by means of which the votes of a large number of people might be recorded in a few minutes.

It was in Boston also that Edison invented his first stock ticker and started a ticker service. He visited New York during the latter part of 1868 in an unsuccessful attempt to sell the ticker machine. Later he developed a duplex telegraph which he took to Rochester, N.Y., to test out on the lines of the Atlantic & Pacific Telegraph between that city and New York. He was unsuccessful in this also and in 1869 went to New York, leaving behind him because of debts what few instruments and books he possessed.

Transition within a few days from this impoverished state to a position earning $300 a month was the result of his understanding of ticker apparatus. While waiting to secure a position as operator with the Western Union, Edison took the opportunity to study the central transmitter in the office of the Gold & Stock Telegraph Co. and happened to be present at the time of a breakdown in service and was able to detect the cause of the trouble. This evidence of his ability led to the offer of a position as superintendent of the entire plant.

Before the end of the year Edison had associated himself with Franklin L. Pope in a "Bureau of Electrical and Telegraphic Engineering." In a small shop in Jersey City he followed up his work on telegraph printers, devising a printer to print gold quotations instead of indicating them. This was sold to the Gold & Stock Telegraph Co., which also took over private telegraph lines which were established by Pope, Edison, and J. N. Ashley, who had become a partner in the firm.

Edison was now definitely launched upon that period of his career so productive that the Commissioner of Patents said, "There is a young man in New Jersey who has made the path to the Patent Office hot." The development and manufacture of stock tickers for the Gold & Stock Telegraph Co. necessitated the establishment of several shops in Newark and the employment of about 300 men. Before he was thirty years old Edison had also perfected the alignment of letters on what is now the Remington typewriter, made many contributions to the development of automatic, duplex, and quadruplex telegraphy, and invented the mimeograph.

On December 25, 1871, Edison married Miss Mary G. Stillwell, and in 1876 they went to live at Menlo Park, N.J., where he established a new laboratory such as he had long desired. Out of this laboratory and adjoining workshops during the next ten years came the carbon transmitter for the telephone in 1876, the phonograph in 1877, the incandescent lamp in 1879, the discovery of the "Edison effect," the fundamental principle on which the radio tube is based, in 1883, and many other inventions of importance.

Concerning the incandescent lamp Dr. Frank B. Jewett wrote as follows in *Science* (January 15, 1932): "For Edison the incandescent lamp was but an incident. . . . As an inventor and experimentalist he had produced an ingenious and potentially valuable illuminating device, but the electrical art of 1879 was almost devoid of the things to make it commercially available. Seldom has the creator of so perfect a new tool found the field so bare. More seldom still has one done so much to create an art around and for his tool as Edison did in the ten years following 1879.

"Generators, systems of distribution and utilization, switchboards, auxiliary equipment and adjunct devices, large and small, in vast profusion flowed unceasingly from Edison and the group of intelligent young men with whom he surrounded himself. In the welter of this torrent of new things no one will probably ever know just who was initially responsible for each and every one of them. Three things are certain, however: first, Edison himself was the creator of the principal items; second, his was at all times the guiding mind and the driving force; third, the group of men who later came to be known as Edison Pioneers, and who were and in large measure still are dominating factors in the electric light and power industry, were youths of his discerning selection."

The first Mrs. Edison died in 1884, leaving him three children, Marion Estelle, Thomas Alva, Jr., and William Leslie. He was married a second time in 1886, to Miss Mina Miller, and in 1887 established a new and more elaborate laboratory in Orange, N.J., where all his later work was done. Much of this work, which brought the record of his inventions at the Patent Office to more than 1200, related to the application of his earlier inventions. There were several other outstanding achievements, however, which should be mentioned in even a brief record. These include the motion picture camera, patented in 1891, and the kinetophone, patented in 1912, and his contributions to the portland cement industry and to the development of storage batteries.

During the World War Edison turned all his energy to work for the Naval Consulting Board, of which he was made president in 1915. He and his associates worked on many problems which again, after many years, gave him an opportunity to work with chemicals. The manufacture of carbolic acid and aniline dyes, and the perfection of many devices for the Navy, made his services of tremendous value to the country.

It was during the War that Mr. Edison became interested in the rubber supply of the United States, and his research work during the last few years of his life dealt largely with that problem.

With Mr. Edison at the time of his death on October 18, 1931, at his home in Llewellyn Park, West Orange, were Mrs. Edison and their three children, Charles, Madeline, and Theodore, as well as the three children by his former marriage. Much of the tranquillity of the latter years of his life was due to the care of Mrs. Edison, whose voice was the only one able to pierce the deafness which had afflicted him for many years.

Edison's achievements were recognized in many honors bestowed upon him. Among these were the Rathenau Medal from Germany, the John Fritz Medal in 1908, and the Congressional Medal in 1928; three degrees of the Legion of Honor of France—Chevalier, Officier, and Commander; honorary degrees of Ph.D. and D.Sc. from Rutgers College, Ph.D. from Union College, D.Sc. from Princeton University, and LL.D. from the University of the State of New York; and honorary memberships in The American Society of Mechanical Engineers (in 1904) and in the American Society of Civil Engineers and the Engineers' Club, New York. He became a member of the A.S.M.E. in 1880, was a Founder Member of the American Institute of Electrical Engineers, and also belonged to the American Institute of Mining and Metallurgical Engineers.

Again and again in the tributes to Mr. Edison following his death his characteristics as an inventor and as a man are brought out:

"Mr. Edison became a great man by virtue of three moral qualities—indomitable courage, keen intelligence, and the will to serve. Widely as his mind ranged, he had the rare power of focusing it on the question in hand and sticking to it till he reached a solution."

"Edison was by far the most successful and, probably, the last exponent of the purely empirical method of investigation. Everything he achieved was the result of persistent trials and experiments often performed at random but always attesting extraordinary vigor and resource. Starting from a few known elements, he would make

their combinations and permutations, tabulate them and run through the whole list, completing test after test with incredible rapidity until he obtained a clue. His mind was dominated by but one idea, to leave no stone unturned, to exhaust every possibility."

"His place is assured as one of the world's greatest benefactors, for the results of his inventive genius have contributed immeasurably to the advancement, comfort, and happiness of mankind. But aside from the splendor of his achievements, the teachings of his life, his tireless pursuit of scientific truth, his all-embracing thoroughness in research, his intelligence, energy, and persistency in dealing with the mighty problems that engaged his eager interest, constitute a legacy of priceless value to the youth of all lands."

VICTOR EVERETT EDWARDS

Victor Everett Edwards, whose invention of flying shears permitted continuous rolling of steel rods, died at his home in West Boylston, Mass., on May 16, 1931. Illness necessitated his retirement in 1928 from the Morgan Construction Company, Worcester, Mass., of which he was vice-president, chief engineer, and director.

Victor Edwards was born at North Chelmsford, Mass., on September 4, 1862, the son of Dr. Nathan Brown and Sibbyl Robbins (Hutchins) Edwards. He attended the North Chelmsford school and was graduated from the Worcester Polytechnic Institute in 1883 with a B.S. degree. In 1927 the Institute conferred the degree of doctor of engineering upon him.

During the seven years following his graduation Mr. Edwards acquired knowledge of motive power and the design of heavy machinery through association with a number of companies. He was first connected with the Merrimack Manufacturing Company, Lowell, Mass., and then with the Deane Steam Pump Company, Holyoke, Mass. He spent about two years with the Washburn & Moen Mfg. Co., Worcester, working on economy testing of engines and boilers part of the time. For about six months he studied water-wheel settings for the Holyoke Machine Company, Worcester. He was in the employ of the Simonds Rolling Machine Company, Fitchburg, Mass., for two and one-half years, setting up and taking charge of their Canadian shop for part of the time, and being assistant superintendent for the company at Fitchburg during the remainder of it. He also served as draftsman for the Otis Steel Co., Ltd., Cleveland, for a period of six months, and assisted Edward Sawyer, of Boston, for a short time.

In 1890 Mr. Edwards accepted a position as chief engineer for the Morgan Construction Company and was elected vice-president the following year. He was also a director of the Massachusetts Mutual Life Insurance Company and had served as a trustee of Worcester Polytechnic Institute. During the World War he was commissioned a major in the U. S. Ordnance Department, being stationed at Washington and the Aberdeen Proving Grounds.

Mr. Edwards held more than one hundred United States and foreign patents, dealing for the most part with the manufacture of steel. The most important of these, the flying shears, were designed to travel along with the hot rods as they emerged from the rolls and cut them while in motion. This device, eliminating the time and expense of stopping the rods for cutting, effected a great saving in the cost of their manufacture. As a hobby Mr. Edwards designed sun-dials.

Mr. Edwards became a junior member of the A.S.M.E. in 1889 and participated in the European trip of the Society that summer. He became a member in full the following year. He was also a member of the American Institute of Mining and Metallurgical Engineers and the American Iron and Steel Institute. His clubs included the Engineers' in New York and Duquesne in Pittsburgh, Pa. He was a member and formerly a trustee of the West Boylston Congregational Church.

Surviving Mr. Edwards are his widow, formerly Janet Gage, of Skowhegan, Me., whom he married in 1896, a son, Kenneth Gage, and a daughter, Ruth Edwards.

HUGH MacDONALD ELDER

Hugh MacDonald Elder died on July 30, 1931, at his home in Fairlie, Ayrshire, Scotland, where he had been since 1927.

Mr. Elder was born on October 21, 1865, the son of George and Jane (MacDonald) Elder. He served an apprenticeship with Alley & McLellan, engineers, boilermakers, and shipbuilders, at their Sentinel Works in Glasgow. He then spent four years in technical studies at the London and City Guilds which granted him a certificate in mechanical engineering in 1890.

For a short time, following the completion of his technical education, Mr. Elder was in charge of an experimental electric light station, at Cathchart, near Glasgow. From then until 1904, with the exception of some time spent in the West Indies and Central America, he was engaged in marine engineering, securing both British and American certificates. His work in the West Indies was as master mechanic of railway shops for the San Domingo Improvement Company, Puerto Plata, San Domingo, and in Central America as contracting consulting engineer, constructing and rebuilding sugar mills, refrigerating plants, pumping machinery, etc., in Guatemala.

In the summer of 1904 he became assistant to the superintendent of the Inspection Department of the Casualty Company of America. In 1907 he was appointed chief engineer for the Trinity Building in New York and two years later took a similar position in the Abraham & Strauss and Simpson Crawford Company stores.

In 1911 Mr. Elder began a connection of more than fifteen years with the American Locomotive Company. He had charge of the power departments in various plants in the United States and in Canada until 1920, when he was transferred to the New York office. In 1924 he was made superintendent of power plants for the company.

Mr. Elder became a member of the A.S.M.E. in 1912. He is survived by his widow, Mary S. (Common) Elder, whom he married in 1904.

HARRINGTON EMERSON

Harrington Emerson, who died in New York, N.Y., on May 23, 1931, is recognized as a pioneer in the popularization of scientific management. Under the term "efficiency" he applied its principles in 1904 to the problems of the Santa Fé Railroad, in a form which caught the public imagination. He defined efficiency as "the elimination of all needless wastes, in material, in labor, and in equipment, so as to reduce costs, increase profits, and raise wages." He applied his theory to a wide variety of industries with startling results, bringing about standardization, cost control, and waste elimination. He met with no serious opposition from employees because he maintained that no improvement could be made permanent unless it helped the employee as well as the employer. He placed increasing emphasis, as the years passed, upon the personal side of efficiency. His ideals for the workman were summarized as follows in his book entitled "The Twelve Principles of Efficiency:" "Let each man work with the reliability of a steam valve, yet with the joy of a hunting dog and the inspiration of the artist." It was this spirit of a crusader and a moralist, together with a keen appreciation of the commercial opportunities presented by the movement, which gave him so large a following.

Harrington Emerson was born on August 2, 1853, at Trenton, N.J., the son of the Rev. Edwin Emerson, a Presbyterian preacher and educator, and Mary Louise (Ingham) Emerson. When he was nine years old the family went abroad in order that the children might be educated there. Harrington Emerson grew up with tutors and in schools in Greece, France, Italy, Germany, England, Switzerland, and Austria. He received an M.A. degree from the Royal Bavarian Polytechnic at Munich, where he was a classmate of the inventor of the Diesel engine.

When Mr. Emerson returned to the United States in 1876, at the age of twenty-three, he was given the chair of modern languages and literature at the University of Nebraska. Subsequently he was appointed registrar and secretary of the faculty. He taught at the university for six years, during which time he was active in developing the institution along more efficient lines.

Between 1882 and 1900, when he established a consulting office in New York, Mr. Emerson built the background for the work which he carried on during the remainder of his life. As a real estate dealer in the Middle West he acquired both wealth and a knowledge of the frontier of that day which led to his appointment as a member of the staff of the general manager of the Chicago, Burlington, and Quincy Railroad. The selection of town sites, excursions of settlers, prospects of coal mines, crop failures, and other problems relating to the extension of the frontier were turned over to him during the several years he served the road.

After about ten years in frontier and railroad life Mr. Emerson consented to act as representative of a British syndicate seeking American investments. In this connection he investigated many mills, factories, and mines in the United States, Mexico, and Canada, and began to discover the causes and remedies of failure. This work, perhaps more than any other, laid the basis for his principles of efficiency.

With the discovery of gold in Alaska Mr. Emerson interested himself in the problems of transportation and communication involved. He spent some time in the Yukon, driving dog sleds himself and studying the conditions of the country. He was instrumental in placing in service the first United States mail routes in Alaska, opening the line for both summer and winter delivery from Juneau to St. Michael. He made a report on the northern submarine cable route to Asia

which was largely followed by the United States War Department in laying its Alaskan cables.

Mr. Emerson had already achieved considerable success in discovering and eliminating wastes and losses in industrial plants and transportation enterprises when he was called to Atchison in 1904 at the time of the machinists' strike on the Santa Fé road. The problem was four-fold: to continue the repair and renewals of the rolling-stock and motive power in spite of the strike; to take care of a sudden 40 per cent increase in business without new shops or equipment; to establish friendly relations between employer and employee where there had been dissatisfaction and unrest for a long time; and, although of less importance, to decrease unit costs. Even though his system was still in its formative stage Emerson made recommendations which were successfully carried out. Not only that, but his principles of efficiency, brought before the public in connection with this job, created wide interest. To handle consulting work of increasing scope and importance Emerson organized a group of associates, The Emerson Engineers, who under his guidance have made contributions of inestimable value to the development of scientific management. He withdrew from active association with the organization about five years before his death.

Emerson's work extended to many countries. During recent years he had served as counselor to the Ministry of Railroads of the National Government of China and had been called in consultation by Japan, Poland, Czechoslovakia, and South American governments. In 1930 he was preparing reports on unemployment for departments of the French and Canadian governments. He had but recently come home from Russia where he was advisor to the Soviet government regarding its contemplated continental railroad system, and was planning to return for further work.

His success in the elimination of waste led to his appointment in 1921 as a member of the Committee on the Elimination of Waste in Industry, called together by Herbert Hoover when he was president of the Federated American Engineering Societies.

The principles of efficiency for which he stood were set forth by Emerson in book form in 1909 in "Efficiency as a Basis for Operation and Wages," and in 1912 in "The Twelve Principles of Efficiency." His "Lessons in Personal Efficiency" have been widely studied and his magazine articles and lectures before scientific organizations were numerous.

The technical societies of which Emerson was a member were many. He joined the A.S.M.E., of which Taylor and Gantt, and later Gilbreth, were fellow members, in 1899. He belonged to the American Institute of Mining and Metallurgical Engineers and was keenly interested in problems of over-production in the mineral industry. The American Electrochemical Society, The American Economic Association, the American Electric Railway Association, the Society of Industrial Engineers, and The Franklin Institute in this country, as well as the Société des Ingénieurs Civils de France, the Verein deutscher Ingenieure, and Polish and Czechoslovakian engineering societies, had all been honored by his membership.

At a notable testimonial dinner in Washington in October, 1930, a group of his associates recognized his achievements by presenting the Society of Industrial Engineers with a plaque designated the Harrington Emerson Trophy, to serve as an annual chapter award of the society.

What Harrington Emerson achieved and those qualities which underlay his success were summarized by Edward J. Mehren in the June, 1931, issue of *Factory and Industrial Management*, as follows:

"He made a full contribution to the art of management. His brilliant powers of analysis, his resourcefulness, his ability to cut through precedent led him to original solutions and marked him a great engineer. Much of his technique, though necessarily changing with conditions, remains a permanent asset in the literature of management.

"But conspicuous as was his engineering achievement, his work as teacher and philosopher will probably stand as his greatest contribution. He had been a college professor. He knew how to clarify and present ideas; he knew how to draw analogy. And that is just what he did—and did brilliantly—for the developing art of management. He carried its lessons to every corner of the industrial world, here and overseas.

"His success as teacher—as publicist of the management movement—was grounded deep. For he was a philosopher. He was interested in fundamental causes. He sought always the principles involved. His twelve principles of efficiency stand today a conspicuous expression of the philosophy of management."

DWIGHT HALL FAULKNER

Dwight Hall Faulkner was born at Ashtabula, Ohio, on September 16, 1895, the son of John C. and Mary H. Faulkner. He studied at the Manual Training High School in Brooklyn from 1909 to 1913 and then entered the Sheffield Scientific School of Yale University which granted him a Ph.B. degree.

During the World War Mr. Faulkner was senior inspector of aviation engines in the Signal Corps of the United States Army. After leaving the service he accepted a position in 1919 as an erecting engineer of the Bethlehem Steel Corporation, Bethlehem, Pa. Later he did similar work for the McIntosh-Seymour Corporation, Auburn, N.Y., and in 1920 and 1921 was research industrial engineer for the Yale & Towne Mfg. Co., Stamford, Conn. In 1921 he became sales engineer for the H. B. Smith Co., New York, the position which he held at the time of his death. He was also president of the Dissolene Corporation of Hempstead, Long Island, N.Y.

Mr. Faulkner became a junior member of the A.S.M.E. in 1921 and was promoted to the grade of an associate member in 1926. He also belonged to the American Society of Heating and Ventilating Engineers, the Yale Club and Yale Engineering Society, and was vice-president of the Anthracite Coal Club. He married Ruth Anderson in 1925 and is survived by her and two children. His death occurred in Brooklyn on February 8, 1931.

HENRY B. FITZPATRICK

Henry B. Fitzpatrick, who in 1930 became a junior member of the A.S.M.E., was born in Honolulu, Hawaii, on June 18, 1904, the son of C. H. and Alice (Brady) Fitzpatrick. He was educated in the high school at Newberg, Ore., and at the Oregon State College, from which he received a B.S. degree in 1929.

After a short time with the Bend Iron Works, Bend, Ore., he entered the employ of the Byron Jackson Company as a machinist. At the time of his death, on March 23, 1931, he was a salesman for that company.

BRUCE FORD

Bruce Ford, second vice-president and a director of The Electric Storage Battery Company, Philadelphia, Pa., and until recently general manager of the company, died on August 10, 1931, at his home at Chestnut Hill, Philadelphia, after a brief illness following several years of poor health. Mr. Ford was also a director of the Chloride Electrical Storage Company, Ltd., of Manchester, England.

Mr. Ford was born in Brooklyn, N.Y., the son of Edward Lloyd and Lucenia (Young) Ford, and was educated at St. Paul's School, Garden City, L.I., and Brooklyn Polytechnic Institute. Later he took evening courses at The Franklin Institute for a year. He served an apprenticeship and worked in the drafting room of the United Electric Improvement Company and the Heisler Electric Company, of Gloucester, N.J., and was employed for six years by the Johnson Company (subsequently the Lorain Steel Company), of Johnstown, Pa., as draftsman, shop foreman, and engineer. During the last year of his connection with this company he developed and installed a contact railway system in Washington, D.C.

In 1899 Mr. Ford became engineer in charge of development and design in The Electric Storage Battery Company and was promoted to the office of vice-president in charge of technical development in 1913. He had to his credit more than fifty inventions in the storage battery field, the patents being owned by the company.

Mr. Ford was elected a member of the A.S.M.E. in 1907. He also belonged to the Society of Automotive Engineers, the American Institute of Electrical Engineers, and The Franklin Institute, as well as to a number of clubs including the Engineers' of New York and Philadelphia, and the Philadelphia Cricket and Old Colony. He inherited membership in the Military Order of the Loyal Legion of the United States from his father, who was a captain in the Union Army during the Civil War. Mr. Ford was a vestryman of St. Paul's Church, in Chestnut Hill. He was an enthusiastic fisherman and big-game hunter and had secured many trophies during hunting trips in Canada. He is survived by his widow, Sophie (du Pont) Ford, formerly of Wilmington.

G. CLINTON GARDNER, JR.

G. Clinton Gardner, Jr., died suddenly of heart failure at Kane, Pa., on June 9, 1931. He is survived by his widow, Elizabeth (Stirling) Gardner, and by two sons, G. Clinton Gardner, 3d, and John McLean Gardner, of Erie, Pa.

Mr. Gardner was born at Altoona, Pa., on March 5, 1873. His father was general superintendent of the Pennsylvania Railroad, so that the natural thing for the son to do, after attending Hill School, Pottstown, Pa., was to enter the shops of that road. He served a special apprenticeship as a machinist and worked as draftsman and inspector in shops at Altoona, Renovo, West Philadelphia, and Jersey

City. In 1902 he was appointed assistant road foreman of engines on the New York Division of the United Railroad of New Jersey a branch of the Pennsylvania Railroad. He had served as general foreman of the Belvidere Division, Lambertville, N.J., Hudson Division, Jersey City, N.J., and Manhattan Division, also with headquarters at Jersey City. For several years he was stationed at Buffalo, N.Y., as assistant engineer of motive power, and then was promoted to the Monongahela Division at Uniontown, Pa., as master mechanic. At the time of his death he was master mechanic of the Renovo Division, with offices at Erie.

Mr. Gardner had been a member of the A.S.M.E. since 1903.

ADOLPH A. GATHEMAN

Adolph A. Gatheman, whose death occurred at the Naval Hospital, Chelsea, Mass., on June 12, 1931, was born at Oldenburg, Germany, on May 4, 1880. His parents, Ludwig and Amalia (Dinklage) Gatheman, brought him to the United States in 1882 and apprenticed him at the age of fourteen to the machinist's trade in Chicago, where he worked at that trade for some years.

At the outbreak of the Spanish-American War, Mr. Gatheman entered the United States Navy and was soon rated as a first-class machinist. He was commissioned chief machinist in 1901, in charge of engineer's departments of different ships. For about a year he was chief engineer of the U.S.S. *Isle de Cuba* and at other periods was stationed at the public works office of the Boston Navy Yard and was an inspector of coal at Baltimore, Md. In 1917 he was commissioned a lieutenant and as assistant to the engineer's office of the Boston Navy Yard was put in charge of repairs and maintenance of the Boston section of the patrol fleet.

He resigned from the navy in 1919 to accept an appointment as supervisory engineer of the Sea Service Bureau of the U. S. Shipping Board. He was in active charge of training engineers on Shipping Board vessels and established schools at the General Electric Company, Schenectady, N.Y., Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa., and McIntosh & Seymour Corp., Auburn, N.Y., for training engineers on turbine reduction gears, electric drive, and Diesel engines. Subsequently he served as port engineer of the New England district of the Shipping Board, making his headquarters at Boston.

Since his resignation from the Board in 1922 Mr. Gatheman had been connected with the sales work of the A. M. Byers Co., Pittsburgh, acting as district manager of New England.

Mr. Gatheman became a member of the A.S.M.E. in 1923. He also belonged to the Society of Naval Architects and Marine Engineers, New England Water Works Association, New England Gas Association, New England Railroad Club, Engineering Societies of Boston, National Association of Practical Refrigerating Engineers, Engineer Club of Boston, as well as several clubs, the American Legion, and the Masonic fraternity. He is survived by his widow, Mabel S. (Obrion) Gatheman, of Boston.

E. FENTON GILBERT

E. Fenton Gilbert, president of The Dampney Company of America, of Boston, Mass., died at his summer home on Cape Cod on August 29, 1931. He was born in Jamestown, N.Y., on June 11, 1886, the son of Albert and Jeannette (Fenton) Gilbert. He was a graduate of Hill School at Pottstown, Pa., and received his technical education at the Sheffield Scientific School of Yale University.

Mr. Gilbert entered the employ of the Art Metal Construction Company of Jamestown, N.Y., in 1907 and was its secretary-treasurer and general manager when he left the company seven years later. He was next identified for a year with the Miami Motor Cycle Company of Middletown, Ohio, as manager of its motorcycle department. For the next two years, he was New England manager of the Toledo Scale Company, of Toledo, Ohio.

During the World War, Mr. Gilbert served in the United States Naval Reserve Corps, being connected with the Aircraft Division of the Bureau of Ordnance, in charge of the production of aircraft armament. Following the war, he operated an independent sales agency in Boston, representing managers of factory equipment. In 1920 he secured a financial interest in The Dampney Company of America. He had been its treasurer and president since that time. This company manufactures protective coatings for metal surfaces, used extensively in power plants.

Mr. Gilbert was a member of several clubs and societies, including the Yale Club of New York, the University Club of Boston, and the A.S.M.E., which he joined as an associate in 1924. Surviving Mr. Gilbert are his widow, Katharine L. (Gifford) Gilbert, Brookline, Mass., whom he married in 1912, and two children, John A. and May Gilbert.

WILLIAM A. GILCHRIST

William A. Gilchrist, who died on December 18, 1930, was born at Alpena, Mich., on October 17, 1875. He attended the University of Michigan and secured some practical experience in the woodworking industry and in transportation on the Great Lakes. From 1899 to 1902 he engaged in construction and operation for the Saginaw Sugar Company and Michigan Sugar Company, manufacturers of beet sugar. During the next few years he worked on construction and repairs of ocean-going steamers. Then for twenty years he was associated in executive capacities with companies handling the design, construction, and operation of forest products plants. At the time of his death he was president of Gilchrist and Company, of Chicago, Ill.

Mr. Gilchrist became a member of the A.S.M.E. in 1922. He was also chairman of a committee in charge of research work for the National Lumber Manufacturing Association at that time.

EDWARD JOHN GOTTSCHALK

Edward John Gottschalk, whose death from diabetes occurred on June 21, 1931, at the Nassau Hospital, Mineola, N.Y., was born at Rockland, Maine, on July 22, 1898. His high school education was obtained at the De Witt-Clinton High School in New York. He subsequently attended classes at Cooper Union and Pratt Institute, being graduated from the mechanical engineering course of the latter in 1920. From then until 1927 he worked as mechanical draftsman for the American Manufacturing Company, in Brooklyn. He designed textile mills and machinery, made power plant layouts and planned the conversion from steam to electric drive at the Charleston, S.C., plant of the company, redesigned the heating system at the Brooklyn mill, prepared the layout for a Philadelphia mill manufacturing jute carpet yarn and jute twines, and assisted in the construction of a mill at Calcutta, India, for the manufacture of jute bagging for use in cotton mills. During the next year he was connected with the E. Y. Sayer Engineering Corporation in New York. From then until his death he was with the Pan American Petroleum & Transport Co. as mechanical engineer.

Mr. Gottschalk belonged to the United States Naval Air Service, 1917–1918, being stationed at the Rockaway Naval Air Station, Long Island. He became an associate member of the A.S.M.E. in 1926. He also belonged to the Masonic fraternity.

THOMAS WILBRAHAM GREEN

Thomas Wilbraham Green, president and treasurer of the Thos. W. Green Engineering Co., Philadelphia, Pa., died on January 6, 1930. He had been a member of the A.S.M.E. since 1902.

Mr. Green was born at Clay Cross, Derbyshire, England, on July 3, 1853. His technical training was secured in the United States. He served an apprenticeship with Wilbraham Brothers, of Philadelphia, from 1869 to 1873 and worked for that company as designer and patternmaker for many years. The name of the company was changed to the Wilbraham Baker Blower Company and later it became the Wilbraham Green Blower Company and relocated in Pottstown, Pa. Mr. Green advanced to the position of chief engineer of the company and also served as its secretary and treasurer. He designed and built the Green pressure blowers, pumps, exhausters, etc.

In 1914 Mr. Green located in Trenton, N.J., and did consulting work there and in Philadelphia. Subsequently he organized the Thos. W. Green Engineering Co.

JOHN GOTTFRIED GUSTAFSON

John Gottfried Gustafson, chief mechanical engineer in the Door Device and Signal Department of the Otis Elevator Company, New York, N.Y., and the inventor of many devices used on elevator doors, died on March 20, 1931, of heart disease at his home in Teaneck, N.J.

Mr. Gustafson was born in New York on May 10, 1893, the son of J. Aaron and Ida C. Gustafson. At the age of eighteen he entered the employ of the Burdett Rowntree Manufacturing Company, New York, designing electric dumbwaiters, pneumatic doors, operating and elevator interlocking engines, and manually operated door and steel devices for street railway cars. He had two years' shop work for this company and spent some time collecting field data and checking outside construction work. From 1917 to 1922 he was engaged in the same sort of work for the Elevator Supplies Co., Inc., Hoboken, N.J., successor to the Burdett Rowntree Manufacturing Company. He also worked upon signal apparatus for elevators, hydraulic hoisting machinery for theater curtains, and electric

magnetic operating devices for elevator hatchway doors. From 1919 to 1922 he was in charge of the special mechanical equipment division of the engineering department of the company.

From 1922 until he became identified with the Otis Elevator Company in 1930 Mr. Gustafson was in charge of the mechanical engineering department of the Graham & Norton Co., Inc., New York.

Mr. Gustafson became a junior member of the A.S.M.E. in 1919 and an associate member in 1925. He also belonged to the Swedish-Norwegian Society of New York and to the Masonic fraternity.

Surviving Mr. Gustafson are his widow, Mrs. Hjordis (Nelson) Gustafson, and their daughter, Phoebe Justine Gustafson.

WILLIAM JOSEPH HAGMAN

William Joseph Hagman was killed as the result of a fall down an elevator shaft at the Eddystone plant of the Baldwin Southwark Corporation, on April 6, 1931.

Mr. Hagman was born at Philadelphia, Pa., on March 23, 1866, the son of Frederick W. and Mary L. (Miles) Hagman. He attended the Central High School in Philadelphia and Lehigh University, and served an apprenticeship as machinist and draftsman for Bement, Miles & Co., Philadelphia, later known as the Niles-Bement-Pond Company, and remained with them until 1925, becoming assistant chief draftsman in 1894, chief draftsman in 1897, and general manager in 1900. He then accepted a position as consulting engineer for the Southwark Machine Company, South Philadelphia. Since its consolidation in 1930 with the Baldwin Locomotive Works, under the name of the Baldwin Southwark Corporation, he had been located at its plant at Eddystone, Pa. At the time of his death he was also vice-president of the Ridgway Machine Company, Ridgway, Pa.

Mr. Hagman became a member of the A.S.M.E. in 1901. He also belonged to the Academy of Arts, the Union League, the Art Club, and the Engineers Club of Philadelphia. He was keenly interested in paintings and engravings. One daughter, Mrs. Newton B. Misell, New York, survives him.

ARTHUR GOODWIN HALL

Arthur Goodwin Hall, who died on August 22, 1930, at his home in Cincinnati, was born at Newport, Ky., on August 11, 1873. His parents were Thomas J. and Anna Theresa (Neighbors) Hall. Prior to his two years at the University of Michigan, Mr. Hall was in the employ of the Cincinnati Traction Company, which he served for seven years as fireman, oiler, dynamo tender, and engineer. After leaving the University of Michigan in 1898, he worked for a year as chief engineer for the Gerke Brewing Company of Cincinnati. The following year he was a lecturer for the International Correspondence Schools.

In 1900 Mr. Hall went to Columbus to work as chief engineer at the Chittenden Hotel. After two years there he accepted the position of chief electrician for The Hoste Brewing Company of Columbus and subsequently worked as chief engineer for the Smith Agricultural Chemical Company of that city. From 1904 to 1907 he was chief engineer of the Wehrle Company, stove manufacturers of Newark, Ohio. He then spent a year each as manager for the Nelsonville (Ohio) Electric Light Company, chief engineer of The Hocking Valley Railway Company, factory superintendent for F. W. Niebling & Co., manufacturers of ice and refrigerating machinery in Cincinnati and superintendent of The Oliver Schlemmer Heating & Power Co. of that city. In 1912 he was employed by the City of Cincinnati as chief smoke inspector and during the next four years was mechanical director of the City of Cincinnati General Hospital. After a year in consulting work he took the position of combustion engineer with the Procter & Gamble Co., which he retained until 1921. From then until his retirement in 1924 he was chief engineer of the Emery Candle Company, Cincinnati.

Mr. Hall became a member of the A.S.M.E. in 1913 and was also active in the National Association of Power Engineers. He belonged to the Masonic fraternity and to the Elks. He is survived by his widow, Ella (Carson) Hall, and by two married daughters.

STEPHEN STOCKING HALL

Stephen Stocking Hall, former president and, at the time of his death on September 8, 1931, chairman of the Board of Directors of The Pickering Governor Company, Portland, Conn., was born in that city in January 18, 1864. He was the son of Jesse and Clara E. Hall. He obtained his early education at the Seabury Institute and the High School there, and began work for The Pickering Governor Company in 1881. He had been active in the conduct of its affairs until within a few years of his death. He held patents for various attachments for governors, including a device for the transmission of linear motion, and a patent for universal joints.

Mr. Hall had been a member of the A.S.M.E. since 1912. He is survived by his widow, Marie E. (Pascall) Hall.

WILLIS EDWARD HALL

Willis Edward Hall, whose death occurred on July 25, 1931, at Skytop, Pa. was born in Philadelphia, Pa., on July 20, 1860, the son of John Augustus R. and Caroline (Alford) Hall. He was given a B.S. degree by the Towne Scientific School of the University of Pennsylvania in 1881, and an M.E. degree two years later. Following his graduation, he spent a year in the design department of the Baldwin Locomotive Works. He then served an apprenticeship with the Pennsylvania Railroad, at the Altoona shops, and was subsequently made assistant master mechanic there. From 1892 to 1902, he was assistant general manager of the Niles Tool Works at Hamilton, Ohio. During the next three years, he traveled in different parts of the world. He established his own office as consulting engineer in New York in 1895, and engaged in that work until his retirement about 1912.

Mr. Hall invented the automatic control for the return system of steam heating now in practically universal use, and other engineering devices and methods. He was one of the early proponents of the electrification of steam railroads and submitted a paper on this subject before the A.S.M.E. in 1890. He was also one of the first to experiment with direct electric drive for machine tools. He was the author of "Car Lubrication" and of various other papers before the A.S.M.E. and other technical societies.

Mr. Hall became a member of the A.S.M.E. in 1889. He was a life member of the Historical Society of Pennsylvania and of the Red Cross. He also belonged to the China Society and to a number of technical and social clubs. He married Ina B. Fenner, of Hillsboro, Ohio, in 1890.

RUDOLPH L. HANAU

Rudolph L. Hanau, president of the Hanau Engineering Company, Buffalo, N.Y., died on November 1, 1930. He was born at Victoria West, Cape Colony, S.A., on November 3, 1881. He was graduated in 1905 from the Technical Institute at Darmstadt, Germany. Prior to that he had served an apprenticeship with the Fabrik für Brauerei Einrichtungen Gehrke, in Berlin. He also had some early experience as assistant director with Siemens & Halske, Frankfurt, Germany. Following the completion of his technical education, he secured a position as draftsman for the Machinfabrik Hyligenstaedt, Giessen, Germany.

Mr. Hanau came to the United States in 1906 and prior to his connection with the Hanau Engineering Company in 1907, at that time located in New York, he was engaged as draftsman and designer at the offices of the New York Central Railroad. The Hanau Engineering Company, manufacturers of dental instruments, transferred its headquarters from New York to Buffalo in 1918.

Mr. Hanau became a member of the A.S.M.E. in 1926. He had contributed numerous articles to the *American Machinist* and to dental journals. He made outstanding contributions to prosthetic dentistry by instrumentation as well as technique; fundamental are his "Laws of Articulation," "The Articulation Quint," and many other results of his researches which he presented to the dental profession in numerous and voluminous articles published in dental journals. His teachings are included in the curricula of many dental colleges in the United States and abroad.

Surviving Mr. Hanau is his widow, Isabella (Lus) Hanau, whom he married in 1929.

FREDERICK AUGUSTUS HANNAH

Frederick Augustus Hannah, who died of heart trouble on December 5, 1930, in New York, was a native of Massachusetts, having been born at Medford, on January 18, 1873. His parents were William Augustus and Amelia (Adams) Hannah. He attended public schools at Chelsea, Mass., and was graduated from the Massachusetts Institute of Technology with an S.B. degree in 1895. During the following year he was an assistant in the applied mechanics laboratory at M.I.T. He spent the summer of 1896 as fireman for the West End Street Ry. Co., now the Boston Elevated, and as draftsman with the B. F. Sturtevant Co., and inspector of foundation construction for Rand & Taylor and Kendall & Stevens in Boston. He returned to M.I.T. that fall as assistant in mechanical drawing. During the next summer he was employed as draftsman by Miller & Shaw, and by E. F. Woodbury, engineer and patent solicitor, Boston. He became an instructor in mechanical engineering at M.I.T. in

the fall of 1897, and held this position for two years. His vacation work during that period included the design of automatic machinery for the Electrical Machine Company, Boston.

During the winter of 1899–1900, Mr. Hannah was an instructor in mechanical engineering at the Polytechnic Institute of Brooklyn. He helped to equip new shops and the hydraulic laboratory, and taught shop work and machine drawing and design. During the next year he designed and superintended the construction of special machinery for the American Tobacco Company at its New York shops. From 1902 until his death he was engaged in consulting work, making New York his headquarters. He served the Crosby Steam Gage & Valve Co. as superintendent for a time and also was connected with the American Smelting & Refining Co., the American Radiator Company, the Wilson Welder & Metals Co., and the Boston & Worcester Street Ry. as general manager for its receiver. For a time he was an instructor in evening courses in industrial management at M.I.T. The year prior to his death was spent in Russia as industrial management engineer for the U.S.S.R.

Mr. Hannah became a junior member of the A.S.M.E. in 1900 and member in 1904. He belonged to the Sons of American Revolution and to the Engineers' Clubs in Boston and New York, as well as to several other clubs. He is survived by his widow, Anne Ruth (Judkins) Hannah, whom he married in 1927.

MERLE W. HATTON

Merle W. Hatton, whose death occurred on May 24, 1931, was born on November 6, 1888, at Alpena, S.D., the son of Richard and Cora (Thomas) Hatton. He received a B.S. degree in mechanical engineering from the Carnegie Institute of Technology, Pittsburgh, Pa., in 1913. During the next four years he was employed by the Mesta Machine Company, Pittsburgh, at first as assistant master mechanic and later as superintendent of electrical and mechanical maintenance. In 1918 and 1919 he served as mechanical expert in the Research Division of the Chemical Warfare Service, at the American University Experiment Station, Washington, D.C.

Since 1919 Mr. Hatton had been connected with the American Sheet and Tin Plate Company, for six years as master mechanic at the Gary Tin Mill, Gary, Ind., and since then as resident engineer at Canton, Ohio.

Mr. Hatton became an associate-member of the A.S.M.E. in 1918. He is survived by his widow, Mrs. Elizabeth Fiske Kennedy Hatton, whom he married in 1919.

ELMER LELAND HAYWARD

Elmer Leland Hayward, who died of heart trouble on May 2, 1930, in Chicago, Ill., was born on May 26, 1866, at Port Henry, N.Y. His parents were Eugene B. and Ellen E. (Phelps) Hayward. Mr. Hayward was educated in the public schools of Davenport, Iowa, and Pratt Institute, Brooklyn, N.Y., from which he was graduated as a mechanical engineer in 1901. His early experience was obtained with Robert Horton on the upper Mississippi River, with the Eagle Lumber Company at Eagle Mills, Ark., of which he was for a time vice-president, in power plant work, and at the Chicago World's Fair in 1892 and 1893, where he installed fire protection and sewerage systems. He was associated with the Kimball Engineering Company and Cable Lumber Company at Davenport, Iowa, and the Belding-Hayward Company at Chicago. He remodeled several water works plants and had charge of motive power and was president of the Freeo Valley Railroad Company and the Nacogdoches & Southeastern Railroad Co. Later he engaged in consulting work and also taught machine shop practice at Pratt Institute. At the time of his death he was vice-president of the North American Timber Holding Company, Chicago.

Mr. Hayward entered the A.S.M.E. as a junior in 1902 and became a member four years later. He was a member of the Chicago Yacht Club and was interested in making small working models of engines. He married Fannie May Pierce in 1886 and is survived by her and by one daughter, Lulu (Hayward) Calkins, of Los Angeles, Calif.

JAMES NICHOLS HEALD

James Nichols Heald, president and general manager of the Heald Machine Company, Worcester, Mass., died at his home in that city on May 7, 1931, of heart disease. He had been in poor health for several years and although he had not entirely given up active participation in the management of the company, he had relinquished many duties to his sons Roger, vice-president and works manager, and Richard, secretary and treasurer. He is survived by two other sons, Robert and Stanley, and by his widow, formerly Mabel Allen, of Barre, Mass., whom he married in 1894.

Mr. Heald was born in Barre on September 21, 1864, the son of Leander S. and Anna A. (Nichols) Heald. His early education was obtained in Barre and he was graduated in 1884 from the Worcester Polytechnic Institute with the degree of bachelor of science. At that time his grandfather, Stephen Heald, who had established the business in 1826, and his father, were associated in the machine shop and foundry work then known as S. Heald & Son. James became superintendent and designer and when his grandfather died in 1888, took over the management of the business, which was reorganized as L. S. Heald & Son. He saw the place which the grinding machine must have in the growing automobile industry and concentrated on this phase of the work of the company. Drill grinders, center grinders, and miscellaneous grinding attachments were products of those early years.

In 1903 James Heald bought out his father's interest, organized the Heald Machine Company, and, in order to secure better labor supply and transportation facilities, transferred the business to Worcester. He now began the development of a long line of internal grinding machines. The Heald ring and surface grinder was the first machine of the new line, designed for the automobile trade. Then he brought out cylinder grinders and internal grinders for mass production at low cost. A subordinate line was magnetic chucks for holding work. Not only the automobile industry, but also manufacturers of all internal combustion engines were greatly benefited by his achievements in the design and manufacture of grinding machines, and he built up in Worcester an industry employing, in busy times, some eight hundred men.

Mr. Heald was for many years a member of the National Machine Tool Builders' Association and served as its treasurer and for three years on its board of directors. He was made a member of the A.S.M.E. in 1917 and belonged also to the Society of Automotive Engineers. Locally he was a member of the Commonwealth, Country, and Rotary clubs, the Worcester Chamber of Commerce, and the Worcester County Mechanics Association. He was a director of the Worcester Bank & Trust Co. and a trustee of the People's Savings Bank. He attended the Plymouth Congregational Church and was a member of the Board of Assessors of Worcester.

WARREN M. HEALEY

Brigadier-General Warren M. Healey, a Civil War veteran and retired manufacturer of carriages and automobile bodies, died at the Neurological Institute of the Medical Center in New York, on March 25, 1931. He had been in ill health for several months. He had resided at the Union League Club in New York and at his country home at Cornwall-on-the-Hudson since his retirement from business in 1919.

General Healey was born in Norfolk County, Massachusetts, on August 3, 1840, the son of Samuel and Mary Thayer (Mansfield) Healey. He was educated as a mechanical engineer in technical schools in France. From 1865 to 1876 he was general manager for Wood Brothers, Bridgeport, Conn. He established his own business as a coach and carriage manufacturer in New York in 1880 under the name of Healey & Co. With the coming of the automobile he adapted his factory to the production of motor car bodies. He invented a machine for centering and boring hubs of wheels, and for the craftsmanship of his products he was made Chevalier of the Legion of Honor of France in 1889.

General Healey served through the Civil War and was a member of the Grand Army of the Republic. He was one of the original commission for the construction of the Grant Monument in New York and was a trustee of the Grant Monument Association. Following the war he became a member of the New York National Guard, attaining the rank of brigadier-general in 1897. He was vice-president of the Union Dime Bank in 1881 and 1882 and a director of the Hudson Trust Company from 1884 to 1919.

General Healey became a life member of the A.S.M.E. in 1925, and also belonged to the Sons of the American Revolution, the Mayflower Descendants, the New England Society, the New York Historical Society, the New York Chamber of Commerce, American Museum of Natural History, and a number of clubs, including the Metropolitan and Army and Navy.

CHARLES LOUIS HEISLER

Charles Louis Heisler, inventor of the Heisler geared locomotive, and many other items, died at his summer home at Rock City Falls, near Schenectady, N.Y., on October 13, 1931.

Mr. Heisler was born on February 22, 1863, at Wapakoneta, Ohio, the son of Christian and Christina (Lang) Heisler. He was educated in the public schools of that town and at the Cascadilla Preparatory School at Ithaca, N.Y. He was graduated from Cornell University

in 1890 with the degree of mechanical engineer. He was a member of the honorary society of Sigma Xi.

Mr. Heisler had special training in carriage smithing, machine work, and patternmaking as a boy in several shops at Wapakoneta, Ohio, and at St. Marys, Ohio. He withdrew from college for a year to work for the Brooks Locomotive Works, at Dunkirk, N.Y., as engineer of tests in charge of the training school for apprentices. After completing his college education, he returned to this company for two years, performing development work for the president of the company.

During this period, the geared locomotive was developed by Mr. Heisler. His conception of the simple locomotive gear drive with bevel wheel and pinion is said to have anticipated the live-axle and cardan shaft principle of the modern motor car. The Baldwin Locomotive Works took over the manufacture of the locomotive until a manufacturing company was capitalized by business men of Erie, Pa., to whom Mr. Heisler disposed of his interests in the invention.

After the death of the president of the Dunkirk Engineering Company, Mr. Heisler accepted a position with Bement, Miles & Co. working on large machine tools in the engineering department. He developed a high-duty reciprocating pumping engine which was manufactured in a plant at St. Marys, Ohio, known as the Heisler Pumping Engine Company. Mr. Heisler operated this plant for some years, as vice-president and chief engineer, until in 1907 the plant was discontinued.

Mr. Heisler then became connected with the American Locomotive Works, into which the Brooks Locomotive Works and other companies had been merged, as a member of its staff of mechanical engineers. While with that company he developed and invented many improvements in locomotive and machine tools.

At the beginning of the World War, Mr. Heisler was located at Providence, R.I., in the plant of the Alco Manufacturing Company, which was controlled by the American Locomotive Company. He developed devices for manufacturing shells and other ammunition, and after a time became associated with the Washington Steel and Ordnance plant, manufacturers of large armor-piercing and high-explosive shells.

His career with the General Electric Company began in 1922. From that year until 1926 he was in charge of the mechanical engineering department of the general superintendent's office. He engaged in the development of sound recorders and combined sound-reproducing and moving-picture projecting machines in the photophone department, mechanical engineering section. At the time of his death he was in the general engineering department, engaged in the development of refrigerating machines and reversible airplane propeller mechanisms. Other results of his ingenuity were pointed out in a statement at the time of his death by Charles E. Eveleth, vice-president in charge of designing, who said, in part, "While with the General Electric Company he made valuable contributions to the design of geared turbines, hundreds of jigs and fixtures to assist in manufacture; he designed wire enameling machines, paper and asbestos covering machines for wire, as well as dozens of improvements in the design of standard apparatus contributed from the manufacturing standpoint to improve quality and reduce costs." Since joining the staff of the General Electric Company he had been granted 21 patents.

His principal inventions were the Heisler geared locomotive engine, various types of pumping engines and machinery, steam road rollers, water-works pumping machinery, barometric condensers, wet type vacuum pumps, combined sound reproducing and moving-picture projecting machines, refrigerating machinery, and various other types of machinery.

Mr. Heisler became a member of the A.S.M.E. in 1895. He also belonged to the American Association for the Advancement of Science and was a 32nd degree Mason.

He is survived by his widow, Mrs. Laura (Barton) Heisler, of Dunkirk, N. Y., whom he married in 1891, by three sons and a daughter, and by one brother.

WILLIAM FERRIS HENDRY

William Ferris Hendry died at Ossining, N.Y., on January 8, 1930. He was born at Wellsboro, Pa., on August 17, 1873, the son of William Montgomery and Henrietta (Ferris) Hendry. His education included three years at Betts Academy, Stamford, Conn., two years at Cornell University in special engineering work, a student course with the General Electric Works, and an apprenticeship with the Baldwin Locomotive Works and Edison Electric Company. He engaged in general construction work and had charge of the power plant at the New York Navy Yard from 1895 to 1898. The following year he was connected with the Sprague Electric Company as an erecting engineer. He then entered the employ of the Western Electric Company as assistant to the factory engineer and remained with this company for some years, serving successively as factory engineer, assistant superintendent, and superintendent, in New York, and then as technical superintendent at the Hawthorne Works, Chicago. At the time of his death he was vice-president and chief engineer of the Manhattan Electrical Supply Company, New York, and also vice-president of the Troy Laundry Machinery Company, Trout Mining Company, and the United States Manganese Company.

Mr. Hendry designed and patented several motor controlling devices for machine tools and automatic-return sand-blast machines, automatic valves, radio tubes, dry-cell batteries, and automatic package machinery. He became a member of the A.S.M.E. in 1907 and also belonged to the American Institute of Electrical Engineers and the American Chemical Society. He was interested in archeology as well as in engineering. He is survived by his widow, Bess (Sheldon) Hendry.

LOUIS GEORGE HENES

Louis George Henes, who died on November 1, 1931, in New York, N.Y., was born in that city on May 25, 1881, the son of Edwin and Caroline (Schiffer) Henes. He secured his preparatory education at the German-American School there and at the City College of New York. He was graduated from Columbia University with an M.E. degree in 1903. His early experience in engineering was secured during his vacations with the A. J. Lindeman & Hoverson Co. in Milwaukee, in the toolroom of the De LaVergne Refrigerating Machine Company, New York, and in the mechanical laboratory at Columbia University. He served an apprenticeship with the Niles-Bement-Pond Company and from 1904 to 1906 was assistant to the manager of the New York Office of the company. He then became manager of the Machinery Department of Harron Rickard & McCone in San Francisco.

Mr. Henes went into business for himself in San Francisco and Los Angeles in 1909 under the name of the L. G. Henes Machinery Company, which was reorganized and incorporated in Los Angeles in 1931 as the Henes-Morgan Machinery Company, Ltd. He was also Pacific Coast manager for Manning, Maxwell & Moore, Inc., and the Commercial Acetylene Company, both of New York, and for the Railway Materials Company of Chicago.

Mr. Henes joined the A.S.M.E. as a junior member in 1903 and became an associate of the Society in 1910.

Surviving Mr. Henes are a son and daughter. He married Evelyn Ellis, of Oakland, Calif., in 1914.

HILARY J. HENNESSY

Hilary J. Hennessy, chief mechanical draftsman for the Northern Pacific Railway Company, St. Paul, Minn., died suddenly of heart trouble on July 12, 1931. Mr. Hennessy became a member of the A.S.M.E. in 1929. He was born at Chicago, Ill., on October 25, 1889. His technical education was secured through the International Correspondence Schools and the Armour and Lewis Institutes of Chicago. From 1907 to 1911, he engaged in drafting for the National Dump Car Company and the Rodger Ballast Car Company in Chicago. During the next two years, he installed and operated laundry power plant machinery for the Lutz Laundry Machine Company and Sheridan Park Laundry of that city. He had been with the Northern Pacific Railway Company since 1913. His father, J. H. Hennessy, of Chicago, and a daughter, Lorraine Jean Hennessy, survive him. Mr. Hennessy belonged to the Masonic fraternity.

GEORGE HILLYER, JR.

George Hillyer, Jr., whose death occurred in Washington, D.C., on April 24, 1931, was born in Atlanta, Ga., on November 6, 1872, the son of Judge George Hillyer and Ellen E. (Cooley) Hillyer. He received an A.B. degree from the University of Georgia, Athens, Ga., in 1893, and an M.E. from Cornell in 1896.

For a year following the completion of his college work Mr. Hillyer was employed by the Atlanta Talc & Soapstone Mining & Manufacturing Co., Murphy, N.C., as mechanical engineer. He then served a two-year special apprenticeship in the drawing room and shops of the Southern Railway at Atlanta, and worked as draftsman for that road at Washington for several years.

In 1901 Mr. Hillyer went to Detroit as draftsman for the American Car & Foundry Co. The following year he secured a position on the operating board of the American Radiator Company in Chicago. Later he was promoted to the engineering department of the company, and in 1905 became its resident salesman in Atlanta. From 1907 to 1910 he was manager of the Atlanta office of the American

Radiator Company. During the next five years he engaged in consulting engineering in Atlanta.

Since 1916 Mr. Hillyer had been with the Southern Railway, for several years as mechanical engineer in its valuation department in Atlanta, and during the remainder of his life as mechanical engineer of the Southern Railway System, with headquarters at Washington, D.C.

Mr. Hillyer joined the A.S.M.E. as a junior member in 1898, and in 1904 became an Associate of the Society. He was a member and former trustee of the Cornell Alumni Association, a member of the Sons of the American Revolution, the Founders and Patriots of America, National Geographic Society, the Virginia and Georgia Historical Societies, the Chi Phi fraternity, and the American Association for the Advancement of Science, as well as several clubs. He was a Thirty-Second Degree Mason, a Knight Templar, and a Shriner. He had been a member of the Second Baptist Church in Atlanta for 44 years.

Mr. Hillyer is survived by his widow, Mrs. Edith (Carter) Hillyer.

HERMAN EDMUND HOFFSTADT

Herman Edmund Hoffstadt, until recently mechanical engineer for the Mayflower Hotel, Washington, D.C., died on February 18, 1931, at his home in New York, N.Y. He is survived by his widow, Marie A. (Spengler) Hoffstadt, whom he married in 1925, and by one son, Herman Edmund Hoffstadt, Jr.

Mr. Hoffstadt was born on October 2, 1884, in New York, and after completing his high-school education served an apprenticeship in the Kingsbridge and Interborough power plants of the New York Central Railroad as an oiler and dynamo tender. He was then engaged in construction work and the installation of ice machines and gas and oil engines for the De La Vergne Machine Company.

In 1905 Mr. Hoffstadt began work as an oiler at the Municipal Ferry, New York. He obtained a marine license four years later and in 1911 his stationary engineer's license. He served the city until 1917 when he enlisted in the U. S. Navy as a warrant machinist for the reconstruction of vessels for sea duty. Following his release from the Navy after the close of the War he entered Teachers College, Columbia University, under Government tuition to study mechanical drawing, machine design, etc. He completed his course in 1923 and during the next two years served as safety engineer and inspector on steam engines and boilers throughout the United States.

Since 1925 Mr. Hoffstadt had been employed by private concerns. For about six months he was chief engineer and building superintendent for the Sam & Lee Shubert Theater Corporation, in charge of the Broadhurst and Plymouth theaters in New York. He then became assistant superintendent of highways and construction for the John H. Sullivan Construction Company, laying water mains and building conduits and roads. Prior to his connection with the Mayflower he was superintendent and chief engineer of the Wall Street Exchange Building, New York.

Mr. Hoffstadt became an associate member of the A.S.M.E. in 1923 and was promoted to full membership in 1927.

LEO HOLDREDGE

Leo Holdredge, who died suddenly at Knoxville, Tenn., on April 12, 1931, was born at Kent, near Akron, Ohio, on July 2, 1886, the son of S. T. and Ida May (Barrick) Holdredge. In his early youth his parents moved to Louden County, Tennessee, where he attended the public schools. He continued his studies at Tennessee Wesleyan College, formerly known as Grant University, at Athens, and then served for a number of years as principal of a Methodist Seminary at Epworth, Ga. This period was followed by a business affiliation at Lenoir City, where he was senior member of the firm of Holdredge and Kollock, hardware dealers.

In 1921, Mr. Holdredge decided to continue his education and entered the Engineering College of the University of Tennessee. During his junior year he was selected as the outstanding engineering student and was awarded the scholarship prize offered by that institution. He was graduated with a B.S. degree in mechanical engineering in 1926 and two years later received his M.E. degree.

During the summer of 1926 he engaged in time study and efficiency tests for The Fulton Company, at Knoxville, Tenn. He returned to the University as instructor in mechanical engineering in September, and in 1930 was advanced to the rank of assistant professor of mechanical engineering. During this period he was the author and co-author of a number of Engineering Experiment Station research bulletins, especially on subjects pertaining to fuel and heating and ventilation, and was at the time of his death assisting in preparing a text on heating and ventilation.

Professor Holdredge became an associate member of the A.S.M.E. in 1926 and was serving the Knoxville Section as secretary at the time of his death. He had just completed a technical research paper in cooperation with Prof. E. L. Carpenter at the University of Tennessee on the "Cotton Oil Industry of the South" for presentation at the Semi-Annual Meeting of the Society at Birmingham, Ala. He was also a member of Phi Kappa Phi and Tau Beta Pi, honorary fraternities, and of the Knoxville Technical Society.

He joined the Methodist Episcopal Church at a very early age and was active throughout his life in the major activities of this denomination, and particularly in the Epworth League and other work with young people.

He married Miss Leta Cardwell of Lenoir City, Tenn., in 1914, and is survived by her and by three sons and one daughter.

EDGAR MALIN HOOPES, JR.

Edgar Malin Hoopes, Jr., member of the staff of the Governor of Delaware, and engineer for the E. I. du Pont de Nemours Co., at Wilmington, Del., died on November 20, 1931. Colonel Hoopes was born on May 18, 1888, at Wilmington, Del., the son of Edgar Malin and Elizabeth (Osborne) Hoopes. He received a Ph.B. degree from the Sheffield Scientific School of Yale University in 1910. He had had some experience during summer vacations in testing materials and in surveying for water supply and sewage systems in Charleston and Romney, W. Va., and in Wilmington. Following his graduation, he was employed by Joseph Bancroft & Sons Co., at Wilmington and Reading, Pa., on the design of power plant additions, special bleaching and dyeing machinery, and as engineer on steam economy tests. In 1911, he became connected with the Board of Water Commissioners of Wilmington. He worked on the design of proposed water-power developments and hydraulic-turbine installations. He was appointed assistant engineer before the end of that year, and in 1915 chief engineer in charge of operation and construction work for the water department, engineering and survey department, and street and sewer department of the City of Wilmington.

In 1918, Mr. Hoopes received a Captain's commission and served at Camp Meade and Fortress Monroe. After the war, he became a member of the engineering staff of the duPont Company and later was engineer for the Elafrael Corporation, composed of the family of Senator du Pont. He was commissioned a colonel on the staff of Governor Buck in 1929. He also served on the Board of Water Commissioners from 1921 until his death, and was a director of the Wilmington Savings Fund, the Wilmington Library, the Fine Arts Society, and the Regional Planning Commission.

Mr. Hoopes became an associate member of the A.S.M.E. in 1917. He married Helen Nelson Bain, at Newburgh, N.Y., in 1913.

CHARLES HENRY HORN

Charles Henry Horn, of Howard Beach, Long Island, N.Y., died on July 25, 1931. Mr. Horn was born at Astoria, Long Island, on March 25, 1882, the son of David and Anna (Beck) Horn. His first position was as toolroom foreman with the John Thomson Press Company in Long Island City, for whom he later became assistant superintendent of tools. After three years with this company, he took a similar position with the Automatic Products Corporation in Long Island City. From 1922 until 1927, when the Tool Room and Production Departments were in his charge, he worked directly with Mr. John Thomson on the perfection of the Thomson Adriance Meter, and also did experimental work on a new feeder. Subsequently, he turned his attention to the design and supervision of the manufacture of drill jigs, dies, etc.

Mr. Horn became an associate member of the A.S.M.E. in 1930, and also belonged to the Masonic fraternity. He is survived by his widow, Minnie (Naumann) Horn, whom he married in 1914, and by three children, Charles W., Howard H., and Anita D. Horn.

THOMAS ROGERS HUGHES

Thomas Rogers Hughes, who died at St. Joseph, Mo., on August 2, 1931, was born at Oxford, Ohio, in 1902. He received an A.B. degree from the Miami University in 1924 and a B.S. degree from the Massachusetts Institute of Technology two years later. He then entered the employ of the American Waterworks & Electric Co., in New York. His duties included the selection and layout of mechanical equipment of waterworks and pumping stations. He was connected with this company at the time of his death.

Mr. Hughes became a junior member of the A.S.M.E. in 1929 and also belonged to the American Society of Civil Engineers.

JOHN W. HUNNS

John W. Hunns, assistant at the Physical Laboratory of The Russell Manufacturing Company, Middletown, Conn., died on February 25, 1931. Mr. Hunns became a junior member of the A.S.M.E. in 1930, and was a member of the Third Congregational Church in Middletown.

The son of George and Mary M. (Williams) Hunns, he was born on June 7, 1907, at Middletown and attended the high school there. In 1926 and 1927 he worked as an inspector on clutch facings and brake linings for the Russell company, and during his vacations while attending Pratt Institute he was also employed by the company, part of the time on inspection and part on production work. He returned to Middletown as assistant at the Physical Laboratory following the completion of his course at Pratt in June, 1930.

Mr. Hunns is survived by his father and mother and by a brother.

HARRY BURLEIGH HUNT

Harry Burleigh Hunt was born in New York, N.Y., on March 8, 1875, the son of James Baldwin and Estelle M. (Hinckley) Hunt. His early education was obtained in the schools of Brooklyn and he was graduated as a mechanical engineer from the Massachusetts Institute of Technology in 1897. He entered the employ of the Erie Railroad at Susquehanna as a special apprentice and remained with that road as draftsman, engineer of tests, and assistant mechanical superintendent for eight years. He resigned in 1905 to accept a position with the American Locomotive Company. During the twelve years he was associated with this company he was a member of its Engineering Committee. He was manager of the Electric Locomotive Department in 1906 and 1907, and for the next two years was employed jointly by the American Locomotive Company and the General Electric Company to investigate the electrification of steam railroads. He was made assistant to the president, Mr. Waldo H. Marshall, in 1914, and placed in charge of sales and the production of artillery ammunition; he was appointed vice-president in February, 1917. He placed the manufacture of munitions on a workmanlike and profitable basis.

In June, 1917, Mr. Hunt was granted leave of absence and entered the U. S. Army as a Major in the Ordnance Reserve Corps. He was assigned to the Gun Division in charge of production of cannon, where his technical knowledge, coupled with his administrative ability and understanding of sound finance, were of inestimable value to the Government. He was promoted to the rank of Lieutenant-Colonel in January, 1918, and in November of that year to Colonel in the Ordnance Department, assigned to the Artillery Ammunition Division. He was discharged from service in January, 1919.

During 1920 he was associated with George W. Goethals & Co., Inc., as vice-president and general manager of the Automatic Straight Air-Brake Company. Ill health caused by intensive work during the war necessitated his resignation at the end of 1920, and since that time he had not been active in business, although he had made some studies of manufacturing for industrial companies.

Mr. Hunt became a member of the A.S.M.E. in 1916 and also belonged to the New York Railroad Club, the Engineers' Club of New York, and the University Club of Chicago. He died in New Rochelle, N.Y., on July 25, 1929, of heart failure. His estate was left in trust for a surviving sister and a friend, with the provision that eventually it should go to the Massachusetts Institute of Technology.

WILLIAM SPENCER HUNTINGTON

William Spencer Huntington was born at Fairbanks, Iowa, on June 23, 1867, the son of William and Maria (Spencer) Huntington. His early edu ation was obtained at Oran and at Oelwein, and he taught in that vicinity before taking up the study of engineering. He was given a B.S. degree in civil engineering by the University of Iowa in 1895. His first position was at Oelwein, where he engaged in water-works construction during the year following his graduation, and was the first superintendent of the works. The next year was spent as draftsman on steel bridges for the Bellefontaine Bridge & Iron Co., Bellefontaine, Ohio. He did similar work for the American Bridge Company, Minneapolis, Minn., from 1897 to 1900. The next eight years were spent with the St. Paul Foundry Co., St. Paul, Minn., first as chief draftsman and then as chief engineer. His duties included the design of structural steel work and the rebuilding of the company's plant.

In 1908 Mr. Huntington went to Kansas City, Mo., to accept the position of chief engineer for the Kansas City Structural Steel Company. He resigned this position in 1913 to engage in general engineering work at Galveston, Texas. From 1915 until his retirement in 1928 he was identified with the C. L. Brooks Engineering Co. and The Moultrie Construction Company of Moultrie, Ga., for which he designed packing houses and cold storage plants in this country and in Cuba. During the World War he was engaged in work on explosives for the United States Government in New York and at Nitro, W.Va.

He had traveled extensively through the United States since his retirement, and was on the way from Oelwein to Florida when his death occurred, at Streator, Ill., on December 5, 1931.

Mr. Huntington became an associate member of the A.S.M.E. in 1917. He had written books on refrigeration and contributed articles on that subject to the technical press. He is survived by his widow, Sarah M. A. (Smith) Huntington, whom he married in 1899.

OREN S. HUSSEY

Oren S. Hussey, retired, resident of Nashua, N.H., died at Bradenton, Fla., on February 8, 1931. Mr. Hussey was born in Nashua on May 16, 1865, and attended the public schools there. He then entered the Massachusetts Institute of Technology, from which he was graduated in 1887 with an S.B. degree. Following his graduation Mr. Hussey worked at the laboratory of Thomas A. Edison in Orange, N.J., engaged primarily in experimental work, until 1890. During this engagement, he designed an air separating plant which was erected at Humboldt, Mich., and a similar installation at Bechtelsville, Pa. From 1890 to 1892 he was mechanical engineer for the Thomson, Houston Motor Company, Boston, Mass., having charge of all engineering work of the company, installation of power plants, etc. He then engaged in business for himself in Boston for two years, in the firm of Bailey & Hussey, wholesale building supply company. In 1894 he became a member of the firm of Gregg & Son, door and sash manufacturers, of Nashua. He retired from this firm in 1904 to become treasurer of the Frank Ridlon Company, Boston, a position he held until 1915, when he retired because of ill health.

During the World War Mr. Hussey held a civilian post in the Ordnance Division of the War Department at Washington, D.C.

Mr. Hussey became a member of the A.S.M.E. in 1897. He is survived by his widow, Mrs. Margie L. Hussey, whom he married in 1889.

FRANKLIN KILSHAW IRWIN

Franklin Kilshaw Irwin died on April 6, 1931, at Ardmore, Pa., where he had resided since his retirement from active business in 1929. Mr. Irwin was the son of Thomas Kilshaw and Mary (Ketchum) Irwin. He was born at Mobile, Ala., on December 4, 1859. He attended Towle's Collegiate Institute there and was graduated from the Stevens Institute of Technology as a mechanical engineer in 1883. His first position was as draftsman with the American Ship Building Company in Philadelphia. From 1884 to 1900 he was connected as machinist apprentice, draftsman, and mechanical engineer with the Wisconsin Central Railroad and continued his studies through the International Correspondence Schools. During the next three years he was assistant engineer in charge of design and construction of repair shops and equipment for the Union Pacific Railroad, Omaha, Neb. After a short time in 1903 with the Galena Signal Oil Company of Franklin, Pa., he accepted a position as engineer of construction with the New York, New Haven & Hartford R.R., being appointed superintendent of bridges and buildings for that road in 1908. From 1911 to 1914 he was special engineer and superintendent of the Boston and Maine Railroad.

Mr. Irwin took up sales work in 1915 as agent for the Babcock & Wilcox Co. of Boston. He was transferred to the Chicago sales force the following year and in 1917 became sales manager of the Philadelphia Niles-Bement-Pond Company. From 1924 until 1929 he was sales engineer for the Combustion Engineering Corporation of Philadelphia.

Mr. Irwin became a member of the A.S.M.E. in 1913. He belonged to the Pennsylvania Historical Society, the Manufacturer's Club of Philadelphia, and the Masonic fraternity. Surviving him is his widow, Mary (McHenry) Irwin of Haverford, Pa., whom he married in 1890, and one son, Kilshaw McHenry Irwin.

CARL ALBERT JOHNSON

Carl Albert Johnson, president of the Gisholt Machine Company, Madison, Wis., died on October 30, 1931. Mr. Johnson was born at Madison on April 21, 1870, the son of John A. and Kaia (Kildahl) Johnson. He received a degree in mechanical engineering from the University of Wisconsin in 1891. During his vacations he had worked in the machine shop and drafting room of the Gisholt Machine Company, and following his graduation he continued in the business of this company. He devoted a great deal of his time to the tools and

appliances used with their machines and traveled both in the United States and in Europe in the interests of the company. He was appointed general manager in 1895 and president in 1908.

Mr. Johnson became a member of the A.S.M.E. in 1901. He had served as president of the Wisconsin Manufacturers' Association for seven years and was a past-president of the National Machine Tool Builders' Association. He was a director of the U.S. Chamber of Commerce, founder and first president of the Madison Association of Commerce, and a director of the First National Bank in Chicago and of the Cutler-Hammer Manufacturing Company of Milwaukee.

Surviving Mr. Johnson are his widow, Mrs. Lucy (McGlachin Berry) Johnson, whom he married in 1927, and Mrs. Ruth (Johnson) Weston, a daughter by a former marriage.

JOHN SAMUEL ADOLPHUS JOHNSON

John Samuel Adolphus Johnson was born at Cismont, Va., on August 10, 1878, the son of Wm. Whitfield and Katherine M. (Anderson) Johnson. He attended public and private schools in that vicinity and was given a B.S. degree in mechanical engineering by the Virginia Polytechnic Institute, Blacksburg, Va., in 1898, and an M.E. degree the following year. He also studied at Cornell University in the summer of 1902 and at Lehigh University in the summer of 1908. Another summer was spent at the Baldwin Locomotive Works to secure practical experience. The remainder of his life was spent in teaching at the Virginia Polytechnic Institute. He was made assistant professor of mechanical engineering in 1900, after having been an instructor in that subject and in military science and tactics since his graduation. In 1906 he was given the chair of experimental engineering. Since 1914, he had been professor of applied mechanics and experimental engineering, and since 1921 had been director of the Virginia Engineering Experimenting Station.

Professor Johnson became a member of the A.S.M.E. in 1911. He also belonged to the American Society for Testing Materials, and the Society for the Promotion of Engineering Education. He had contributed a number of articles to the technical press.

Surviving Professor Johnson are his widow, Margaret M. A. (Smyth) Johnson, whom he married in 1905, three daughters, and a son. Professor Johnson, who had been in ill health and had been advised to give up his work for a time, committed suicide on October 21, 1931.

JOSEPH EDWARD JOHNSON

Joseph Edward Johnson died of heart failure on May 16, 1930, at his home in Meriden, Conn. He was a native of Sweden, having been born at Elssborg on September 22, 1865, the son of Johan Persson and Bendikta (Carlson) Johnson. He attended the public schools of Elssborg, served an apprenticeship at the Halmstead's Mechanical Works, and gained experience as a toolmaker at plants of other Swedish firms. He came to the United States in 1889 and worked at his trade with different firms here, among which was the New Haven Clock Company, for which he designed special subpress dies for blanking finished clock wheels and other parts from sheet metal in one operation. From 1897 to 1905 he was connected with The M. B. Schenck Co., now the Bassick Company, of Meriden, Conn., designing all machine tools and automatic machines used in the manufacture of furniture and truck casters. He was in charge of the tool and machine department of the company and redesigned and improved the Schenck hydraulic water-wheel governors, manufacturing, selling, and installing them in some of the largest power plants in the eastern part of the country. In 1905 Mr. Johnson went to Rochester, N.Y., as factory superintendent for the Pritchard-Strong Mfg. Co., manufacturers of lanterns and household utensils. He installed new automatic stokers and high-speed steam engines, as well as a complete lighting and heating system. He returned to the M. B. Schenck Co. in 1907 and remained with them for two years in work similar to that which he had previously done for the company. He then entered the employ of the Connecticut Telephone & Electric Co. of Meriden, as superintendent of the mechanical department, as well as of the different buildings of the company. When its molded insulation department was taken over by the Monowatt Company, he entered the employ of that firm as chief engineer.

Mr. Johnson had taken an active interest in civic affairs in Meriden, having been connected with the city government for more than twelve years. He was police commissioner under the former Mayor Daniel J. Donovan, and had served as alderman and mayor pro tem. He was chairman of the aviation commission of Meriden. He was a member and former president of the John Ericsson Republican Club of that city and treasurer of the state league of John Ericsson Republican Clubs.

Mr. Johnson became a member of the A.S.M.E. in 1916. He also belonged to the Odd Fellows and to several other fraternal societies. In 1898 he married Anna Louise Nelson, who, together with three daughters, survive him. Mr. Johnson held patents on a take-up device for crank-pin connections, speed regulator for hydraulic water-wheel governors, casters, and coffee percolators.

LEWIS ERNEST KEIL

Lewis Ernest Keil died on February 26, 1931, at the Walter Reed Hospital, Washington, D.C., where he had been confined since the middle of October, 1930. He was buried with full military honors in the Arlington National Cemetery. Mr. Keil served in the Ordnance Department from 1917 to 1919, being stationed at Camp Funston, Kansas, and using the shops of the Kansas State Agricultural College for the manufacture of precision clinometers of his own design for use in laying machine guns for the 89th Division.

Mr. Keil was born at Chicago, Ill., on May 15, 1887, the son of Louis E. Keil and Lorena C. Keil. His technical training was secured at the Chicago Manual Training School, which he attended for three years, and as a special apprentice in the locomotive shops of the Chicago Division of the Illinois Central Railway from 1904 to 1906. He received an E.E. degree from George Washington University in 1929 and an M.E. degree the following year.

During 1907 Mr. Keil was a draftsman and templetmaker for the Pullman Car Company, at Chicago, and from 1907 to 1908, draftsman and estimator for the Hicks Locomotive Company, Chicago Heights. As draftsman for four years with the Northern Pacific Railway Company at St. Paul, Minn., Mr. Keil worked on shop layouts and electrical wiring diagrams, as well as locomotive parts, cars, and trucks. In 1913 he gained experience in the design of refrigerator cars and trucks for the Moore Patent Refrigerator Car Company, St. Paul, building and equipping cars and testing them for a number of roads.

The following year he went to St. Louis, Mo., as chief draftsman for the St. Louis Car Company, in 1915 was appointed chief engineer, and the next year took on the additional duties of superintendent of the automobile shops. He was with this company until he entered the Army in 1917. Under his supervision some three hundred electric cars and trucks were built for the New York Railway Company, and as many more for the Manhattan and Queens Electric Railway, the Kansas City Railway Company, and the Pittsburgh Railway Company. He also directed the construction of a fleet of 150 all-steel motor buses for the Chicago Motorbus Corporation, and a variety of high-speed electric interurban equipment and trucks, including an express train equipment for the Grand Rapids, Kalamazoo High Tension Line of the Michigan Railway Company designed to operate on 600–1200 and 2400 volts d.c. As superintendent of the automobile shops he had charge of the machining of shrapnel cases for holding the fuse, bullets, and explosive, and designed the tools and jigs used for this work.

After the close of the War Mr. Keil became chief draftsman for the Texas & Pacific R.R. at Marshall, Texas, where he remained until May, 1920. During the remainder of that year he was connected with the Havana Central Railroad in Cuba. From then until 1925 he was mechanical engineer in the Bureau of Locomotive Inspection of the Interstate Commerce Commission at Washington, D.C. From 1925 to April, 1927, he was engaged as mechanical engineer by the Railway Locomotor Company, Chicago, and since July 1, 1927, had been assistant mechanical engineer at the Bureau of Yards and Docks of the U.S. Navy Department, Washington, D.C.

Mr. Keil was the patentor of electric motor car truck brakes used by the Pittsburgh and other electric railway companies and of a special design of electric railway trucks used under small electric railway cars built to meet bus competition. The St. Louis Car Company acquired rights to both these patents.

Mr. Keil had been an associate member of the A.S.M.E. since 1919.

Surviving him are his widow, Venita (Worley) Keil, his father, and one son, Lawrence E. Keil, of Chicago, Ill.

FRANK BOCKIUS KING

Frank Bockius King, retired naval architect and marine engineer, died at his home in Washington, D.C., on January 26, 1931, after an illness of three months.

Mr. King was born in Germantown, Philadelphia, Pa., on March 28, 1855, the son of Charles M. and Anna Louisa King. He attended the public schools of Philadelphia, and was graduated from the old Polytechnic College of Pennsylvania there with an M.E. degree in 1875. After some early training in the machine shop of John Roach & Sons, Philadelphia, he entered the marine field as an employee of Wm. Cramp & Sons, of that city. Another early contact was with the Pusey & Jones Co., in Wilmington, Del. In 1884 he was ship

draftsman at the Navy Yard at Washington and subsequently was associated with the chief constructor of the Bureau of Construction and Repair of the United States Navy in a study of European naval construction.

In December, 1886, Mr. King became superintendent of the Columbian Iron Works at Baltimore, Md., when the contract was secured for the construction of the gunboat *Petrel* for the U.S. Navy.

Mr. King was appointed marine superintendent of the Maryland Steel Company at Sparrows Point, Md., early in 1890, and a number of harbor and sea-going vessels were constructed under his supervision during the next four years.

In 1895 Mr. King opened an office in Baltimore for the practice of consulting marine engineering and naval architecture. He relocated in Washington the following year, and maintained his office there until his retirement.

Mr. King was twice married. In 1893 he married Helen Waldie Clark, of Philadelphia, who died in 1916. In 1918 he married Laura Larzelere Hofmann, also of Philadelphia, by whom he is survived. He was a member of the Society of Naval Architects and Marine Engineers, the American Society of Naval Architects, the Institution of Naval Architects, and the Washington Society of Engineers. He became a member of the A.S.M.E. in 1892.

HERMAN ARNOLD KIRBERG

Herman Arnold Kirberg, works manager of the Hanson-Van Winkle-Manning Company, Newark, N.J., died on November 25, 1931. Mr. Kirberg was born in New York, N.Y., on August 7, 1884, the son of Louise Veron and Herman Kirberg. After two years at Columbia University he entered the employ of the Western Electric Company, New York, where he was engaged for three years in making estimates and cost analyses. During the next four years he was employed by the Manhattan Electrical Supply Company at its Jersey City plant. He conducted research work on stock control and improved manufacturing methods. For a time he was in charge of the purchasing department and later general assistant to the works manager. From 1912 to 1917 Mr. Kirberg was connected with the Westinghouse Electric & Manufacturing Co. His duties included time studies, rate setting, and the improvement of manufacturing methods. For four years (1917–1921) Mr. Kirberg was assistant to the works manager of the Edison Phone Works in West Orange, N.J. He was production engineer on the Bates numbering machine and assisted in improving the existing designs and developing new ones. Since 1922 Mr. Kirberg had engaged in industrial engineering, largely in Newark and vicinity. For a time he was connected with Roth Brothers of Chicago, going from there to Newark to become works manager for its associated organization, the Hanson-Van Winkle-Manning Company.

Mr. Kirberg became an associate member of the A.S.M.E. in 1921. He is survived by his widow, Sadie E. (Welsh) Kirberg, whom he married in 1907.

LOUIS W. KLEIN

Louis W. Klein, whose death occurred at his home in Chicago on October 7, 1931, was a native of New York, N.Y., where he was born on April 2, 1874. He spent two years at the City College of New York, and four years with Hoe & Co., where he served an apprenticeship as a machinist, and studied in its mechanical and technical school. He remained with this company until 1913, serving successively as journeyman machinist, assistant foreman of manufacturing, and production engineer. During the next two years, he was foreman, first of the Ericsson Manufacturing Company, in Buffalo, N.Y., and later for the Packard Auto Company, Detroit. From 1916 to 1925 he was connected with the Coxe Traveling Grate Company, Port Carbon, Pa., as works manager. In this capacity, he designed many special tools, fixtures, and additions to the plant. Since 1925 he had been vice-president in charge of plant operation of the Green Engineering Company, East Chicago, Ind. He was also a vice-president of the Combustion Engineering Corporation, of which the Green Engineering Company is a subsidiary.

Mr. Klein became an associate member of the A.S.M.E. in 1921. He is survived by his widow, Lillie (Sommer) Klein, and by a son and two daughters.

EDWARD KNOWLES KNOWLTON

Edward Knowles Knowlton, power superintendent of the Bayonne Works of the Standard Oil Company, died of heart trouble at his home in Bayonne on May 14, 1930. Surviving him are two children, a son and a daughter.

Mr. Knowlton was born in Albany, N.Y., on October 16, 1871. He attended the Peekskill Military Academy and served an apprenticeship as a machinist with the Schenectady Locomotive Works in 1889 and 1890. During the next two years he was an apprentice, assisting in testing street railway motors, with the Edison Machine Company, now the General Electric Company, Schenectady. From 1892 to 1895 he was connected with the Fonda, Johnstown & Gloversville R.R., equipping interurban cars and instructing operators of them. He also operated engines and generators in the power house. For a time in 1895 he was employed by the Herkimer, Mohawk & Ilion Electric R.R. at Mohawk, N.Y., as assistant engineer of construction of electric power plants and equipping cars. The following year he went to Troy to design and superintend the installation of steam power plants for the Hammett Machine Company. After two years with this firm he went to Miami, Florida, where he had charge of the power plant for the Royal Palm Hotel during its reconstruction, and subsequently served as its chief engineer and had charge of all mechanical and electrical apparatus for the hotel. He also installed a gasoline electric plant for lighting in Miami and erected transmission lines there.

From 1900 to 1911 Mr. Knowlton was connected with Westinghouse, Church, Kerr & Co. and the Westinghouse Machine Company. He conducted experimental work on large double-acting gas engines and had charge of the construction of gas-electric power plants for the Winchester Repeating Arms Company of New Haven, the Atlantic Refining Company, Philadelphia, and other companies, and for the Pennsylvania State College, where he also was instructor for about three months in gas engineering. During this period he designed a combined governing and lubricating system for the Westinghouse gas engines. For two years after leaving the Westinghouse Company, he was identified with the Roessler & Hasslacher Chem. Co., Perth Amboy, N.J., where he was engineer in charge of the construction of a gas electric power plant extension.

Mr. Knowlton had been associated with the Standard Oil Company since 1913. He had charge of all power apparatus, steam, gas and electric; reconstructed the boiler plants for the company; designed electric drives for new box factory machinery; and introduced new methods of operation and improvements in the boiler and electric plants which resulted in large savings in generating steam and electricity.

Mr. Knowlton had been a member of the A.S.M.E. since 1915.

WINFIELD BRADFORD KNOWLTON

Winfield Bradford Knowlton, supervising engineer of the American Woolen Company, Shawsheen Village, Andover, Mass., died at his home in Andover on February 17, 1931, after a few days' illness. He is survived by his widow, Mrs. Minnie Knowlton, and by one daughter, Marjorie Knowlton.

Mr. Knowlton was born at Lowell, Mass., on May 18, 1877, the son of George Winfield and Etta F. (Watson) Knowlton. He attended the Lowell Textile School and then served an apprenticeship with the Kitson Machine Company and the Tremont & Suffolk Mills, at Lowell. He became connected with the American Woolen Company as draftsman in 1899 and was appointed construction engineer in 1918. He succeeded to the office of supervising engineer of the company on January 1, 1920, taking the place made vacant by the death of William S. Whitney. Among the buildings erected under Mr. Knowlton's supervision were two brick mills at the Assabet Mills, Maynard, a concrete mill and a concrete storehouse at the Riverina Mills, West Medford, two mills and a storehouse at the Washington Mills, Lawrence, two mills, a power house, and a ten-story warehouse at the Shawsheen Mills, and a number of structures in Shawsheen Village, including an administration building. He also supervised the construction of the Andover Country Club House.

Mr. Knowlton became a member of the A.S.M.E. in 1912. He belonged to the Andover (Mass.) Country Club and was a thirty-second degree Mason.

FREDERICK GOTTHOLD KRETSCHMER

Frederick Gotthold Kretschmer, whose death occurred on June 14, 1931, was born at Stuttgart, Germany, on May 15, 1862. He studied classical languages and mathematics at the Realgymnasium in that city, and later attended the Baugewerkschule there. He came to the United States in 1881 and secured his early shop experience with R. Hoe & Company, New York, N.Y. In 1893 he represented F. Krupp of Essen, at the Chicago exposition. The following year he became manager of the New York office of the Brown & Sharpe Mfg. Co. of Providence, R.I.

Mr. Kretschmer returned to Europe in 1897 as representative of the Brown & Sharpe Mfg. Co. In 1899 he was appointed technical manager of Gustav Diechmann & Son, in Berlin, agents, not only

for the Brown & Sharpe Mfg. Co., but also for the Pratt & Whitney Co., Niles-Bement-Pond Company, and many other American manufacturers. From 1901 to 1905 he represented the Bergmann Electrical Works in London.

F. G. Kretschmer & Co. was formed in 1905 at Frankfort on the Main, acting as agents for the Brown & Sharpe Mfg. Co., the Pratt & Whitney Co., the King Machine Tool Company, of Cincinnati, Ohio, and other American manufacturers. In 1912 the firm became agent for the Taylor Manufacturing Company of Philadelphia, in Continental Europe. Several other American machine-tool builders enjoyed representation by Mr. Kretschmer.

As agent for these American companies Mr. Kretschmer specialized in the practical and commercial end of the machine tool business, introducing new machines and methods throughout Europe. He had contributed a number of articles on machine tool industry to the technical press.

Mr. Kretschmer became an associate of the A.S.M.E. in 1895. He resigned his membership in 1908 but was reinstated in 1913. He was transferred to the grade of full membership in 1928. He was also a member of the Verein Deutscher Ingenieure and Verein Deutscher Giesserei-Fachleute. He is survived by his widow, Mrs. Linda Kretschmer.

JAMES B. LADD

James B. Ladd, the son of William W. Ladd and Sarah H. (Phillips) Ladd, was born at Througe Neck, Westchester County, N.Y., on June 27, 1860. He was given an M.E. degree by the Stevens Institute of Technology in 1881. He spent the next six years as draftsman and mechanical engineer for the Southwark Foundry & Machine Co. of Philadelphia. From 1887 to 1890 he was chief engineer in charge of design and construction of the Maryland Extension of the Pennsylvania Steel Company at Sparrows Point, Md. This company was later known as the Maryland Steel Company and is now part of the Bethlehem Steel Company. Mr. Ladd resigned his position with the company in 1890 to become chief engineer for the Robert Poole & Son Co. of Baltimore, with whom he remained for six years. Ill health caused his retirement from active business for about two years, after which he opened an office in Philadelphia for consulting work.

Mr. Ladd became a member of the A.S.M.E. in 1884. He was married in 1889 to Rebecca H. Serrill, of Darby, Pa., who died in 1927. One daughter, Frances (Ladd) Keicher, survives him.

GEORGE CRAMTON LAMMERT

George Cramton Lammert died at Oak Park, Ill., on October 25, 1931, after a month's illness. Since his graduation from the Massachussets Institute of Technology with a B.S. degree in 1927, he had been associated with his father in the engraving and engineering business of Lammert & Mann, Chicago, Ill.

Mr. Lammert was born in Chicago on December 24, 1903, the son of Ferdinand E. and Florence J. Lammert. He secured his preparatory education at the Culver Military Academy. He is survived by his widow, Eleanor (Danker) Lammert, and infant daughter, Barbara.

A. ARCHER LANDON

A. Archer Landon, president since 1925 of the Landon Radiator Co., Inc., died on January 5, 1930. He is survived by his widow, Emma J. Landon, whom he married in 1891, and by one married daughter, Mrs. Hilda (Landon) Conant, of Dedham, Mass.

Mr. Landon was born at Detroit, Mich., on February 5, 1868, the son of John Mason and Melvina (Akerman) Landon. He supplemented his education in common and evening schools with an apprenticeship as a machinist with the Detroit Bolt & Nut Corp. After the completion of his apprenticeship in 1887, he was employed by the American Radiator Company, serving successively as foreman of the machine shop, toolmaker, and superintendent of its Kalamazoo plant, manager of its Titusville plant, assistant manager of its Pierce plant, general superintendent of plants, and vice-president. He was located at Buffalo, N.Y., from 1902 until 1930, with the exception of four years (1905–1909), when he was general superintendent of the manufacturing department in Chicago.

Mr. Landon was a member of several clubs in Buffalo and vicinity, and during the World War was connected with its Chamber of Commerce. He became a member of the A.S.M.E. in 1902. He was a member of President Wilson's Industrial Conference Board and chief of aircraft production of the National Industrial Conference Board. He was active in organizing War Labor Boards.

A number of United States patents, as well as many in England, Germany, France, and Italy, were taken out in Mr. Landon's name and assigned to the American Radiator Company, the Crane Company, and Kalamazoo Spring & Axle Co. These included radiators and radiator sections, molding and casting machines, ramming and heating apparatus, pneumatic tongs, and other equipment.

OLIN HENRY LANDRETH

Olin Henry Landreth, consulting engineer of New York, died of pneumonia on November 6, 1931. He is survived by two daughters, with one of whom he made his home in Mount Vernon, N.Y., and by a son.

Professor Landreth was born at Addison, N.Y., on July 21, 1852, the son of the Reverend James and Adelia (Comstock) Landreth. He attended academies at Pen Yan, Dundee, and Canisteo, N.Y. He then entered Union College, from which he received a C.E. degree in 1876, A.B. in 1877, A.M. in 1880, and honorary D.Sc. in 1905.

During college vacations Professor Landreth secured some early experience in various railway positions and in making surveys and estimates for water works at Schenectady and Olean, N.Y. He was an instructor in the physical laboratory at Union College in 1876 and 1877, professor of civil engineering at Vanderbilt University from 1879 to 1894, and dean of the engineering department, which he initiated there, from 1886 to 1894. He then returned to Union College as professor of engineering, remaining there until 1919. Since that date he engaged in consulting engineering in New York. He was also chief engineer of the Eastern Potash Corporation from 1919 to 1921.

While at Vanderbilt University, Professor Landreth was superintendent of buildings and grounds for about eight years, in charge of new construction and the maintenance and operation of water supply, sewerage, and steam heating systems. He also made surveys, estimates, and investigations for railroads, water works, and electric light systems in several southern cities. He served as consulting engineer to the Board of Public Works in Nashville from 1883 to 1885. He was consulting engineer to the Davidson Company, the Tennessee Bridge Commission, the West Nashville Water Company, and the Tennessee State Board of Health. He made examinations and reports on smoke prevention and on the sanitary effect of the construction of proposed locks and dams on the Cumberland River. In 1893 he served as special commissioner to the World's Fair in Chicago.

After returning to New York, Professor Landreth served as consulting engineer to the New York State Board of Health from 1896 to 1901 and for five years thereafter was connected with the State Department of Health. He was a member of the New York State Water Storage Commission in 1902 and 1903, the New York Bay Pollution Commission from 1903 to 1905, and the Metropolitan Sewage Commission of New York from 1906 to 1908.

Professor Landreth was assistant astronomer at the Dudley Observatory at Albany from 1877 to 1879 and had charge of the expedition to the Helderbergs in May, 1878, to observe the transit of Mercury. In 1882, he was in charge of the Vanderbilt Observatory during the transit of Venus. He was a member of the Advisory Council of the Chicago Engineering Congress in 1893 and had served on the New York State Waterways Association, the Merchants' Association of New York, and various committees on waterways, water storage, sanitation, and public health. He was a delegate to the New York State Constitutional Convention in 1915.

Professor Landreth had contributed papers to the technical press and before various engineering organizations on a wide range of subjects. He was the author of Instructions for Finding the True Meridian, With Tables for Surveyors and of Metric Conversion Tables for Engineers. He collated the civil engineering terms for the last edition of the Century Dictionary. During the World War he served as mechanical engineer in the Ordnance Department at Washington, D.C.

Professor Landreth became a member of the A.S.M.E. in 1883 and served as vice-president of the Society in 1885 and 1886. He was past-president of the American Society for Testing Materials, past director of the American Society of Civil Engineers and of the Society for the Promotion of Engineering Education, a member of the Council of the American Institute of Consulting Engineers from 1923 to 1926, past secretary of the Engineering Association of the South, past-president of the Society of Engineers of Eastern New York, and past vice-president of the American Association for the Advancement of Science.

Professor Landreth married Eliza Taylor at Canisteo, N.Y., in 1879.

GIFFORD LECLEAR

Gifford LeClear, a member of the A.S.M.E. since 1927, died on September 24, 1931, in Boston, Mass. Mr. LeClear was born at

Rutherford, N.J., on January 3, 1874. His parents were Thomas and Cornelia (King) LeClear. He received an A.B. degree from Harvard University in 1895 and his master's degree the following year. He then worked for a year in various capacities for the West End Railroad now the Boston Elevated Railroad. In 1897 he formed a partnership with Edward D. Densmore under the firm name of Densmore and LeClear. The work of the firm was initially confined to small engineering work, later developing into larger work and ultimately into a combined practice of engineering and architecture. Mr. H. C. Robbins was latter admitted to the firm, the name of which became Densmore, LeClear, and Robbins.

This firm, aside from a large architectural practice, has been connected with the installation of power plants, heating and ventilating systems, electrical and refrigeration equipment, and steam transmission systems for many building projects in Boston and vicinity as well as in other cities.

Among its clients have been the New England Telephone & Telegraph Co., The American Telephone & Telegraph Co.; The Massachusetts General Hospital, the Boston City Hospital, Newton Hospital, Beth Israel Hospital; Harvard University, Simmons College, Smith College, Williams College; Boston & Albany, Boston & Maine, and New York, New Haven & Hartford railroads; and many other private, municipal, and state interests. For a number of years the firm acted as consulting engineers for the Mutual Boiler Insurance Company of Boston.

Mr. LeClear was chairman of the Elevator Board for the state of Massachusetts for two years and in that capacity prepared one of the first standard sets of rules and regulations for elevators. He lectured at Harvard University on mechanical plants of buildings, power plants, etc. He also lectured at the Harvard Business School and Massachusetts Institute of Technology on power plant work.

For a number of years he was the chairman of the Traffic Committee of the Boston Chamber of Commerce, and as such was responsible for many of the early developments in signal control in the city of Boston.

In 1903 he married Helen Parker of Boston, and their children are Margaret (LeClear) Gray and Donald LeClear.

JOHN GEORGE LEHMAN

John George Lehman, president and general manager of the Bethlehem Foundry & Machine Co., Bethlehem, Pa., died on May 14, 1931.

Mr. Lehman was born in Bethlehem on January 15, 1874, the son of Bernhard E. and Harriet M. (Fetter) Lehman. He prepared for Lehigh University at the Moravian Preparatory School in Bethlehem, and began the mechanical engineering course at Lehigh in 1892. Following his graduation in 1896, he devoted himself to the operation of the brass foundry and machine shop which his father, who died in 1892, had owned. This foundry was established in 1832 by his grandfather. He sold the business in 1902 to the Bethlehem Steel Company and entered its employ as superintendent of its first brass foundry department, the layout for which was made by him.

Mr. Lehman became identified with the Bethlehem Foundry & Machine Co. in 1906, when it took over the old brass works plant. His work included the development of new designs of apparatus as applied to the chemical industry for the production of acids, dyes, and inorganic compounds, the development of special irons for use in such apparatus to resist chemical action, and the development of machinery for cement plants. He was in direct charge of all sales and all operating departments of the company.

In addition to the A.S.M.E., which he joined in 1921, Mr. Lehman belonged to the American Iron and Steel Institute.

He is survived by his widow, Emily M. (Myers Fuller) Lehman, whom he married in 1918.

WARREN B. LEWIS

Warren B. Lewis, consulting engineer of Providence, R.I., who for the last 35 years had specialized in the design and operation of industrial power plants, died on July 18, 1931.

Mr. Lewis was born at Providence on April 10, 1870, the son of Mr. and Mrs. George W. Lewis. After completing his course at the English High School in that city, he worked in the laboratory at Cornell University with Prof. W. A. Anthony on mechanical and electrical problems. He then spent two years in the experimental department of the Thomson Electric Welding Company, Lynn, Mass., two years in the district engineer's office of the General Electric Company, and one year as manager of central stations for the United Traction & Electric Co.

In 1896 he opened his own office in Providence as a consultant on power plant problems.

He was interested both in the design and construction of industrial power plants and in the efficiency of their operation, in connection with which he studied processes, equipment, and personnel. He had contributed a number of articles on these subjects to the technical press.

Mr. Lewis became a member of the A.S.M.E. in 1915 and also belonged to the American Institute of Electrical Engineers and the Providence Engineering Society. Surviving him are his widow, Lela V. (Hutton) Lewis, whom he married in 1901, and three daughters.

LEROY N. LIEBERKNECHT

Leroy N. Lieberknecht was born at Letts, Iowa, on April 6, 1895, and died on March 13, 1931, at Beloit, Wis. His parents were Henry and Salome (Wiedenecht) Lieberknecht. He received his education in the public schools of Iowa. His work at Iowa State College was dropped that he might enlist for service in the U.S. Army during the World War. After being honorably discharged from Camp Dodge, Iowa, he completed his college work, being graduated from the engineering school at Ames in 1920. He was a member of Tau Beta Pi, honorary engineering fraternity, and of Theta Nu Epsilon.

Following his graduation he accepted a position with Fairbanks, Morse & Co. at Beloit. For several years he was engaged in testing and development work in the Experimental Department, and at the time of his death was head shipping inspector in the Large Engine Department. He had been an associate member of the A.S.M.E. since 1926.

He is survived by his widow, Frances (Newell) Lieberknecht, whom he married in 1923, and by three daughters.

HENRY P. LUDEWIG

Henry P. Ludewig was killed in an airplane crash at the Thornburg Country Club, Pittsburgh, Pa., on June 6, 1931. He had been an associate member of the A.S.M.E. since 1930.

Mr. Ludewig was born in New York, N.Y., on September 19, 1897. He took courses in mathematics, drafting, and engineering in evening trade schools. From 1918 to 1920 he was in charge of the development of moving picture apparatus for the Daylight Film Corporation, New York. During the next five years he was plant superintendent in charge of all tooling, designing, and production connected with an adding machine for the Type Adder Corporation, New York. He then spent three years as salesman of lubricating oils and greases for the Vacuum Oil Company, New York, and a year as salesman of plain and fabricated steel for Egleston Bros. & Co., Inc., Long Island City, N.Y. Since 1929 he had been sales engineer for the Carboloy Co., Inc., Pittsburgh, Pa. His duties included the analysis of jobs and the design of tools to speed production.

Mr. Ludewig is survived by his widow, Olive Marguerite (Hansen) Ludewig, whom he married in 1922.

CHARLES W. LUHR

Charles W. Luhr died suddenly at his home in West Orange, N.J., on December 21, 1930. He is survived by his widow, Kathryn (Drudy) Luhr, whom he married in 1903, and by three sons and a daughter.

Mr. Luhr was born at Ober Ingelheim, Germany, on November 15, 1873, the son of John and Elizabeth (Buchenhauer) Luhr. His education was obtained in the United States where he attended public and private schools and Cooper Union, New York, N.Y. In 1887 he was apprenticed to Hugo Kayser, New York, where he was engaged in experimental work and in making models and special machinery. After two years with Mr. Kayser, he was apprenticed to the Western Electric Company, New York, on telephone and telegraph instrument and switchboards, and also in 1889 was employed by the Standard Cash Register Company of Orange, N.J.

Mr. Luhr's first association with the Edison interests was in June, 1889, in the Edison Phonograph Works at West Orange, where he was employed in the manufacture of phonographs until 1903. He was then promoted to the position of foreman with the Edison Manufacturing Company, making electro-therapeutic instruments, X-ray apparatus, fan motors, moving picture machines, and other special appliances. In 1911 he was made foreman of the second floor of Mr. Edison's laboratory, engaged in experimental and precision work. Two years later he was made superintendent of the laboratory in charge of all work, under the immediate direction of Mr. Edison, and in 1917 was promoted to the vice-presidency and general managership of the Edison Phonograph Works, a position which he occupied at the time of his death.

During the early days of the motion picture industry, Mr. Luhr was

instrumental in perfecting and improving projecting machines. He also helped to perfect the first apparatus for laboratory development and projections of the motion picture film now known as the 16 MM. and, in conjunction with Adolph Gall, perfected the first daylight printing machine for the use of the Edison home kinetoscope. He was granted a patent for a machine for shaving sound record tablets and various other patents which were turned over to the Edison Company. He read widely on the subject of mechanics, had a thorough knowledge of higher mathematics, and assisted in solving many mechanical difficulties.

In 1901, at the time of the assassination of President McKinley, Mr. Luhr was sent by Mr. Edison to Buffalo with X-ray instruments to assist physicians in locating the bullet. Mr. Luhr became a member of the A.S.M.E. in 1916. He was president of the Edison Building and Loan Association, a member of the Elks and the Foresters of America, and interested in all civic matters in the Oranges.

FRANK MAGAW MALANY

Frank Magaw Malany, president and general manager of the F. M. Malany Co., New York, N.Y., died in Brooklyn, N.Y., on March 22, 1931.

Mr. Malany was born on August 22, 1882, at Chester, Pa., the son of Magaw and Isabel (Traynor) Malany. He served a five-year apprenticeship in the Seattle Machine, Boiler & Pipe Shop. From 1903 to 1916, with the exception of three years as chief engineer of the Philippine Steamship Company, he belonged to the Army Transport Service, serving in the Philippines, China, France, and on both the Atlantic and Pacific coasts. He received his first chief engineer's license at San Francisco in 1914, and later secured a chief engineer's license for any ocean. He retired from active service with the rank of Captain. From 1916 until 1918 he engaged in the sale of machinery and supplies.

As a Captain in the U.S. Corps of Engineers during the World War, Mr. Malany spent 13 months at Bordeaux, France, in charge of erecting machine shops, pumping stations, and refrigerating plants and repairing steamers arriving at that port. After returning to this country in 1920 he was appointed assistant superintendent engineer by the War Department. For a time he was stationed at Hoboken, N.J., to superintend repairs on transports, and then he directed the construction and conducted trials of ten self-propelled barges for use on the Erie Canal between New York and Buffalo. After the fleet had been put into operation he was assigned as chief engineer in charge of maintenance and repairs on these and other barges.

In 1921 Mr. Malany became connected with the United States Mail Steamship Company with the rating of assistant to Captain O. W. Koester, vice-president in charge of construction and repairs. Mr. Malany directed the reconditioning for passenger service of several steamers used for transport service during the War.

From 1922 to 1924 Mr. Malany was sales engineer for marine machinery and equipment, and represented the firm of Griffin & Cathcart, consulting engineers, as professional observer in such tests as the trial trip of the S.S. *Leviathan*. For two years, beginning in 1924, he was manager of the Marine Department, New York, of the Hagan Corporation, combustion and chemical engineers, Pittsburgh, Pa. He then spent a few years as manufacturers' agent, and in 1927 organized the firm of F. M. Malany Co., Inc. He kept in close touch with marine affairs as head of this company and also as a marine distributor for the Westinghouse Elec. & Mfg. Co.

Mr. Malany became a member of the A.S.M.E. in 1925. He also belonged to the American Society of Naval Architects, the Society of American Military Engineers, and the Reserve Officers' Association of the United States. He was an active member of the Masonic fraternity and belonged to a number of clubs, including the Propellor Club of the United States, Port of New York, and the Marine Square Club, of New York. He was buried in the National Cemetery at Cypress Hills, Brooklyn.

GUSTAV FRANCIS STOVEL MANN

Gustav Francis Stovel Mann, estimator for Harlow Lewis, New York, N.Y., died on March 5, 1931, after a short illness. He is survived by his widow, Gwendolyn L. (Stapleton) Mann, whom he married in 1923.

Mr. Mann was born in Edinburgh, Scotland, on June 16, 1892, the son of Gustav and Agnes (Sinclair) Mann. His technical education was obtained at Vanderbilt University, Nashville, Tenn., from which he received a B.S. degree in 1918. He taught thermodynamics at the university during his last year there.

For a number of years Mr. Mann engaged in shipbuilding. His first experience in this work was secured in the summer of 1917, when he was in charge of hull drafting for two mine sweepers built by the United States at Shooters Island, N.Y. Following the completion of his college work he took charge of hull and engine drafting and routing of materials for the Alabama Dry Dock & Shipbuilding Co., Mobile, Ala. Articles by him on the "Effective Arrangements of Departments in Shipyard Organization" and "Effective Layout of the Fabricating Shop in Shipyards," were published in the November, 1918, and March, 1919, issues, respectively, of *International Marine Engineering*. In August, 1920, the same journal published a paper on "A Method of Estimating Steel Hull Weights," which dealt with the work he did in 1920 for the Bethlehem Shipbuilding Company, Bethlehem, Pa. He also was in charge of fitting out three ships built by that company at Three Rivers, Quebec, Canada. Mr. Mann also worked for a few months in the production department of the Newburgh Shipyards, Inc., was traveling hull inspector for the United States Shipping Board, in the Gulf District, and designed barges for the transportation of fuel oil for the Tidmex Oil Company, Tampico, Mexico.

During part of 1921 and 1922 Mr. Mann was acting division office engineer for the Tennessee State Highway Department and supervised the construction of some Federal Aid roads. He then went to Barahona, Dominican Republic, as surveyor for the West Indies Sugar Finance Corporation. After returning to the United States he was connected successively with the Charles Ward Engineering Works as ship draftsman at the Yedo Yatch Basin, Brooklyn, the Bartlett-Hayward Company, New York, N.Y., as valuation engineer, the New York State Bridge & Tunnel Comm., New York, as assistant engineer, the Bridge Department of the Delaware, Lackawanna & Hudson R.R. Co., Hoboken, N.J., as designer, and from 1927 to 1930 with the New York Central Railroad, New York, as assistant engineer in the design department of that road.

Mr. Mann became an associate-member of the A.S.M.E. in 1923 and also belonged to the National Geographic Society and the Army Ordnance Association.

ARTHUR STENNETT MARTIN

Arthur Stennett Martin, manager of the Firthite Division and of the Engineering Department of the Firth Sterling Steel Company, McKeesport, Pa., with which he had been connected since 1927, died at the Homeopathic Hospital in Pittsburgh on April 21, 1931.

Mr. Martin was born in Elizabeth, N.J., on October 16, 1885, the son of George J. and Norma (Gartz) Martin. He was graduated from the Battan High School in Elizabeth in 1903 and took a commercial course in the Elizabeth Commercial College. He was then employed by the Pennsylvania Railroad in New York, N.Y., resigning to take a position with B. Nicol & Co. He entered Cornell University in 1906 and was graduated in 1910 with an engineering degree. While at Cornell he joined the Theta Xi Fraternity, won his numerals rowing in the Freshman Crew, was a member of the Varsity Crew Squad, and on the Board of the Sibley Journal of Engineering; after graduation he became a director of the Theta Xi Association of Cornell. During summer vacations he had worked in the shops of the Singer Manufacturing Company, Brunsdorf-Miller Tool Company, and Niles-Bement-Pond Co., and he became sales engineer in charge of the New York State territory for the latter company in 1911.

In 1915 he accepted a position as construction superintendent and treasurer of the Moyer Engineering & Construction Co., Brooklyn, N.Y., specializing in the design and erection of reinforced-concrete industrial buildings. While with this company he worked on the reconstruction of the Edison Plant in West Orange after it had been partially destroyed by fire, and among other things he designed and constructed an addition to the R. G. Dunn Printing Plant in Brooklyn.

In 1916 he became associated with Newton C. Failor to form the Failor-Martin Corporation., which represented various machine manufacturers in the New York Metropolitan area. In 1918 he designed and directed the construction of a machine shop for the Virginia Iron Works, Norfolk, Va., and selected and laid out the equipment for it.

The latter part of 1919 the Failor-Martin Corporation sold their machine business and started the manufacture of stainless steel products, specializing in kitchen cutlery. Mr. Martin had entire charge of the manufacturing operations, involving the development of the original equipment and processes. He worked out many new methods and his process of etching was later installed and used in other similar factories. The plant was originally located in Bloomfield, N.J., and later moved to larger quarters in Belleville, N.J.

In 1927 Mr. Martin became associated with the Firth Sterling organization of McKeesport, Pa., and was located in the New York office where he was in charge of advertising and also engaged in the application and heat treatment of stainless steel for industrial purposes, at the same time specializing in super-high-speed steels. In 1929 he went to McKeesport, as the manager of the Engineering

Department and became also manager of the Firthite Division, in charge of the manufacture and sale of tungsten-carbide tools.

Mr. Martin became a member of the A.S.M.E. in 1928. He was also a member of the American Society for Steel Treating, the Masonic fraternity, and the Engineers Club of Montclair, N.J. Surviving Mr. Martin are his widow, Adaline (Moyer) Martin, and two daughters, Clara and Katharine, all of Pittsburgh, Pa.

ALAN RAMSAY McFARLAND

Alan Ramsay McFarland, who died in Syracuse, N.Y., on January 13, 1931, was born in Painesville, Ohio, on May 26, 1870. His parents were Walter and Mary (Ashfield) McFarland. He attended public schools in Oswego, N.Y., and New Haven, Conn., and took a special three-year course in mechanical engineering at Cornell University.

In July, 1891, Mr. McFarland entered the employ of the Solvay Process Company, Syracuse, as assistant to the manager of the Bicarbonate Department. He was made acting manager of that department the following year. During the next 15 years, he was connected with the Soda Ash Department, first as assistant to the manager, and later as manager. From 1907 to 1915, he was works manager of the company, and during the next seven years chief engineer. In 1922 Mr. McFarland accepted a position as chief engineer of the Charcoal Iron Company of America, at Marquette, Mich., where he remained for four years. Since that time he had been connected with the Onondaga Brick & Tile Corp., of Syracuse, N.Y., as president since 1928.

Mr. McFarland became a member of the A.S.M.E. in 1902. He is survived by his widow, Ella (Belden) McFarland, whom he married in 1896, and by two sons and a married daughter.

CLARK YOUNG McGOWN

Clark Young McGown, associated with the Consolidated Gas Company of New York, N.Y., since early in 1929, died on January 17, 1931. He was a native of New York, where he was born on January 9, 1895, the son of Clark Elbert and Ettie McGown. He was graduated from the Stevens Institute of Technology with an M.E. degree in 1916. During the next two years, he was employed by the British Guiana Sugar Factories, Ltd., of Demerara, British Guiana, as second operating engineer in its sugar factory at Plantation Wales. His work during this time included the study of the analysis and chemical control of sugar in the laboratory, and the supervision of factory operation and of overhauling of old machinery and installing new machinery.

During the World War, Mr. McGown served in the Ordnance Department of the Army, being stationed at Camp Hancock, Ga. He re-entered the employ of the British Guiana Sugar Factories as chief engineer in 1919 and remained there for two years. He then returned to the United States as inspection foreman for the Western Electric Company in Newark, N.J. The following year he became superintendent of shops and the commercial distribution departments of the Northern Union Gas Company in the Bronx, N.Y., a position which he held until he became affiliated with the Consolidated Gas Company in 1929.

Mr. McGown's first position with the Consolidated Gas Company was that of superintendent of the Utilization Laboratory and of the Refrigeration and Maintenance Departments. Later, he took charge of two warehouses and fleets of trucks handling appliances for the company. In 1929 he was given the position of engineer of utilization, supervising these various departments.

During his connection with the British Guiana Sugar Factories, Mr. McGown designed a new link belt chain carrier for conveying sugar cane from an unloading platform into the mill, and other equipment for the company.

Mr. McGown was a student member of the A.S.M.E. while at the Stevens Institute of Technology and in 1918 became a junior member of the Society. He was promoted to the grade of associate membership in 1922 and to full membership in 1930. He also belonged to the American Gas association and the Masonic fraternity.

Surviving Mr. McGown are his widow, Mildred (Burr) McGown, whom he married in 1919, and two sons, Kenneth and Donald.

ELWOOD FRANCIS McLAUGHLIN

Elwood Francis McLaughlin, for twenty-five years connected with the General Electric Company, died on January 8, 1931, at Atlanta, Ga. Mr. McLaughlin was born in Philadelphia, Pa., on October 4, 1883. He was graduated from the Pennsylvania State College in 1905 with the degree of B.S. in Mechanical Engineering.

Following his graduation, Mr. McLaughlin spent five years with the General Electric Company at Lynn, Mass. Here he had experience in the Steam Turbine Testing Department and the Designing Department, was for two years chief computer for the company, and served as assistant engineer for more than a year. He was then transferred to the Railway and Lighting Department of the Company at Atlanta. For a number of years he represented the company, both in an engineering and commercial capacity, in connection with the sale and installation of steam turbines and electrical equipment for central stations. Much of his time was also devoted to the study of transportation conditions and the requirements of street railway companies in cities throughout the entire Atlanta district, which comprised all the southeastern states.

At the time of his death Mr. McLaughlin was district manager of the Central Station Department of the General Electric Company at Atlanta.

Mr. McLaughlin became a member of the A.S.M.E. in 1919. He is survived by his widow, his daughter, Dorothy F. McLaughlin, two sons, James A. and Elwood F. McLaughlin, Jr., all of Atlanta, and by his mother, Mrs. James F. McLaughlin, of Philadelphia, Pa.

NATHAN ATHERTON MIDDLETON

Nathan Atherton Middleton, of Middleton & Co., Inc., Cleveland, Ohio, died on May 4, 1931. He was born at Louisville, Ky., on February 28, 1887, the son of Harvey and Cornelia Speake (Scribner) Middleton. He was graduated from the Massachussetts Institute of Technology with an S.B. degree in mechanical engineering in 1907 and during the next two years was a special apprentice in the machine and erecting shops of the American Locomotive Company, Richmond, Va. In 1908–1909 he was an assistant in the foundry of Bartlett-Hayward Company, Baltimore Md., following which he spent two years with the Thos. C. Basshor Co. of that city, designing boilers and special machinery and during part of the time serving as general superintendent of the boiler and machine departments.

In 1911 the Whiting-Middleton Construction Company, Baltimore, was organized to handle concrete sewer, bridge, and road construction, with Mr. Middleton as its president. Two years later it was reorganized as the Middleton-Thompson Company, and continued in business until 1915. After engaging in consulting work covering valuation reports for the Western Union Telegraph Company and factory investigations for the Boston Confectionery Company, Cambridge, Mr. Middleton accepted a position with Hornblower & Weeks, Boston, as engineer in charge of the investigations and reports regarding industrial corporations which that company financed or proposed to finance. He also represented the company on the boards of directors of its clients and served a number of companies as consulting engineer, designing and constructing additions and improvements in the plants of the Parish & Bingham Corp. and Ohio Body & Blower Co., Cleveland, Mullins Body Corporation, Salem, Ohio, Will & Baumer Co., Syracuse, N.Y., Lee Tire & Rubber Co., Conshohocken, Pa., and Atlas Tack Company, Fairhaven, Mass.

In 1922 Mr. Middleton went to Cleveland, continuing his connection with Hornblower & Weeks and at the same time serving as president of the Ohio Body & Blower Co. The following year he was made president of the Cleveland Windshield Company. He severed these connections in 1927 when he became a partner in the firm of Middleton, Eden & Co., Cleveland. This company was reorganized the following year as Middleton, Worthington & Co., Inc., and shortly before Mr. Middleton's death became Middleton & Co., Inc. Mr. Middleton had also been director of the Swartout Company, Cleveland.

Mr. Middleton was commissioned a captain in the 23d Engineers in 1917, and in 1918 and 1919 was a major, commanding the 3d Battalion, 23d Engineers, in France, and was also regimental adjutant. He worked on the design and construction of army railroad yards and storage depots for munitions, and during the Argonne-Meuse Offensive served as executive officer of the engineer of roads, First Army, in charge of all road construction of the Army. After the Armistice until the end of May, 1919, he had entire charge of all engineering work in the Second Army area. He also was regimental historian.

Mr. Middleton was elected an associate member of the A.S.M.E. in 1917 and was promoted to full membership three years later. He belonged also to the American Society of Military Engineers and a number of clubs in Boston, New York, and Cleveland. He married Mina Winslow DeHart in 1913 and is survived by her.

WALTER MILLER

Walter Miller, a member of the A.S.M.E. since 1886, died on February 26, 1930. Mr. Miller was born in Scotland. He learned the machinist's trade, after which he worked for three years at Springfield, Ohio, as draftsman for Whiteley, Fassler & Kelley. He then held a similar position for six years at the Globe Iron Works in

Cleveland, and from then until his retirement conducted a consulting practice as a mechanical engineer in Cleveland. Since his retirement from active business he had made his home at Painesville, Ohio.

THOMAS WILLIAM MILNOR

Thomas William Milnor, president of the Kline Hardware Company, Inc., Allentown, Pa., since 1913, died on November 12, 1930. He was born at Burlington, N.J., on October 27, 1866. He attended Stevens Institute of Technology and was graduated with a degree in mechanical engineering from Cornell University in 1889. His early practical experience was secured as special apprentice at the Jeansville Iron Works of the J. C. Haydon & Co.

Following his graduation, Mr. Milnor was employed by the Link-Belt Engineering Company for two years, and by the McNeal Pipe & Foundry Co. of Burlington, N.J., as superintendent in 1893. During the next five years, he managed his father's coal business and designed and built a plant for the Sanitary Enameled Clay Company, in Burlington, for the manufacture of enameled bricks. From 1899 to 1903, he was connected with the Borgner Cement Company and interested in firebrick. He accepted the office of treasurer for the Mickle-Milnor Engineering Company of Philadelphia in 1903. This company was devoted to the selling of radial brick chimneys, air compressors, and Corliss engines. From then until the establishment of the Kline Hardware Company, Mr. Milnor was associated with the Donaldson Iron Company of Emaus, Pa., engaged in the design and construction of their pipe foundry plant.

Mr. Miller became a member of the A.S.M.E. in 1915.

HARRY MORTIMER MONTGOMERY

Harry Mortimer Montgomery, retired mechanical engineer, died suddenly on February 24, 1931, of a heart attack, at his home in Evanston, Ill.

Mr. Montgomery was born in Lafayette, Ind., on July 28, 1861, the son of Henry and Estelle Helène (Von Flierden) Montgomery. He took a special course in mechanics at the Massachusetts Institute of Technology from 1877 to 1879, and then spent three years as apprentice and draftsman in the locomotive shops of the New York & New England R.R., at Readville and Norwood, Mass. During 1882 and 1883 he was engaged by the Hinkley Locomotive Works, Boston, to make detail drawings of locomotives and to erect them at Culebra for the Compagnie Universelle du Canal Interoceanique. Returning to this country upon the completion of this work, Mr. Montgomery became chief draftsman for the Rhode Island Locomotive Works, Providence, R.I. After a short time with this company he accepted a similar position with the New York, Lake Erie & Western R.R., being stationed at the Susquehanna, Pa., locomotive shops.

Mr. Montgomery returned to New England in 1885 to take charge of the design of locomotives, passenger and freight cars, and machine tools for the New York & New England R.R., at Norwood, and also had charge of the drafting office, pattern shop, and brass foundry. After six years in this position he was engaged for about a year each by the Boston Bridge Company, Boston, Morton Safety Car Heating Company, Baltimore, Md., Babcock & Wilcox Co., New York, N.Y., A. C. Frost & Co., Chicago, Ill., and Munoz Boiler Company, New York, N.Y. His work for these companies related to design and construction, and, in the case of Babcock & Wilcox, reorganization of the drafting office and force and systemization of drawings and other engineering records.

After a few years as manufacturer's agent in New York, Mr. Montgomery went to Chicago in 1905 as manager of the Alberger Condenser company. Upon its reorganization under the name of the Alberger Pump & Condenser Co., in 1913, he was appointed district manager. He retired from this position the following year to become a member of the firm of Humphrey and Montgomery, Chicago, consulting and constructing engineers. In 1916 and 1917 he designed and superintended the construction of a pulp mill digester house for the Eastern Manufacturing Company, Bangor, Me.

In 1918 Mr. Montgomery entered the U.S. Army as a major in the Quartermaster Corps. He directed the construction of warehouses, water works systems, and a fire engine house at Jeffersonville, Ind., and St. Louis, Mo. After the close of the war he was appointed commissioner of public works and city engineer by the City of Evanston, where he had made his home for some years. He retired from business upon the completion of this service in 1920.

Mr. Montgomery had been a member of the A.S.M.E since 1889. He also belonged to the Engineers' Club in New York, the Chicago Engineers' Club, and the Technology Club. Surviving him are his widow and five children, Helen and Louis W. Montgomery of Evanston, H. M. Montgomery of New York, and Mrs. Herbert G. Clifford and Mrs. Richard M. Illsley of Pasadena, Calif.

J. CLARK MOODY

J. Clark Moody, experimental engineer and head of the experimental department for the Chicago Pneumatic Tool Company, Franklin, Pa., died at Garden City, Kansas, while visiting there on March 14, 1931.

Mr. Moody, the son of Alpheus Phelps and Mary Elizabeth (Wheaton) Moody, was born on August 15, 1893, at Judsonia, Ark. He attended high school at Bald Knob, Ark., and was graduated from the University of Arkansas in 1919, receiving his mechanical engineering degree three years later. His early experience was secured during summer vacations as foreman in charge of locomotive and repair shop equipment on small logging railroads in Arkansas. Following his graduation he secured a position as test engineer with the Chicago Pneumatic Tool Company, Franklin, Pa., his duties including the erection and test of steam, electric, and oil-engine-driven air compressors and Diesel and semi-Diesel oil engines. After four years with this company, he went to Cambridge, Mass., as instructor in the mechanical engineering department of the Harvard Engineering School. He returned to the employ of the Chicago Pneumatic Tool Company after one year at Harvard and remained with it until his death. He had full charge of all experimental work done by the company. His experiments resulted in a number of inventions in the field of pneumatic tools.

Mr. Moody became a member of the A.S.M.E. in 1928 and also belonged to the Pennsylvania Engineering Society and the American Legion. He was in the Coast Artillery Service at Fort Monroe, Va., during part of the World War.

Mr. Moody is survived by his widow, Ruth (Smith) Moody, of Garden City, Kansas, whom he married in 1920.

MILLER F. MOORE

Miller F. Moore, who died of heart failure at Roselle, N.J., on November 29, 1930, was born at Corning, N.Y., on May 29, 1842. His parents were Samuel Lovell and Frances Ann (Armstrong) Moore. He was educated in the public schools and Christ Church School at Elizabeth, N.J. At the age of 19 he entered the employ of Bridges and Lane, New York, N.Y., building and selling machine tools and supplies. After five years with this firm he became secretary and general manager of the Locomotive Engine Safety Truck Company, in New York, where he remained until 1881. At that time he accepted a position as general manager for the Edison Company in New York. In 1884 he was general agent of Armington & Sims Co. for a short time, and during the next 17 years secretary and treasurer of Samuel L. Moore & Sons Co. at their Crescent Ship Yard and Iron Works, Elizabeth, N.J. From 1901 to 1903 he was president of the Marine Engine & Machine Co., Harrison, N.J. He retired in 1904.

Mr. Moore became a member of the A.S.M.E. in 1891 and also belonged to the Naval Architects and Marine Engineers, the Masonic fraternity, and the Machinery and Lawyers Clubs in New York. He is survived by his widow, Helen Serena (Brown) Moore, whom he married in 1867, and by a son and a daughter.

WILLIAM JAMES PERRY MOORE

William James Perry Moore, whose death in Detroit, Mich., on November 2, 1930, was caused by a cerebral hemorrhage, was born in New Britain, Conn., on May 4, 1858. He was the son of James Pericles and Lucy Eliza (Clark) Moore. After completing the dynamical engineering course at the New Britain High School, he entered the Sheffield Scientific School of Yale University which conferred a Ph.B. degree upon him in 1878. After his graduation he was connected for a short time with the New York office of the Buckeye Engine Company of Ohio. From 1883 to 1901 he was in the employ of the Worthington Pump & Machinery Corp. of New York, first as a salesman in the New York office, and later as European representative of the company, with headquarters in London. He established its business in various countries on the Continent, particularly in Russia, and later became vice-president of the company. When the Alberger Pump & Condenser Co. of New York was organized in 1902, Mr. Moore was made its general sales manager. He was connected with the company until 1918, serving in an advisory capacity during the last six years.

Since 1918 Mr. Moore had been engaged independently on the improvement in design of high grade automobiles and trucks. He had made numerous mechanical inventions, expecially relating to automobiles, and had taken out many patents in England and the United States for vehicle wheels, frames for automobiles, pneumatic tires, radiators, and designs for automobile bodies and chassis. A

number of applications for patents were on file at the Patent Office at the time of his death.

Mr. Moore entered the A.S.M.E. as a junior member in 1885 and became a member a few years later. He withdrew from membership in 1908 and was reinstated in 1912. He is survived by his widow, Mary Anastasia (Tobin) Moore, whom he married in Jersey City, N.J., in 1906.

GUSTAV MORTENSEN

Gustav Mortensen, production and sales engineer for the Giant Heating Company, Los Angeles, Calif., since 1928, died in National City, Calif., on August 3, 1930. Mr. Mortensen was born in Copenhagen, Denmark, October 4, 1895, the son of Frederik L. P. and Nicoline H. (Hansen) Mortensen. He spent four years at De Forenede Kirkeskoler in Copenhagen, and received a B.A. degree in mechanical and electrical engineering from the Teknisk Selskabs Skole of that city in 1917. For two years following his graduation, he was connected as mechanical engineer and draftsman with F. L. Smith & Co. of Copenhagen, engaged in the design of cement mill machinery and general plant layout. During the following year he was employed by Riedel & Lindegaard of that city as mechanical engineer on the design of steam power plants, engines, cranes, and general machinery.

Mr. Mortensen came to the United States late in 1920 and was employed for two years as mechanical draftsman by the Holt Manufacturing Company, Stockton, Calif., manufacturers of caterpillar tractors and power harvesters. During the next two years he was connected with S. G. Fyfe, consulting engineer of Los Angeles, in the design and supervision of boiler, heating, and ventilating plants. For a short time in 1924 he designed marine and stationary steam power plants for the Liberty Boiler Works of Los Angeles. In October of that year he entered the employ of the Pan American Oil Company of Wilmington, Calif., to design oil refinery equipment. After two years with this company he became associated with the Summerbell Truss Company of Los Angeles as structural engineer, and subsequently worked as designer and served as general manager of its Oakland, Calif., office. He resigned this position in 1928 to accept the one which he held at the time of his death.

Mr. Mortensen became an associate member of the A.S.M.E. in 1930. He also belonged to the Ingenior & Konstructor Sammenslutningen of Copenhagen. He was an enthusiastic yachtsman, belonging to an amateur yacht club in Copenhagen for many years and receiving many trophies there, and serving as manager and president of the Viking Athletic Club of Los Angeles from 1924 to 1928. A brother, Karl Mortensen, of Los Angeles, survives him.

JOHN JAMIESON MUIR

John Jamieson Muir, who died on April 7, 1931, at the Providence Hospital in Seattle, Wash., was a native of Ireland. He was born at Waterford on May 4, 1860, the son of Alexander and Marion (Jamieson) Muir. He studied steam engineering and applied mechanics at the College of Fine Arts in Glasgow, and served an apprenticeship from 1875 to 1880 with the Muir and Huston Harbour Engine Works in that city. His first position was as assistant engineer for the Transatlantic State Line on the S.S. *Craigrownie*. In 1882 he entered the drawing room of the Fairfield Works of John Elder & Co., at Govan, Scotland, where he remained for about a year. The next four years were spent as engineer and superintendent with MacNicoll & Co., naval architects of Glasgow. He then went to Calcutta as deputy superintending engineer for the India General Steam Navigation Company. From 1890 to 1892 he was assistant manager for John King & Co., shipbuilders and general contractors, Howrah, Bengal, East India.

Mr. Muir came to the United States in 1893 and for two years was associated with the Fort Wayne Electric Corporation, Fort Wayne, Ind., as engineer and draftsman. For four years beginning in 1896, he was chief draftsman for the Bass Foundry and Machine works of Fort Wayne, and manager for the following year of the National Steel Casting Company of that city. He went to Pittsburgh, Pa., in 1903, and during the next year worked for the Engineering Sun Company of Marcus Hook, Pa., as marine engineer and superintendent. He returned to Fort Wayne in 1905, and after some general engineering work there, again went to Pittsburgh to work for the American Sheet & Tin Plate Co. From 1908 to 1918 he was mechanical superintendent of the Bass Foundry and Machine Works at Fort Wayne, with which he had earlier been connected as chief draftsman. He had been located in Seattle since 1919, for a year as superintendent of the Patterson-MacDonald Shipbuilding Company, after which he retired, except for some consulting work.

Mr. Muir became a member of the A.S.M.E. in 1896. He is survived by three daughters. His wife, Elizabeth (White) Muir, also died in 1931.

AUGUSTUS F. NAGLE

Augustus F. Nagle, who became a member of the A.S.M.E. in 1880, died at the Brooklyn Home for Aged Men, in Brooklyn, N.Y., on December 5, 1920. He entered the Home in June of that year.

HOWARD B. NASH

Howard B. Nash, who died on July 5, 1931, had been chief engineer of the Armour Leather Company, Chicago, Ill., since 1914. He had charge of the design, construction, and maintenance of a large number of industrial plants, as well as the selection and purchase of equipment for them.

Mr. Nash was born at Platte City, Mo., on June 7, 1884, the son of William and Annie Nash. He was graduated from the Manual Training High School at Kansas City, Mo., and studied mechanical engineering through the International Correspondence Schools, day classes at the Chicago School of Technology, and night classes at Armour Institute. He learned the machinist's trade with Swift & Company at Kansas City and the Atchison, Topeka & Santa Fé Railroad at Argentine, Kansas. He worked as a machinist for a year for Armour & Company of Kansas City and for the St. Louis & Santa Fé Railroad, for which he subsequently became roundhouse foreman at Enid, Okla. In 1909 he served as floor foreman at Kansas City for the Kansas City Terminal Railroad. The following year, he went to Indiana Harbor as draftsman for the American Steel Foundry. In 1910 he accepted a similar position at the Armour Glue and Soap Works in Chicago, being promoted from that position to the one which he held at the time of his death.

Mr. Nash became a member of the A.S.M.E. in 1921. Surviving him are his widow, Rose M. (Kill) Nash, whom he married in 1918, and three children, Howard B., Marjorie M., and Catherine A. Nash.

EMIL ARTHUR NELSON

Emil Arthur Nelson, who died on January 10, 1931, at Springfield, Ill., after a brief illness, was born at Reading, Pa., on June 14, 1894. His parents were Adolph and Edna Nelson. He was educated in Sweden, at the Ostersund High School and the Royal Technical University, Stockholm, from which he received his M E. degree in 1921. During the summer before his graduation he was employed as a designer for the Swedish State Railways, with whom he had served an apprenticeship and worked as a machinist at an earlier time. He had also been employed as an estimator at the Stockholm Machine Shop during one vacation.

Mr. Nelson returned to the United States in 1921 and for a year was machinist and acting shop draftsman for the Long Island Railroad. During the next two years he was draftsman for the Babcock & Wilcox Co., at Bayonne, N.J., working in their proposal department on standardization and other developments. He had been with the Springfield Boiler Company as mechanical engineer from 1924 until the last year of his life, during which he was combustion engineer for the Municipal Light and Water Department of Springfield.

Surviving Mr. Nelson are his widow, Caroline M. (Bretscher) Nelson, whom he married in 1926, and an infant daughter. Mr. Nelson had been an associate member of the A.S.M.E. since 1929.

ALMON EVANS NORRIS

Almon Evans Norris, for many years chief engineer for the Mead-Morrison Manufacturing Company, and prior to that mechanical engineer for its predecessor, the Rawson & Morrison Manufacturing Co., died on January 15, 1931, at his home in Brookline, Mass.

Mr. Norris was born at Orford, N.H., on May 29, 1866, the son of John Langdon and Martha (Evans) Norris. He received an S.B. degree in mechanical engineering from the Massachusetts Institute of Technology in 1890. Following his graduation he spent several years in the drawing-room of Edward Kendall & Sons, Cambridgeport, Mass., with whom he had secured his first experience as a machinist before entering M.I.T. This company were builders of hoisting engines. He was later associated with the Leavett Engineering Company for about three years.

Mr. Norris also had some early work in the design of telescope mountings for Alvan Clark & Sons, including the Bruce photographic telescope for Harvard University and a 24-in. telescope for Lowell's Observatory at Flagstaff, Ariz. The telescope mountings for Harvard University were built at the factory of Rawson & Morrison.

The Rawson & Morrison Manufacturing Co., with which Mr. Norris became associated in 1896, manufactured hoisting engines and machinery and automatic coal handling machinery. It was located at Cambridgeport at that time. The company was reorganized in 1904 as the Mead-Morrison Manufacturing Company and

moved to East Boston in 1913. Mr. Norris was their chief engineer from the date of their organization until the time of his death. He was responsible for the machinery manufactured by them, and was considered a peer in their field of specialized engineering.

Mr. Norris became a member of the A.S.M.E. in 1900. Surviving him are his second wife, Maud F. (Conrad) Norris, whom he married in 1905, and five daughters and a son.

WILLIAM TULLY NORTON, JR.

William Tully Norton, Jr., died on April 28, 1931, at Lakewood, Ohio. He was born on October 20, 1880, in Cleveland, Ohio, the son of William Tully and Harriet Reid (Haller) Norton. He attended the University School and Case School of Applied Science, both of Cleveland. Prior to this graduation he worked as a draftsman for the General Automobile Company in Cleveland, securing there his first experience in the automotive industry. After completing his work at the Case School in 1903 he was employed for a time by the Cleveland Telephone Company. He then was engaged by Howard E. Coffin of the Olds Motor Works in Detroit, Mich., as experimental engineer and for two years worked on the design and testing of motors. During the next two years he was successively assistant superintendent of the Phoenix Truck Company, Detroit, designer for the Aero Car Company, Detroit, and superintendent of the Kansas City Motor Car Company, Kansas City, Mo. From 1908 to 1910 he was chief inspector for the Chalmers Motor Company, of Detroit. He organized the inspection department of the company and had supervision of the testing and chemical laboratories, controlling inspection of raw material, finished machine parts, and completed cars. After leaving the Chalmers Motor Company he did some experimental work for James Cunningham, Son & Co., Rochester, N.Y., and then, in 1911, became chief engineer and general superintendent of the Selden Motor Vehicle Company, Rochester, N.Y. He was connected with this company until 1918, designing trucks and having direct supervision over producing methods and inspection.

Mr. Norton spent several months of 1916 in England and France checking the performance of trucks operating under war conditions. He served in 1916 and 1917 as a member of the Military Truck Committee sponsored by the Society of Automotive Engineers and the National Automobile Chamber of Commerce and cooperating with the Government in the standardization of army trucks. In October, 1917, he supervised the construction in seven days at the Selden factory of a sample Class B or Liberty truck and himself drove it to Washington for delivery and demonstration. In February, 1918, he was sent to France as civilian technical advisor to the chief of the Motor Transport Corps, A.E.F.

From January to June, 1919, Mr. Norton was chief engineer of the Russell Motor Axle Company, of Detroit, Mich., designing a new line of gear driven truck axles, and supervising laboratory and road tests of materials and axles. From June to October of that year he held a similar position with the Republic Motor Truck Company of Alma, Mich. He was then called to Washington as assistant chief engineer of the Motor Transport Corps, subsequently being stationed at Camp Holabird, Md., and Lakewood, Ohio. Since his retirement from that position in 1922 he had been engaged as chief engineer for the L. M. Axle Co., Cleveland, and the Sarasin Six-Wheel Truck Company, Haverhill, Mass., and for the past five years had done consulting work, mostly of a private character for large corporations and banking institutions.

Mr. Norton became a member of the A.S.M.E. in 1920. He also belonged to the Society of Automotive Engineers and to the Masonic, Beta Theta Pi, and Theta Nu Epsilon fraternities. He is survived by his widow, Mildred French (Williams) Norton, whom he married in 1905, and by his mother.

GEORGE MILTON OGLE

George Milton Ogle, who died on October 14, 1931, was born at Louisville, Ky., on February 23, 1891, the son of George J. and Nellie (Lane) Ogle. He was graduated from Vanderbilt University with a B.E. degree in 1912. His college vacations were spent with the Louisville & Nashville R.R., the American Car & Foundry Co., and the Chicago, Rock Island & Pacific Ry. Co. Following his graduation, he spent a year as draftsman in the signal department of the Pennsylvania Railroad at Pittsburgh, Pa. During the next four years he was connected first as chief draftsman and subsequently as superintendent of machinery with the Heywood Bros. & Wakefield Co., Wakefield, Mass. He next spent two years with the Edison Electric Illuminating Company in Boston, as commercial engineer, and a year with the Massachusetts Chocolate Company as mechanical and electrical superintendent. In 1918 he was identified with the Westinghouse Electric & Manufacturing Co., East Pittsburgh, for which he handled rubber, paper, and cement mill electrifications. In 1919 he was chief electrical engineer, division of construction, Northern Atlantic District, U.S. Shipping Board. His duties included the layout of power plants, air compresso plants, etc., necessary for ship construction.

From 1920 to 1922, Mr. Ogle was chief electrical engineer for the Vulcan Iron Works, Jersey City, N.J., specializing on the applications of electrical driven auxiliaries for merchant marine vessels. Under his supervision, the U.S. Shipping Board's S.S. *Eclipse* was electrified. For two years, beginning in 1923, Mr. Ogle was power specialist for the General Electric Company in the New York district, advising salesmen in regard to the proper application of power equipment and economic operation of power plants. From 1925 to 1928 he was consulting power engineer for the General Engineering & Management Corp., New York, N.Y. Since that time he had been manager of the power and industrial department of the National Electric Power Company, New York.

Mr. Ogle joined the A.S.M.E. as a junior in 1921 and attained full membership in 1927. He also belonged to the American Institute of Electrical Engineers and the Society of Terminal Engineers, and he had contributed a number of articles to the technical press. He is survived by his widow, Grace (Kiernan) Ogle, whom he married in 1916, by one son, George M. Ogle, Jr., and by his mother.

WILLIAM POOL PARKER

William Pool Parker, whose death occurred on June 6, 1931, was born at Owosso, Michigan, on May 8, 1871. His parents were Henry Warner and Agnes (Nicholson) Parker. Mr. Parker was graduated from the University of Michigan with a B.S. degree in civil engineering in 1894. During the next six years he worked on the design of bridges for the Keystone Bridge Works of the American Bridge Company and for the Pittsburgh Bridge Company, both of Pittsburgh, Pa., and also for the Massillon Bridge Company of Massillon, Ohio. He obtained experience in the design of foundry equipment with the Whiting Foundry Equipment Company of Harvey, Ill., and in heating and ventilating with the Detroit Heating & Lighting Company of Detroit, Mich. For a time, he was located in the chief engineer's office of the Cambria Steel Company of Johnstown, Pa. Here, he worked on the design of all classes of steel structures.

From 1902 until 1912, Mr. Parker was chief engineer for the A. M. Blodgett Construction Company, Kansas City, Mo. During the last three years, he was in direct charge of the construction of the Galveston Causeway. As a member of the firm of the Concrete Steel Engineering Company of New York, Mr. Parker was connected with the construction of the reinforced-concrete arch bridge over the Mississippi at Minneapolis and the bridge over the Hudson River at Glens Falls, New York.

For five years, beginning in 1915, Mr. Parker was associated with the Keystone State Construction Company, representing the Underpinning & Foundation Co. of New York. He was construction engineer for the City Hall Subway in Philadelphia. After the completion of this work, he became general manager for Ballinger & Perrot, later known as The Ballinger Company. Since 1924 he had engaged in general practice as a consulting structural engineer in Philadelphia.

Mr. Parker became a member of the A.S.M.E. in 1921. He was an honorary member of Tau Beta Pi fraternity, and also belonged to the American Society of Civil Engineers and the Engineering Institute of Canada, as well as to a number of clubs in the vicinity of Philadelphia. He was a former vice-president of the Engineers Club of that city. Mr. Parker served with the Engineers Corps during the Spanish-American War.

Surviving Mr. Parker are his widow, Emilie Whittaker (Evans) Parker, whom he married in Pittsburgh in 1899, and a son and three daughters, Henry Warner Parker, Mrs. Anthony V. Pickard, Mrs. Herman S. Thoenebe, and Emilie Evans Parker.

SIR CHARLES ALGERNON PARSONS

Pioneer in steam-turbine engineering, honorary member of many engineering societies, including The American Society of Mechanical Engineers, and the first engineer to receive the Order of Merit, the Hon. Sir Charles Algernon Parsons died February 11, 1931, while on a cruise to the West Indies. He was 76 years old, and through his death engineering science and industry lost a leader and pioneer. More than any other person, by his development of the steam turbine, he contributed to the greatest advance in steam practice since the advent of the steam engine. He was not the first to conceive of a rotary steam prime mover, but due to his labor and ingenuity the

steam turbine became the prime mover for the generation of electricity in the largest power stations, and for the propulsion of the largest and fastest ships. He had a very broad knowledge of scientific activities, was very sympathetic in his nature, and a most agreeable and interesting companion.

The scientific bent in Sir Charles was in some measure hereditary, and was nurtured under his distinguished father, the Earl of Rosse, of astronomical fame. From his birth, on July 13, 1854, he lived in a scientific environment, for Birr Castle was an open house to men of high achievement. Sir Charles learned the principles of mechanical construction from his father, and also had a succession of distinguished tutors. After a short period at Dublin University he entered St. John's College, Cambridge, at the age of 19. His three-year course terminated with the winning of high honors as eleventh wrangler in the Mathematical Tripos of 1876. He next became pupil apprentice at Armstrong's works at Elswick, where he continued for four years. His brilliance as a scientist and technician thus had a practical foundation, and he attained mechanical skill of a high order. Later in the development of his inventions he was capable of taking the tools from a workman's hand and showing him how to do the job. As an apprentice he invented several things, among which were valve motions and a four-cylinder epicycloidal rotary engine with revolving cylinders which were then manufactured by Messrs. Kitson, of Leeds. It then followed naturally that with the termination of his apprenticeship he should join the latter firm, where he continued for two years, his chief work being on the propulsion of torpedoes by rockets.

In 1883 Sir Charles became a partner in the firm of Clarke, Chapman, and Company, and shortly afterward began experimenting with the steam turbine, about one year after it had been invented by de Laval of Sweden. Sir Charles devised the scheme of compounding the blading, and developed the reaction turbine in which steam flows through a series of alternate stationary and moving blades of similar form. By this arrangement he was able to reduce the speed of the rotor so that higher efficiencies could be obtained at speeds much more practical than had been possible in the de Laval turbines. It was due to this invention and to his subsequent activities that the steam turbine became of such commercial importance and the building of turbines in large sizes was made possible.

The turbine blades at first were milled out of the solid metal on the periphery of brass rings, a construction subsequently superseded by one more perfect, in which the blades were set and wedged with distance pieces in rings with their roots in grooves on the drum and on the casing, as is now a familiar construction. At that time dynamo construction was in its infancy and the speeds necessary had a great influence on the design of the steam turbine and added to the problems that had to be solved. Sir Charles believed that there was a great future for the supply of electricity, and he knew that its generation was then limited and ever would remain so unless the limitations of the reciprocating engine were overcome. He had in mind a prime mover of small dimensions, naturally balanced, which could run at the speeds necessary for efficient dynamos. It was the combination of steam turbine with dynamo that led to his success, and although the first turbine was less efficient than the reciprocating engine, its particular qualities for dynamo driving compensated for the loss. The steam turbine has now entered the entire field of the steam engine, and is the parent of a large family of high-speed devices designed particularly to take advantage of the characteristics which it affords.

The first Parsons steam turbine was completed in 1883. It was a 4-kw. machine with a series of rotating wheels, each complete in itself and corresponding to a parallel-flow water turbine. The mechanical difficulties were great, but they were overcome. The steam consumption of the first turbine was about 200 lb. per kw-hr., with a pressure of only 60 lb. per sq. in. This type of machine continued to be built for six years in small sizes for running dynamos, the largest machine being of only 100 hp.

When he withdrew from partnership in the firm of Clarke, Chapman, Parsons, and Company in 1889, Sir Charles released his patent rights. He then established his own turbine and dynamo works at Heaton-on-Tyne, and there continued for five years experiments toward producing a radial-flow turbine, although recognizing that his original invention, based on a parallel-flow system, was preferable. In 1892 he constructed a radial-flow turbine of 100-kw. output which consumed but 27 lb. of steam per kw-hr. In 1894 he purchased his original patent rights, and from then on advance was rapid.

By 1895 steam-turbine developments had attracted considerable interest in the United States, and a license agreement for building Parsons steam turbines for land purposes was entered into by the Westinghouse Company. About this time the principal shipbuilders also secured a license for turbines applicable to marine propulsion. In 1905 the Allis-Chalmers Company also secured a license. These licenses have been renewed from time to time, and the companies have continued to enjoy the advantages of technical information from Sir Charles's organization.

In 1913 Sir Charles furnished a 25,000-kw. tandem-compound turbine generator and condenser for the Fisk Street Station of the Commonwealth Edison Company, concerning the efficiency of which unprecedented guarantees were made. This turbine set a new standard of performance in this country. In the estimation of the purchaser, this machine has a standard of reliability to be compared favorably with any other on this system. In 1924 a second turbine was furnished the Commonwealth Edison Company, of 50,000-kw. capacity, of unique design in the low-pressure range of expansion. The object of this was to secure low leaving loss. The machine has operated excellently. Both of these turbine units were the largest of their time.

In 1894 Sir Charles devoted particular attention to the application of the turbine to the propulsion of ships. Owing to the necessarily high speed of rotation, difficult problems connected with propeller efficiencies were at once encountered, and it was not until after three years of experimental investigation that the famous *Turbinia* was completed. The speed performance of the *Turbinia* at the Victorian Diamond Jubilee Naval Review in 1897 attracted great attention to the system, and so rapid was the conversion of technical and public opinion in favor of the turbine for marine propulsion that by 1917 Atlantic liners with turbines of 80,000 hp. were in use in the merchant service.

Concurrently with improvement in the land turbines, corresponding advances in the marine turbine were made, so that today the coal consumption has been reduced to less than 1 lb. per shaft hp. per hour. The turbine is now universally adopted for the propulsion of warships and all high-speed vessels in the merchant service use the same prime mover. A development invaluable to oceanic transport service was Parsons' introduction of the helical gear for reducing the relative speed of the turbine to that of the propeller, so that the turbine could run at the maximum rated speed required for high thermodynamic efficiency, while the propeller revolved at the low speed conducive to propulsive efficiency. This latter invention is almost as great in its commercial results as the invention of the turbine itself.

Sir Charles did not restrict his exceptional knowledge and insight to the solution of problems of immediate utilitarian value. He studied the behavior of different bodies, particularly carbon under high pressure, and succeeded in converting this substance into graphite. Another important investigation which he carried out in conjunction with S. S. Cook, was the determination of the cause of erosion of high-speed screw propellers. This showed that a very important part in this phenomenon was played by water hammer. The main research on this subject was dealt with in a paper read before the Institution of Naval Architects in 1929.

Sir Charles showed a great interest in optics in later years. He took over the Derby Crown Glass Works and interested himself in constructing searchlights and their mirrors, and in solving other problems in connection with light. Later with Sir Howard Grubb he formed a company to make astronomical instruments and telescopes.

Perhaps his most important contribution to engineering development was to show that by employing higher steam pressures and temperatures in marine work it was possible to obtain higher efficiencies and a proportional economy in coal consumption. This demonstration was made in 1926 and took the form of equipping the Clyde pleasure steamer, *King George V*, with boilers supplying steam at a pressure of 550 lb. per sq. in. and a temperature of 800 deg. fahr. to a 3500-hp. turbine. The experiment was in the main successful, and the correctness of the ideas has been proved by the results obtained on the most recent vessels of the Canadian Pacific Company.

Sir Charles never lost his interest in steam turbines, and reviewed the progress that has been made recently in their design in two important papers which he read before the British Association in 1929 and at the World Power Conference in Berlin in 1930. In these he pointed out that in addition to an increase in the steam pressure and temperature, higher economy was to be sought in the use of regenerative heating, both of the feedwater and of the steam after partial expansion. He added that these developments, accompanied by the demand for larger and larger units, emphasized the value of a more economical utilization of materials used and led to the increasing employment of higher quality steel, multiple exhausts, and important modifications in the design of the condensing equipment. These changes are reflected in the design of more recent units for which Sir Charles was responsible, one of the most noticeable of which is the 40,000-kw. set installed in the Brimsdown station of the North Metropolitan Electric Power Company.

Sir Charles was a frequent contributor to the proceedings of the leading engineering societies, by which his preeminence as a scientist and pioneer was fittingly recognized in the course of his long connection with them. He was elected a Fellow of the Royal

Society as long ago as 1898, and was awarded the Rumford Medal in 1902 and the Copley Medal in 1928, being the first engineer to attain this high distinction. He became a member of the Institution of Civil Engineers in 1892, and served on its Council and that of the Institution of Mechanical Engineers. He was made an honorary member of The American Society of Mechanical Engineers in 1920, and of the Institution of Mechanical Engineers in 1925. He joined the Institution of Electrical Engineers in 1888, and was elected an honorary member in 1909, being awarded both the Franklin and the Faraday Medals of that Society. The three British institutions conferred the Kelvin Medal upon him in 1926. A further technical honor was the Bessemer Medal of the Iron and Steel Institute, which was awarded him in 1929. He was an honorary member of the Société des Ingénieurs Civils de France, and a member of the Verein deutscher Ingenieure. He was vice-president of the Institution of Naval Architects and president in 1913 and 1914 of the North East Coast Institution of Engineers and Shipbuilders. In 1904 he was president of Section G (Engineering) of the British Association, and in 1917 became president of the whole body. He also served as president of the Institute of Physics, the Institute of Marine Engineers, the Tyneside Unionist Association, and the Literary and Philosophical Society.

The Liverpool University conferred the degree of Doctor of Science on Sir Charles in 1909. He was also awarded similar honors by the Universities of Glasgow, Durham, Toronto, and Pennsylvania. He was made a C.B in 1904, a K.C.B. in 1911, and became a member of the Order of Merit in 1927.

During the World War he served on a number of committees, and was an original member of the Advisory Council of the Department of Scientific and Industrial Research and of the Fuel Research Board. Sir Charles threw himself with characteristic wholeheartedness into this work; he was tireless in suggesting new methods, in giving opportunities for experiments on a large scale, and in helping by his advice and criticism those who were carrying out investigations for the Board. He was a very agreeable as well as most efficient colleague, and through the questions before the Committee were such as to give room for wide differences of opinion, there was no friction and the proceedings were harmonious. In 1883 he married Katherine Bethell, who is a founder of the Women's Engineering Society. Their only son was killed in action in 1918. Their only daughter passed through the Engineering School at Cambridge, and served in the Training Department of the Ministry of Munitions.

Sir Charles was a generous benefactor of science, having given large amounts to the Royal Institution and to the British Association. He was one of the governors of the Dominion Students' Hall Trust, the object of which is the building of a hall of residence, London House, for Dominion, Colonial, and British students who come to the University and schools of London. In addition to a generous contribution, he gave much time to interesting the engineering profession in the scheme.

At the time of his death Sir Charles and Lady Parsons were on a pleasure trip on board the steamer *Duchess of Richmond*. After an illness of four days he died shortly before the vessel reached Kingston, Jamaica. His body was taken back to England on the first available boat. A memorial service was held in Westminster Abbey on March 3, attended by relatives, friends, and representatives of scientific and engineering societies. The American Society of Mechanical Engineers was represented. A memorial service was also held at St. Nicholas Cathedral, New-Castle on-Tyne, the same day, and interment was at Kirkwhelpington Churchyard.—C.W.S.

WORTHINGTON ELMORE PLATT

Worthington Elmore Platt, who died at Norwich, Conn., on August 7, 1931, was born in New Haven on April 15, 1903, the son of Elmore Wooster and Helena (Lowe) Platt. His early education was secured in the schools of New Haven, and he was graduated from Lehigh University, Bethlehem, Pa., in 1926, with a mechanical engineering degree.

For the next four years Mr. Platt was cadet engineer with the Consolidated Gas Company of New York, serving in each branch of the company. Since then he had been employed as superintendent of the Customer's Service Department of the East River Gas Company, Long Island City, N.Y. He perfected a cam used by that company.

Mr. Platt belonged to the Gas Engineering Society of New York and had been a junior member of the A.S.M.E. since 1926. He also belonged to the Masonic fraternity and was interested in boys' clubs and had served as camp councilor for the Y.M.C.A. Surviving him are his widow, Lois Stone (Caswell) Platt, whom he married in 1928, and one son, Charles Elmore Platt.

PAUL ALBERT POPPENHUSEN

Paul Albert Poppenhusen, who died on May 24, 1930, was born at College Point, L.I., N.Y., on November 4, 1873. His parents were Herman C. and Caroline S. (Funke) Poppenhusen. He received instruction from a private tutor until he was ten years old, when he was sent to boarding school at Kiel, Germany. After returning to this country he spent a year at Northwestern University, Evanston, Ill.

His first position was in the wholesale drygoods business and subsequently he was connected with the Bismarck Hotel Company of Chicago. In 1897 he organized the Green Engineering Company, East Chicago, Ind. for manufacturing Green chain-grate stokers, ash-handling equipment, and cast-iron storage tanks. He served as president of the company from 1906 to 1924. Since 1926 he had been a member of the firm of Acers-Poppenhusen, Chicago, Ill., dealing in power-plant equipment.

Mr. Poppenhusen belonged to the Western Society of Engineers, the Masonic fraternity, and a number of clubs in New York and Chicago. He became a member of the A.S.M.E. in 1912. He was interested in big game hunting and belonged to the Caughuawana Fish and Game Club of Quebec. He is survived by his widow, Helen Burk (Durham) Poppenhusen.

EDWARD BORIS POULSEN

Edward Boris Poulsen, whose death occurred on March 9, 1931, following an operation for appendicitis, was born in Oberpalen, Russia, on May 12, 1900. His technical education was secured at the Montana State College of Agriculture and Mechanic Arts, Bozeman, Mont., which conferred upon him the degree of B.S. in M.E. in 1923.

Following his graduation Mr. Poulsen accepted a position as draftsman for the Wheeling Steel Corporation at its Steubenville plant, Steubenville, Ohio. After two years there he went to Gary, Ind., as draftsman for the Gary Tube Company. In 1928 he became draftsman for the Michigan Copper & Brass Co., now known as the Michigan Division of the Revere Copper & Brass Co., at Detroit. He remained with this company until his death.

Mr. Poulsen became a junior member of the A.S.M.E. in 1924. He was a member of the Masonic fraternity and a Lutheran by faith. He had been a member of the Reserve Officers Training Corps at Montana State College.

Mr. Poulsen is survived by his widow, Katherine (Camp) Poulsen, whom he married in 1930.

FREDERICK RANDOLPH PRATT

Frederick Randolph Pratt, chief engineer in the Steam and Electric Department of the Hollingsworth & Vose Co., East Walpole, Mass., died on April 3, 1931. He was a native of Boston, where he was born on December 25, 1876. After completing high school he was employed as a machinist by the textile industry in New England until 1908. Among the companies by which he was engaged were the Crompton Company, Gregorys Woolen Company, E. H. Smith and C. L. Blanding mills in Rhode Island, Warrenton Woolen Company and Old Mystic Woolen Company in Connecticut, and Grant Yarn Company, in Fitchburg, Mass. He became master mechanic for the last-named company. From 1904 to 1908 Mr. Pratt also did some turbine assembly and road work for the General Electric Company, in West Lynn, Mass.

In 1908 Mr. Pratt accepted a position as master mechanic and chief engineer for the Shetucket Company, Norwich, Conn. The following year he went to Adams, Mass., to work in the same capacity for the Berkshire Manufacturing Company, with which he was connected until 1917. He was in East Walpole in 1917 as mechanical engineer for Bird & Son, in Pawtucket in 1918 and 1919 as mechanical superintendent for D. Goff & Sons, and in Providence in 1920 with B.B. & R. Knight. He next went to Shanghai, China, for the Whiting Machine Works. After returning to this country he became chief engineer for the St. Croix Paper Company, Woodland, Maine, where he remained until 1930. He had been with the Hollingsworth & Vose Co. since May 12, 1930.

Mr. Pratt became a member of the A.S.M.E. in 1914.

CHARLES JOHN PRENDERGAST

Charles John Prendergast, manager of the St. Louis office of the Bailey Meter Company, died on November 21, 1931, from injuries sustained the preceding day in a building collapse at Paoli, Ind.

Mr. Prendergast was born at North East, Pa., on May 14, 1903. He attended the high school there and was graduated from the

Case School of Applied Science in 1925 with a B.S. degree in mechanical engineering. During his summer vacations, while in college, he secured practical experience with several companies, among them the Bailey Meter Company of Cleveland. Following his graduation, he returned to the employ of this company as Cadet Engineer, Senior Division. Before being sent to the St. Louis office in 1930, he had served as sales representative in the Boston and Springfield, Mass., territories.

Mr. Prendergast became a junior member of the A.S.M.E. in 1925. He is survived by his widow, Thelma E. (Luce) Prendergast, whom he married in 1928, and by his mother, Mrs. Florence Prendergast.

ROBERT WESTALL PRYOR, JR.

Robert Westall Pryor, Jr., a member of the firm of Koithan and Pryor, New York, N.Y., died on June 10, 1931, at his home in Newark, N.J., after a six months' illness.

Mr. Pryor was born in Newark on July 31, 1881, and attended the public schools there. He received an M.E. degree from the Stevens Institute of Technology in 1902. He immediately took a position as sales engineer with the Buffalo Forge Company of Buffalo, N.Y., and after a period of intensive training was assigned to the branch office in New York. He engaged in many phases of paper mill work and in heating, ventilating, drying, and kindred lines, and was in charge of installation in many parts of the United States and Canada. His work involved a considerable amount of designing to effect the adaptation of equipment to existing plants.

In 1909 he joined in forming the partnership of Koithan and Pryor, sales engineers and representatives in charge of the New York Branch of the Buffalo Forge Company, the Carrier Air Conditioning Company, and the Buffalo Pump Company.

He became a junior member of the A.S.M.E. in 1907 and a member in 1913. He also belonged to American Society of Heating and Ventilating Engineers and to several clubs.

He is survived by his widow, Lillie (Keepers) Pryor, and by one daughter.

WILLIAM WOOD RICKER

William Wood Ricker, who died on January 27, 1931, was born at Portsmouth, Ohio, on August 22, 1873. His parents were James Wood and Louisiana (Moore) Ricker. At the age of seventeen he became an apprentice in the machine shop of the Portsmouth Foundry & Machine Works, where he remained until he entered Cornell University in 1892. He secured further experience there and with the Burgess Iron & Steel Co. during his college vacations, and following the receipt of his M.E. degree in 1896 went into the machine shop of the Brown Hoisting Machine Company, at Cleveland, Ohio. During 1898 he was a draftsman for this company and in 1899 helped to build a cantilever crane at Wm. Cramp & Sons, Philadelphia, and had charge of the construction of a coal handling plant at the New London Navy Yard.

After the completion of this work Mr. Ricker went to Honolulu, where he remained for about three years, engaging in the importation of plantation and railway supplies and the design and installation of special machinery and cranes. He then returned to the employ of the Brown Hoisting Machine Company as salesman and during his two-year connection in this capacity made two trips to Cuba for the company.

In 1904 the Niles Tool Works, Hamilton, Ohio, engaged Mr. Ricker as superintendent, and early in the following year, he was appointed a salesman for the Niles-Bement-Pond Company, with headquarters in New York, N.Y. In 1908 he became secretary and treasurer of the Guarantee Construction Company, New York, and retained the office of treasurer until his retirement at the close of 1929.

When the United States entered the World War Mr. Ricker was commissioned a major in the Ordnance Department and was stationed at Washington for some months. He was then sent to France, and subsequently promoted to the rank of lieutenant-colonel and served in France, Italy, and England, also being sent on a mission to Denmark.

Mr. Ricker became a member of the A.S.M.E. in 1906. He was also a member of the University, Cornell, and the Engineers' Clubs in New York, the Apawamis Club of Rye, N.Y., and several other clubs in Havana, Cleveland, and Washington, including the Army and Navy Club in Washington. He is survived by his widow, Abby Putnam (Morrison) Ricker, whom he married in 1925.

WINFIELD SCOTT ROGERS

Winfield Scott Rogers, pioneer of the ball bearing industry, died on February 17, 1931, at his home in South Orange, N.J., where he had lived since his retirement from business in 1929.

Mr. Rogers was born at Waynesburg, Pa., on July 21, 1853. While still attending public school he had his first experience in the operation of engines as a night employee of the Union Stock Yards at Columbus, Ohio, and subsequently he served a three-year apprenticeship as a machinist and draftsman with the C., H.V. & T. R.R. With this training he secured a position on the Columbus and Chicago Division of the P.C. & St. L. R.R. and was with this road and the Chicago Division of the Baltimore & Ohio R.R. for a number of years, working as fireman, brakeman, conductor, and engineer. He then withdrew from railroad service, accepting a position as diemaker and foreman of the sewing machine room of the Longstreth & Ayer Mfg. Co., Columbus. After a year with this company he spent about two years as superintendent of the Columbus Basket Company, three as draftsman, salesman, and general mechanical engineer for P.G. March & Co., Cincinnati, builders of machine tools, and some months as draftsman for the Black & Clawson Co., paper mill machinery manufacturers of Hamilton, Ohio.

At the time Mr. Rogers became a member of the A.S.M.E. in 1888 he was draftsman and assistant to the general manager of the Universal Radial Drill Company, and senior member of the firm of Jones and Rogers, in Cincinnati. The following year he was appointed mechanical engineer at the Watervliet (New York) Arsenal. In 1890 and 1891 he was superintendent of the estate of F. W. Richardson in Troy, N.Y., and then he was employed until the latter part of 1892 as superintendent and mechanical engineer for H. G. Hammett, of Troy. Subsequent to a year of general mechanical engineering practice he became engineer and airbrake instructor for the Delaware & Hudson Canal Co., at Green Island, N.Y. In 1895 he had an office in Buffalo and in 1896 in Philadelphia. During the next few years he was engaged in the manufacture of laundry machinery, first as superintendent of the American Laundry Machinery Company, Cincinnati, and in 1898 as a member of the firm of Rogers & Co., New York.

Mr. Rogers entered the ball bearing field in 1898 as an associate of Henry M. Whitney in The Ball Bearing Company, Boston. He was treasurer and general manager of the company until 1901. During the latter part of that year he served as vice-president and general manager of The Steamobile Company of America, Keene, N.H., and in 1902 was also connected with the Roller Bearing Equipment Company of that place. He then went to Bantam, Conn., as general manager of The Bantam Manufacturing Company. In 1903 he was appointed receiver for the company which he reorganized in 1905 as the Bantam Anti-Friction Company. He remained president and active head of the company until its reorganization in 1921 under the name of The Bantam Ball Bearing Company, of whose Board of Directors he was made chairman. The company relocated in South Bend, Ind., in 1928.

It was largely the result of the early pioneering efforts of Mr. Rogers that the value of anti-friction ball bearings became recognized in the mechanical world. His was the first commercial catalog of ball bearings published in this country. Among others of his works were the following: "Primer of the Air Brake," "Time and Cost Keeping," "Sketches of an Apprenticeship," "The Romance of Anti-Friction," a history of the ball bearing industry, completed shortly before his death, and "Oil Heating the Home," giving the results of five years of tests with various burners.

During his residence in Connecticut Mr. Rogers served two terms in the state legislature and was active in all movements for the benefit of his community. He was a former member of the Society of Automotive Engineers. Surviving him are his widow, Nellie S. Rogers, and two daughters.

JOHN L. ROSS

John L. Ross died on April 23, 1930, at his home in Roselle Park, N.J., of heart trouble. Mr. Ross was born in Detroit, Mich., on April 17, 1878, the son of Edward Holland and Susanna (Roche) Ross. He spent two years at Detroit College and then served an apprenticeship from 1897 to 1903 with the Detroit United Railroad, learning the machinist's, patternmaking, and electrical trades. In 1903 he established the Ross & Young Machine Co. in Detroit for the manufacture of automobile and marine internal-combustion engines. In its early years the firm built the Walker two-cylinder opposed motor, a line of four- and six-cylinder overhead valve and cam auto type marine motors, and the Demot two-cylinder opposed motors, transmissions, rear axles, and complete cars. Later a pattern shop and foundry were built, making it possible to produce the necessary castings, and the company manufactured four-cylinder motors, transmissions, and complete Paige "25" cars for the Paige-Detroit Motor Car Company. In 1915 the Ross Automobile Company was organized to build an eight-cylinder V-type motor and car known as the "Ross Eight." This plant was sold in 1917.

Mr. Ross then accepted a position with the Wright-Martin Aircraft Corporation at New Brunswick, N.J. He was the representative of the factory manager to outside manufacturers for the Wright-Martin Aircraft Corporation, having full charge of blueprints and specifications, tools, and materials for them. For a time after the War Mr. Ross was located in the mechanical department of the Kelly Press Division of the American Type Founders Company, Elmora, N.J. He established his own business as a consulting engineer in New York in 1925.

Mr. Ross had been a member of the A.S.M.E. since 1912. He is survived by his widow, Ruth (Porter) Ross, whom he married in 1917.

CHARLES M. ROSWELL

Charles M. Roswell, mechanic power and electrical engineer for the Western Electric Company, New York, N.Y., was born at Sparta, Ky., on December 3, 1884, the son of William Thomas and Mary E. (Rice) Roswell. He received a B.M.E. degree from the University of Kentucky, Lexington, Ky., in 1908 and an E.E. degree from the same institution three years later. The electrical work which led to the latter degree was done with the Jackson Elec. & Hydraulic Mfg. Co., at Jackson, Ky., where he was located in 1908 and 1909, and with the General Electric Company at Schenectady, N.Y., where he completed the regular student test course.

Mr. Roswell remained with the General Electric until 1918. He tested large vertical and horizontal turbines and worked on design in the direct current and central station departments, serving for four years as assistant head of the latter. In 1918 he went to Bluefield, W. Va., to supervise the operation of two water power plants and three steam plants, totaling 40,000 kw., for the Appalachian Power Company. Later he had charge of the design and construction of a new substation and made a layout for a new relay system covering the entire operation of the plant. From Bluefield he went to Charleston, S.C., in 1920, as superintendent of the electrical department of the Charleston Consolidated Railway & Lighting Co. He was in charge of a 10,000-kw. steam power plant and the distribution and street lighting system and designed transformer stations and line extensions.

Since 1921 Mr. Roswell had been connected for short periods each with a number of companies. He designed and estimated the cost of electrical stations and substations for the subsidiary properties of the United Gas Improvement Company, Philadelphia, Pa., supervised the testing and inspection of a new 20,000-kw. steam power plant at Hunlock Creek, Pa., for the Luzerne County Gas & Elec. Co., and made steam heating surveys in Philadelphia for Day & Zimmermann, Inc. He engaged for a time in sales engineering, buying and selling turbines, boilers, transformers, motors, and engineering supplies, and also did sales work for the McClave-Brooks Company, selling grates and stokers in the Philadelphia and Scranton territories, and for the D. B. Wilson Co., of Philadelphia, selling refractories and powdered fuel equipment. For about a year he was connected with Louis T. Klauder, consulting engineer of Philadelphia, for whom he checked the electrical drawings for the new Princeton University chemical laboratory and made surveys and wrote specifications for other construction jobs. He made inspections of machinery and did survey and appraisal work for the Scofield Engineering Company, Philadelphia, the largest assignment of which was the appraisal of the John Lang Paper Company, in that city. Prior to his connection with the Western Electric Company he was located at Scranton, Pa., as mechanical engineer for the new Masonic Temple and Scottish Rite Cathedral built by the Central Construction Corporation, of Harrisburg, Pa.

Mr. Roswell died at the Fairmount Private Hospital in Jersey City, N.J., on July 2, 1931. He is survived by his mother, living in Carrollton, Ky., and by a son, Charles Alfred Roswell. His wife was Miss Emma Baker of Lexington, Ky. He had been a member of the A.S.M.E. since 1928 and belonged also to the American Institute of Electrical Engineers.

ETHELBERT A. RUSDEN

Ethelbert A. Rusden, president of The Textile-Finishing Machinery Company, Providence, R.I., died on February 16, 1931. Mr. Rusden, who was born in Bowden, Cheshire, England, on September 5, 1859, was prominent in the industry not only in this country, but in a number of foreign countries, where he spent some time before coming to the United States.

At the age of fifteen, Mr. Rusden entered the service of the firm of Mather and Platt, at the Salford (England) Iron Works. He was connected with this company for seventeen years, five of which he spent in Russia as engineer at the Schlusselberg Print Works. At the age of twenty-one he was commissioned to manage the firm's exhibit of machinery at the Paris Exposition, and for his services was awarded the gold medal of the company. He was also sent to China and Japan to supervise the construction and equipment of manufacturing plants and made connections in those countries which he maintained throughout the succeeding years.

In 1887 Mr. Rusden came to the United States and established the Rusden Machinery Company at Warren, R.I. This firm was later merged with others to form The Textile Finishing Machinery Company of Providence, manufacturers of bleaching, dyeing, and printing machinery. Mr. Rusden was its general manager until 1910, and had been president since then.

Mr. Rusden was for years a prominent figure in the business life of Rhode Island. He had a notable career in the textile industries of America and foreign countries. A mechanical engineer of high attainments, he was regarded as an authority in this industry throughout the world.

Mr. Rusden became a member of the A.S.M.E. in 1920. He also belonged to the Providence Engineering Society, the Ancient and Honorable Artillery Company of Boston, the Masonic fraternity, and a number of clubs, including the Engineers' Club of New York. A man of unusual culture and refinement, he was a patron of the arts, a great lover of music, and a world traveler.

Surviving Mr. Rusden are his widow, Margaret (Lawlor) Rusden, and Ethelbert A. Rusden, Jr., a son by his first wife, who died in 1918.

RUDOLPH W. RUSTERHOLZ

Rudolph W. Rusterholz, associated for more than twenty-five years with the Ingersoll-Rand Company, died in Johannesburg, South Africa, on June 12, 1931.

Mr. Rusterholz was born at Peoria, Ill., on September 18, 1877. He was the son of R. &. R. (Hoffman) Rusterholz. He was graduated from Purdue University, Lafayette, Ind., in 1906 with a degree of Bachelor of Science in Mechanical Engineering. He entered the employ of the Ingersoll-Rand Company as sales and mechanical engineer and, during 1907 and 1908, was in charge of irrigation work in the South. During the next two years, he designed and installed various municipal waterworks plants in the Middle West. He then went to South Bartonville, Ill., where he designed and installed a water system for the Illinois General Hospital for the Incurable Insane. In 1911 he also installed a system for the County Infirmary at Oak Forest, Ill. In 1911 and 1912 he was advisory engineer for a complete compressed air installation in the L. &. H. Ry. shops at South Louisville, Ky., and also designed and acted as advisory engineer in connection with the McCarron Lake Pumping Station for the St. Paul (Minneapolis) Waterworks. From 1913 to 1918, Mr. Rusterholz was manager for the Ingersoll-Rand Company at Butte, Mont. He had been located in Johannesburg since 1918 as chairman and managing director of the Ingersoll-Rand Company (S.A.) Ltd.

Mr. Rusterholz belonged to the U.S. Engineer Corps during the Cuban War and during the World War served as Captain in the U. S. Air Service in France. In 1919 he was attached to the American Peace Commission in Paris.

In addition to the A.S.M.E., of which he had been a member since 1913, Mr. Rusterholz belonged to the American Institute of Mining and Metallurgical Engineers, the South African Institute of Engineers, the Engineers Club of London, and a number of clubs in South Africa. He is survived by his widow, Nita M. (Kabat) Rusterholz, whom he married in Portland, Ore., in 1912.

CARL HUGO SATHERBERG

Carl Hugo Satherberg was born at Carlstrona, Sweden, on August 29, 1853. He attended the Technical School of Norrköping, Sweden, and the Polytechnic Institute of Stockholm. His early experience was secured in the drafting room of the Motala Iron Works, Sweden, and the shops of the Swedish government at Liljeholmen, Stockholm. He went to Manchester, England, in 1877 to work in the locomotive shops of Beyer and Peacock.

Mr. Satherberg came to the United States in 1880 and for seven years was employed as chief draftsman at the Midvale Steel Works in Philadelphia. During the next three years he was construction engineer for the Pennsylvania Steel Company, Steelton, Pa. He returned to the employ of the Midvale Steel Company in 1890 and remained with it as mechanical engineer and chief draftsman until he retired in 1917. His most important contribution to the industry was in the design of melting and heating furnaces, heavy hydraulic forging presses, car-wheel rolling mills, and other heavy steel-mill machinery.

Mr. Satherberg became a member of the A.S.M.E. in 1893. He also belonged to the Art Club of Philadelphia. His wife, Eliza-

beth T. (Brant) Satherberg, died in 1929. His own death occurred on July 19, 1931, at the University of Pennsylvania Hospital in Philadelphia.

WILLIAM LAWRENCE SAUNDERS

William Lawrence Saunders, chairman of the board of directors of the Ingersoll-Rand Company, New York, N.Y., died on June 25, 1931, at Teneriffe, Canary Islands, while on a tour of the world. His career as an engineer extended over more than half a century, with a record of distinguished public service in his later life. Perhaps the most important public position he occupied was that of chairman of the Naval Consulting Board, to which he was appointed in 1915. He applied himself to the study of submarine warfare and evolved the theory that an unsinkable boat could be had by filling the inner hull with a large number of wooden boxes, the buoyancy of which would keep the ship afloat no matter how riddled by torpedos she might be. Such a boat the *Lucia*, was fitted out and used for transporting troops to France, but Mr. Saunders' theory was never tested, as the ship was not struck.

Mr. Saunders was born at Columbus, Ga., on November 1, 1856, the son of William Trebell and Eliza (Morton) Saunders. He was graduated from the University of Pennsylvania in 1876 with the degree of Bachelor of Science and was given an Sc.D. degree by that institution in 1911. In 1925 he received the honor cup of the University, awarded annually to its most distinguished alumnus. He had served as president and member of the council of the University of Pennsylvania Club of New York.

For several years following his graduation Mr. Saunders engaged in newspaper work in Philadelphia, and in 1878 he began his engineering career with the National Storage Company of Communipaw, N.J., where he was in charge of hydrographic work and subaqueous rock excavation from 1878 to 1882, in connection with the building of docks, warehouses, and a ship channel at Black Tom Island, New York harbor. Later he was successively engineer, vice-president, and president of the Ingersoll-Sergeant Drill Company, and president of the Haesler Pneumatic Tool Company and the New York Imperial Tool Company. When the Ingersoll-Sergeant Drill Company was succeeded by the Ingersoll-Rand Company he became president and ultimately chairman of the board. He had also been a director of the A. S. Cameron Steam Pumps Works, American International Corporation, and New York & Honduras Rosario Mining Co., and president of the Stayley-Saunders Corporation.

Chief among the inventions by Mr. Saunders was the apparatus for drilling rock under water and the Ingersoll track and bar channelers and gadders for quarrying dimension stone, marble, and limestone. He also invented the radialaxe system for coal mining and the method of pumping liquids by compressed air. For some years he was the editor of *The Compressed Air Magazine*, and he edited "Compressed Air Data," a handbook issued by the publishers of that journal in 1924. He also edited "Compressed Air Information," a symposium of papers on the production, transmission, and use of compressed air, and was the author of "Compressed Air Production," as well as of many pamphlets and articles for the technical press. He was a co-author (with Gilbert and Wightman) of "Subways and Tunnels of New York" and (with Dana) of "Rock Drilling."

During the World War Mr. Saunders also served on the district committee on capital issues of the Federal Reserve Board, on the advisory board of the Fuel Administration, and on the Military Engineering Committee, which at its own expense equipped a regiment and sent it to France.

One of the most prominent members of the Democratic party in New Jersey, Mr. Saunders had been a member of the state's Democratic committee and in the presidential campaign in 1916 was one of New Jersey's representatives on the Democratic National Committee, as personal representative of Mr. Wilson. He was a delegate to the Democratic National Convention in 1924. He had twice been mayor of North Plainfield, N.J., and maintained a home in Plainfield as his voting residence. He had been a member of the New Jersey Harbor Commission and the New Jersey Board of Commerce and Navigation. He was a former governor, director, and deputy chairman of the Federal Reserve Bank of New York, a member of the special committee on workmen's compensation legislation of the New York Chamber of Commerce, a member of the Federal Trade Commission, chairman of the Permanent Group Committee for Nicaragua, and a member of the International Fixed Calendar League. He had been chairman of the industrial training committee of the National Civic Federation, a member of the committee on education of the National Foreign Trade Council, and a member of the United States Chamber of Commerce.

Mr. Saunders was greatly interested in discovering a cure for cancer. In 1926 he announced through the American Society for the Control of Cancer that he would present a prize of $50,000 for the discovery of a positive cure. After two years he was convinced of the futility of the search and withdrew his offer. He was honorary governor of the New York Skin and Cancer Hospital.

Mr. Saunders was very active in the work of engineering and scientific organizations, particularly that of the American Institute of Mining and Metallurgical Engineers, of which he was president in 1915. He represented the Institute at the First Empire Mining and Metallurgical Congress at the Wembly Exposition, London, in 1924. Two years later he founded the Mining Medal of the Institute, to recognize achievement in mining. He was also a member of the Mining and Metallurgical Society of America and the American Iron and Steel Institute.

Others of the four national societies to which he belonged were the American Society of Civil Engineers and the A.S.M.E., which he joined in 1928. He was a former president, trustee, and member of the finance committee of the United Engineering Society.

Mr. Saunders was a fellow of the American Geographical Society, past-president of the American Manufacturers' Export Association, honorary vice-president of the International Benjamin Franklin Society, and a member of the Academy of Political Science and the American Academy of Political and Social Science. Although a resident of the northern part of the United States for many years he kept in touch with the South, where he was born, through membership in the New York Southern Society. His club memberships included the Machinery, of which he was a past-president, and the India House, Engineers', and University Clubs in New York.

His wife, Bertha L. (Gaston) Saunders, whom he married in 1886, died in 1906, leaving him two daughters, Mrs. Maxwell Evarts Perkins and Mrs. John E. Lancaster, Jr.

THOMAS H. SAVERY, JR.

Thomas H. Savery, Jr., who died on December 31, 1930, as the result of a fall, was born at Wilmington, Del., on May 31, 1871, the son of Thomas H. and Sarah (Pim) Savery. He received an M.E. degree from Cornell University in 1896 and during the next five years he was employed in the manufacturing and sales departments of the Pusey & Jones Co., in his native city. His duties included the supervision of manufacture and installation of machinery for pulp and paper mills.

In 1901 he went to Denver, Colo., as manager and treasurer of the Rocky Mountain Paper Company. Mr. Savery had full charge of the purchase and erection of all machinery for the ground wood and sulphite plants as well as the paper mills. A special plant for handling and de-inking waste papers was designed and developed under his supervision.

From Denver, in 1912, he transferred to Sandusky, Ohio, where for four years he was connected with the manufacturing and sales departments of the Sandusky Foundry & Machine Co. He developed and patented several important features in connection with special suction rolls, including rubber covered suction press rolls, suction screen plate cleaners, and top felts for suction rolls to eliminate marking.

Since 1916 Mr. Savery had been in business for himself as manufacturer's agent, with headquarters in Chicago, selling pulp and paper mill machinery exclusively and representing several companies in this line, including the Nash Engineering Company, South Norwalk, Conn., and the Bird Machine Company, South Walpole, Mass. He served also as a consulting engineer, recommending special machinery to meet the varying conditions in pulp and paper mills.

Mr. Savery had been a member of the A.S.M.E. since 1917. He also held membership in the Technical Association of the Pulp and Paper Industry, and the Society of Cornell Engineers, as well as in several clubs.

Mr. Savery married Lida Moore, of Wilmington, in 1897, and is survived by her and by five children, Esther, Helen, Thomas H., 3d, Donald Fenwick, and Eleanor Irving.

WILLIAM SAXON

William Saxon, vice-president in charge of manufacturing for the Miehle Printing Press & Manufacturing Co., Chicago, Ill., died on October 5, 1931, at his residence in Oak Park, Ill. He had not been active in business for several years because of illness.

Mr. Saxon was born in Christiania, Norway, on July 6, 1857, the son of Christian and Alice (Tomilson) Saxon. Later he moved with his parents to Manchester, England, where he spent three years at the Mechanics Institute, being graduated in 1878. He also served an apprenticeship as a machinist with Parr, Curtis & Company of Manchester. He returned to Norway as machinist and draftsman for J. & A. Jensen & Dahl. He came to the United States in 1883 and

secured work as draftsman and machinist on steam engine work for William McGregor & Co., in Chicago.

Mr. Saxon had his first experience in the printing industry as shop foreman from 1887 to 1889 for a Chicago firm in printing press work. During the next two years, he was foreman for the S. K. White Company of Chicago, which at that time had undertaken the manufacture of the then practically unknown Miehle printing press. This company was taken over in 1891 by the Miehle Company, of which Mr. Saxon became general superintendent and later vice-president.

Mr. Saxon joined the A.S.M.E. in 1906. Surviving him are his widow, Marie (Jacobsen) Saxon, whom he married in 1881, and three children, Mrs. Alice Press, Mrs. Curtis Crafts, and William J. Saxon.

ALBERT JEFFERSON SAYERS

Albert Jefferson Sayers, associated with the Link-Belt Company since 1899, died at his home in Chicago, Ill., on October 11, 1931.

Mr. Sayers was head of the Coal Tipple and Coal Washery Department of the Link-Belt Company. He was well known throughout the entire coal industry, having designed and built numerous plants for the handling and preparation of coal, in all parts of the country, and was recognized as one of the leading designers of this type of equipment.

He was born on September 10, 1870, at Troy, Ohio, the son of Judson and Mary (Gearhart) Sayers. He was educated at the University of Illinois, from which he was graduated in 1895 with a B.S. degree in mechanical engineering.

Following his graduation, he was engineer in charge of the steam plant of the Illinois Central Railway at New Orleans, La., for a year, and spent two years as engineer in charge of the Physical Testing Laboratory of the Sargent Steel Company of Chicago. In the thirty-two years of his affiliation with the Link-Belt Company, Mr. Sayers became a leading factor in the mechanical handling, screening, and washing of coal.

He had been a member of the A.S.M.E. since 1913 and belonged to the Manufacturers Division of the American Mining Congress and to other engineering organizations, where his contributions to the advancement of engineering principles adaptable to the industry were recognized factors of importance.

Mr. Sayers is survived by his widow, Mabel J. (Penfield) Sayers, whom he married in 1901, and by two children, Edith E. and Robert P. Sayers.

FRANZ SCHULTZ-BALLUFF

Franz Schultz-Balluff, whose death occurred in Germany on August 5, 1931, was born at New York, N.Y., on December 15, 1887. He was educated in Germany, attending the Realgymnasium at Stuttgart for ten years and securing his engineering degree from the Technische Hochschule there in 1911. He entered the Diesel Department of the Motorenfabrik Deutz A.G., as draftsman, and remained with this company until his death, with the exception of four years during the World War which he spent in the Heavy Artillery.

Dr. Schultz-Balluff devoted a large part of his time to research work. He was made chief of the Research Department of the company in 1918, technical manager in 1920, and vice-president in 1923. He was also technical manager of the Motorenfabrik Oberursel A.G. from 1921 to 1927, reorganizing its plant near Frankfort on the Main. From 1925 to 1927 he was engaged as technical manager of the Maschinenbauanstalt Humboldt at Köln-Kalk, a firm dealing with the standardization of steam-turbines, ice-machines, and compressors.

Dr. Schultz-Balluff studied and designed a large number of Diesel engines, decreasing the number of types and making different types adaptable to all fuels and services. He had been a member of the A.S.M.E. since 1927 and also belonged to the Verein Deutscher Ingenieure.

Surviving Dr. Schultz-Balluff are his widow, Ida Schultz-Balluff, and three children, Michel, Volker, and Renate Schultz-Balluff.

SEATON MACKENZIE SCOTT

Seaton Mackenzie Scott, who died of heart failure on January 12, 1931, in Brooklyn, N.Y., was born on February 19, 1860, at Montreal, Canada, the son of Thomas Seaton and Mary (Mackenzie) Scott. He studied mechanical engineering at the Collegiate Institute, Ottawa, Canada, and served a three-year apprenticeship in the pattern shops and foundry of the Vulcan Iron Works there. He secured his drafting and shop experience in the office of the chief engineer of the Canadian Government Railway at Ottawa and at the Kingston Locomotive Works at Kingston, Ontario. He was employed by the Canadian Pacific Railway and Grand Trunk Railway as locomotive foreman, traveling inspector, and draftsman.

Mr. Scott left Canada in 1885, and later became a naturalized citizen of the United States. He was employed by the C. &. C. Motor Company, Chicago, the Brooklyn Distilling Company, and the Kelly & Jones Co., Greensburg, Pa., prior to 1898, when he began consulting work in New York. The Goldschmidt Thermit Company (now the Metal & Thermit Corp.) and Abendroth & Root Mfg. Co. were among the companies by which he was employed about 1900. Later he was engaged for a number of years in power plant work for the N.Y. Edison Co. and other Edison organizations. He helped to design and supervised the construction of several plants, including one in Philadelphia and the Waterside Station in New York, and also had charge of some construction work at the laboratories in West Orange.

Mr. Scott served as a Major in the United States Air Corps during the World War. He was connected with the Plant Appraisal Bureau, making appraisals of many ma'or aircraft plants, including the Wright-Martin Aircraft Corporation and the Aeronautical Engine Corporation. Upon completion of his war service, he re-entered commercial engineering as designing engineer for Ophulls & Hill, Inc., refrigeration engineers, of Brooklyn, a position which he held until shortly before his death.

Mr. Scott became an associate of the A.S.M.E. in 1896 and a member the following year. He is survived by his widow, Louise (Ludlum) Scott, and by two sons, Seaton M. Scott, Jr., and Maxwell J. Scott.

DWIGHT SEABURY

Dwight Seabury, a native of Providence, R.I., died in that city on April 21, 1931, following a brief illness.

Mr. Seabury was born on February 14, 1863, the son of Dr. Frederick and Amelia (Wheaton) Seabury. He attended the Providence High School and the Mowry & Goff (technical training) School in that city, and in 1881 began a three-year apprenticeship with Thompson & Nagle, Providence, in general mill construction. He then spent two years in the Providence mills of B.B. & R. Knight, following which he secured a position with the firm of F. P. Sheldon. After nine years with this company, during three of which he was in charge of the office, he went into business for himself, with an office in Pawtucket, R.I.

Throughout his life Mr. Seabury engaged in mill engineering and architecture. Among the Pawtucket mills designed and built under his supervision are the plants of the Royal Weaving Company, E. Jencks Manufacturing Co., A. T. Athetton Machine Co., Solvay Dyeing and Textile Co., and the Lumb Knitting Company. He designed the G. W. Parks Co., Waite-Thresher, N-Barstow Company, and Hanley buildings in Providence, the plant of the Portland Silk Company, Middletown, Conn., the Brighton Mills, Passaic, N.J., Pennekees Mills, Valley Falls, R.I., and Fort Dummer Mill, Brattleboro, Vt.

Mr. Seabury became a member of the A.S.M.E. in 1906. He also belonged to the New England Manufacturers' Association, the Southern New England Textile Club, and the Dixon-Spencer Associates, a social organization of prominent mill men. He is survived by his widow, Hattie I. Seabury.

BRUCE SETTLE

Bruce Settle, whose death occurred on December 29, 1930, was born at Gainesboro, Tenn., on February 7, 1898. He attended the United States Naval Academy and was appointed a Midshipman in June, 1917. Three years later he was commissioned a regular Ensign. He served for a time as First Assistant Engineer on the U.S.S. *Raleigh* and, at the time of his death, was stationed at the United States Navy Yard at Washington, D.C.

Lieutenant Settle received an M.S. degree from Columbia University in 1927. He became a junior member of the A.S.M.E. in 1928.

WILLIAM H. SHELMIRE, JR.

William H. Shelmire, Jr., died in his native city of Philadelphia, Pa., on July 31, 1931. He was born there on October 1, 1855, the son of William H. and Sarah Ann Shelmire. After completing his high-school education, he was apprenticed to A. L. Archambault, builder of marine and stationary engines. The business failed before Mr. Shelmire had compled his apprenticeship and he spent a short time in the engine room of the steamer *Juniata*, shipping to Cuba and New Orleans. Later, he completed his apprenticeship with G. C. Howard of Philadelphia. He also availed himself of the lectures and library of The Franklin Institute.

Upon the completion of his apprenticeship. Mr. Shelmire went to Terre Haute, Ind., where he spent three years in the employ of the

Vandalia Railroad Company as gang foreman in the erecting shop, draftsman, and mechanical engineer. He then returned to Philadelphia to work, first as draftsman, and later as assistant to the superintendent at the Pencoyd Iron Works of A. & P. Roberts & Company.

At the time Mr. Shelmire became a member of the A.S.M.E. in 1889 he was assistant superintendent of the Diamond State Iron Company at Wilmington, Del. Shortly afterward, he was engaged by the Kensington Engine Works, Ltd., of Philadelphia, where he remained for about two years. From 1903 to 1905 he was general superintendent of the Creswell & Waters Company, Nicetown, Pa., in 1906 mechanical engineer for the Cambria Steel Company, Johnstown, Pa., and from 1907 to 1913 mechanical and supervising engineer for the John Lang Paper Company of Philadelphia. In the intervening periods, and from 1913 until his death, he engaged in consulting work. The firm of Charles W. Shelmire, Inc., of which he was vice-president for the first year and subsequently president, was established in 1926.

Mr. Shelmire was a member of the Masonic fraternity and of the Society of Friends. He is survived by his widow, Lucy (Cope) Shelmire, whom he married in 1882.

VISCOUNT EI-ICHI SHIBUSAWA

Japan's "grand old man," Viscount Ei-ichi Shibusawa, died on November 11, 1931, He was 91 years old. One of America's earliest and most steadfast friends in Japan, he was called the "people's foreign minister" because of the trips he made to the United States, particularly when Japan's immigration question was under discussion. For years he was the nation's leading banker, merchant, industrialist, mine owner, and one of the founders of Japan's modern business system. His power at home was exceeded only by that of the royal family.

Viscount Shibusawa's friendship for the United States was apparent as early as 1861 when, at a time when the intentions of foreigners were seriously misunderstood in Japan, he stood guard with others night and day before the residence of Townsend Harris, the first Consul-General of America in Japan.

Those were turbulent days in Japan, for among its people were many who bitterly censured the Shogunate for alleged blunders in the conduct of diplomatic affairs and who nursed intense hatred toward foreigners, and frequently resorted to violence. In January, 1861, Henry C. J. Heuskin, Mr. Harris' valued associate, was assassinated by rowdies. All the foreign ministers then in Japan, with the exception of Mr. Harris, blamed the Shogunate for its incompetency to protect them, and closed their legations and withdrew. Mr. Harris did not approve of this step and remained to attend to his duties. It was then that the guard was placed before his residence. Commenting on this action later Viscount Shibusawa wrote: "The courageous and magnanimous attitude taken by Mr. Harris on this critical occasion made a strong appeal to the imagination of our people, who were now convinced of the genuineness of his sentiment toward them, and who from that moment began to put trust in America as a true friend of Japan."

His regard for the United States is revealed by the fact that he headed many of the Empire's delegations to American shores in the interests of friendship and economic exchange between the two nations. He was president of the American Relations Committee of Japan, and later became chairman of the American-Japan Society of Tokyo, similar to the organizations in the United States.

Viscount Shibusawa was made an honorary member of The American Society of Mechanical Engineers in 1929 during the visit of American engineers to the World Engineering Congress at Tokyo. This was in recognition of the important part he played in the industrial and economic development of the Japanese nation, his efforts toward the promotion of world understanding, and his support of and interest in physical and chemical laboratories and in all branches of engineering in Japan.

Even after he had reached an advanced age Viscount Shibusawa continued his world travels, the last of his several visits to the United States being made in 1921 at the age of 81. Even then, though he had retired from many of his business connections, he was most active, rising at six or seven in the morning and working from 12 to 15 hours a day. During his life he was president of more than 70 corporations, for nearly three-quarters of a century was a trusted adviser of successive Japanese governments.

Viscount Shibusawa was born on February 13, 1840, the son of an indigo merchant, in a small village near Edo, the present city of Tokyo. In his boyhood he received a thorough education in the Chinese classics and learned the art of the sword from the masterful Chiba. Going to Kyoto, then the capital, he entered the service of Lord Hitotsubashi, member of an influential branch of the Shogunate Tokugawa family. In 1867, a year before the downfall of the Shogunate, he was sent to France with Lord Bimbu, a brother of the reigning Shogun. When the imperial government was established in 1869 he was made a vice-minister in the Finance Department. Three years later he left the government service and began his great business career. One of his first acts was to establish the First National Bank, the pioneer national bank in Japan. He helped to found the Tokyo Chamber of Commerce, the Tokyo Bankers' Association, the Bankers' Clearing House, the Tokyo Stock Exchange, the Tokyo Marine Insurance Company, the great maritime firm Nippon Yusen Kaisha, and scores of other businesses of world-wide influence.

He was a leading figure in the organization of the Japan Mail Line and the expansion of the nation's maritime efforts. The development of the cotton spinning industry in Japan is largely due to his initiative, as well as other manufacturing enterprises such as silk weaving, hemp and rope mills, concrete plants, sugar refining, and affiliated projects of this modern age. In cooperation with government authorities he acted to improve Japan's natural production and resources by reclamation work and harbor construction, by improving agriculture to increase returns, and through the medium of introducing artificial fertilizers and high grade cattle and stock breeding.

Early in the Meiji era, feeling that Japan's younger generation needed more facilities for commercial education, Viscount Shibusawa established a training school which became the forerunner of the present Tokyo Higher Commercial College. In charity movements his name always led all the rest. He was a leader in the establishment of the Tokyo Poor House and other charitable organizations. His promotion to the peerage in 1900 was in consideration of the immense service he rendered to the country's commercial and industrial development as well as to the cause of public welfare.

As Baron Shibusawa, he made his first trip to this country in 1902 to study commercial and industrial progress. In 1909 he visited the United States as the head of an official Japanese business delegation invited to come here by the Chamber of Commerce of the United States. In 1915 Baron Shibusawa returned to the United States to visit the San Francisco exposition and discuss certain commercial and political questions.

One day of each month Viscount Shibusawa devoted to religious matters, having read to him passages from Confucius. He was one of the greatest authorities in Japan on Chinese classics, and was an ardent follower of Confucian precepts. It is probable that the fortune he had left is the largest ever accumulated by any individual business man in Japan.—C.W.S.

EDWARD ALFRED SIMMONS

Edward Alfred Simmons, president of the Simmons-Boardman Publishing Corporation, New York, N.Y., died of cerebral hemorrhage at his home in Brooklyn on September 30, 1931. He was born in Brooklyn on March 20, 1875, the son of John Wesley and Agnes Anne (Owers) Simmons. He was educated in the public schools there. He entered the employ of the *Railroad Gazette*, a predecessor of the present *Railway Age*, at the age of 14 and was continuously associated with this publication until his death. In 1911 Colonel Simmons effected the merger of the *Railroad Gazette* and the *Railway Age* (Chicago), which were thereafter published as the *Railway Age-Gazette*, later simplified to *Railway Age*, becoming president of the Simmons-Boardman Publishing Company which was organized to take over the properties. The *Railway Review* was merged with the *Railway Age* in 1926.

In 1928 the Simmons-Boardman Publishing Corporation was formed under the presidency of Colonel Simmons. It acquired control of the Simmons-Boardman Publishing Company, publisher of the *Railway Age*, *Railway Mechanical Engineer*, *Railway Engineering and Maintenance*, *Railway Electrical Engineer*, *Railway Signaling*, *The Boiler Maker*, and *Marine Engineering and Shipping Age*, and also of the American Builder Publishing Corporation, publisher of the *American Builder*, with which several other building periodicals have since been combined under Colonel Simmons' leadership.

Colonel Simmons was chairman of the board of directors of the American Saw Mill Machinery Company, which he organized in 1904; president of the American Saw works, organized in 1907; and chairman of the board of the American Machine Tool Company, organized in 1915, Rogers-Eagle Grinding Machine Company, organized in 1926, and the Corley Manufacturing Company and Eagle Saw Company, purchased in 1926 and 1927, respectively, by the American Saw Mill Machinery Company.

Colonel Simmons was regarded as an authority on water and rail transportation problems. As chairman of the American Marine Standards Committee he was an outspoken advocate of the development of a great merchant marine in the United States. He was chairman of the United States delegation to the International Railway Congress in London in 1925 and to the Madrid Congress in 1930.

For his work as a major in the Construction Division of the Army

during the World War, Colonel Simmons was praised highly in a citation signed in 1919 by Brigadier-General R. C. Marshall, Jr. The citation read: "A spirit of loyalty and competent effort has characterized your service in the Construction Division and was particularly helpful in the successful prosecution of the largest war construction program ever undertaken." Colonel Simmons supervised army construction work in and around New York, and during 1918 was in charge of twenty-two separate projects. These included the $2,500,000 Government hospital at Fox Hills, Staten Island. When the War closed he was under orders to go to France.

After leaving the army he was commissioned lieutenant-colonel in the Reserve Corps in 1919. Three years later he became a colonel. He was active in 1919 in organizing several posts of the American Legion. He was made commander of the Kings County (N.Y.) Post No. 500 and was a representative of the Legion at the funeral services for the first soldiers brought from France to the United States for reburial. Later he took charge of many similar services in Brooklyn and Hoboken, N.J. He was commander of the Legion in Kings County in 1923 and became president of the Department of New York of the Reserve Officers Association of the United States in 1930. He had also served two terms as president of the Brooklyn Chapter of the Reserve Officers Association and was a member of the Society of American Military Engineers, Military Order of the World War, New York Society of Military and Naval Officers of the World War, and the Quartermasters Association.

Colonel Simmons became an associate of the A.S.M.E. in 1919. He also belonged to the American Society of Civil Engineers and the Society of Naval Architects and Marine Engineers. He was chairman of the Endowment Committee of Engineering Foundation, Inc.

Colonel Simmons was re-elected honorary president of the New York State Division of the Izaak Walton League of America in September, 1931. His clubs included the Engineers', Bankers, Railroad, Union League, Automobile, and Army and Navy, of New York; Engineers' Country of Roslyn, L.I.; Congressional Country, and Army and Navy of Washington, D.C.; Adirondack League Club of Old Forge, N.Y.; Thendara Golf Club, Thendara, N.Y.; Inlet Golf Club, Inlet, N.Y.; Lake Placid; and Sojourners and Union League, Brooklyn. He is survived by his widow, Ida Boone (Rines) Simmons, whom he married in 1914, and by three daughters, Aline, Doris, and Elizabeth.

ORVILLE CAMPBELL SKINNER

Orville Campbell Skinner died unexpectedly on May 19, 1931. He had been a member of the A.S.M.E. since 1915 and also belonged to the American Institute of Mining and Metallurgical Engineers, American (and International) Society for Testing Materials, American Iron and Steel Institute, and Engineers Society of Western Pennsylvania.

Mr. Skinner, the son of Orville Bird and Helen (Campbell) Skinner, was born at Cleveland, Ohio, on January 19, 1872. He was graduated from the Case School of Applied Science, Cleveland, with a B.S. degree in 1894 and spent the next year in the Chemical Laboratory of the Otis Steel Company in Cleveland. After a year as chief chemist for Corrigan McKinney & Co., Cleveland, he went to Pittsburgh, Pa., to accept a similar position for the Schoenberger Steel Company. Subsequently he was made superintendent of the Open Hearth Department of that company, with which he remained until 1899. During the next two years he was manager of the Open Hearth Department of the Tidewater Steel Company at Chester, Pa. He became connected with the Standard Steel Works Company, Burnham Pa., a subsidiary of the Baldwin Locomotive Works of Philadelphia, in 1901, serving first as superintendent of the Open Hearth Department, and then successively as assistant superintendent, superintendent of manufacture, and works manager. In all of these positions his work included assistance in the design and installation of extensions and improvements. He resigned on April 1, 1930, to become president of the Lewiston (Pa.) Aircraft & Airways, Inc.

Mr. Skinner is survived by his second wife, Mrs. Flo (Fisher) Skinner, whom he married in 1929, and by two children, Mrs. Charlotte (Skinner) Neher and Orville Bird Skinner.

WILLIAM SMITH, SR.

William Smith, Sr., the son of William and Lillias Smith, was born at Glasgow, Scotland, on May 26, 1861. He attended the Tennents Industrial School at St. Rollox, Glasgow, and served an apprenticeship as a machinist in the toolroom and millwright department of the Hydepark Locomotive Works, Springburn, Glasgow. His first position was in the Caledonian railway shops of that city. From 1880 to 1883 he was employed by John Scott & Co., shipbuilders, of Greenock, Scotland. He then went to Paisley, Scotland, where he was engaged in 1884 in the erection of steam steering gear and sugar mill machinery. During the next two years he was employed as assistant foreman at the South Works and North Works, Forth Bridge, of Wm. Arroll & Co., Queensferry, Scotland.

Mr. Smith came to the United States in 1887, and from then until 1891, and again from 1893 to 1895, was millwright foreman for the Lower Union Mills of the Carnegie Steel Company, Pittsburgh, Pa. In the interim he was metal patternmaker for the Westinghouse Air Brake Company, of Pittsburgh and Wilmerding, Pa. In 1896 ant 1897 he was engaged in the erection of blast furnace machinery and mill machinery for the Keystone Bridge Company, Pittsburgh. He then went to Duquesne to serve as millwright foreman at the Duquesne Steel Works and later as assistant master mechanic of the Duquesne Blast Furnaces and Carnegie Steel Company. For six years beginning in 1900 he was connected with the Jones & Laughlin Steel Co., of Pittsburgh, as master mechanic at the Eliza Furnaces, and during the next seven years he held a similar position at the Soho Furnaces of the Company. He was next engaged as general master mechanic of the Pennsylvania Steel Company, Steelton, for two years. In 1917 and 1918 he was employed by the Midvale Steel Company, Coatesville, Pa., first as assistant to the superintendent, and later as general master mechanic. He accepted a similar position at the Coatesville Works of the Midvale Steel & Ordnance Co. in 1920 and remained there for three years. He was then connected successively with the New Process Copper Castings Company, Pittsburgh, of which he was president; the Lumen Bearing Company, Youngstown, Ohio; the Falcon Bronze Company, Youngstown, Ohio, as sales manager; the Lawrenceville Bronze Company, Pittsburgh, as sales manager; and the Copperweld Steel Company at Glassport, Pa. He was superintendent of the Rankin Works of this company in 1928 and 1929, and since then had held the position of maintenance superintendent of its Glassport Works.

Mr. Smith became a member of the A.S.M.E. in 1911. He also belonged to the Engineers Society of Western Pennsylvania and the Masonic fraternity. He is survived by his widow, Agnes (Patterson) Smith, whom he married in 1885, and by three sons. He died on November 10, 1930, at Glassport.

JOHN EDWIN STARR

John Edwin Starr, president of the Starr Engineering Company, consulting refrigerating engineers, New York, N.Y., died on June 27, 1931, at his summer home in Mattituck, Long Island, N.Y.

Mr. Starr was born in Litchfield, N.Y., on February 24, 1860, the son of John and Susan Starr. After being graduated from the Utica (N.Y.) Academy he went to St. Louis, Mo., where he entered the refrigerating field as draftsman for the St. Louis Automatic Refrigerating Company. Subsequently he designed and built refrigerating plants and cold storage warehouses for the Denver (Colo.) Automatic Refrigerating Company, and the Atlantic City (N.J.) Cooling Company, the Manhattan Refrigerating Company, New York, Kings County Refrigerating Company, Brooklyn, Baltimore (Md.) Refrigerating & Heating Co., and many other companies. He organized the Starr Engineering Company in New York in 1898.

In addition to the design and construction of refrigerating apparatus and plants Mr. Starr conducted tests of refrigerating machinery and insulating materials, made special investigations in his field, and contributed many articles to the technical publications dealing with refrigeration. He was the inventor of a low-temperature machine which attains temperatures of from 30 to 75 deg. below zero and of several absorption refrigerating devices. He was regarded as a leader in the establishment of refrigerating engineering among the sciences.

Mr. Starr presided at the first meeting of The American Society of Refrigerating Engineers, held in 1904, and served as president of the society for one year. In 1929 he was elected an honorary member of the organization. He was the American delegate to the International Congress of Refrigeration in Paris in 1908, and was the recipient in 1896 of the Franklin Institute Medal, in recognition of the work he had already done in refrigeration. He had been a member of the A.S.M.E. since 1905 and belonged also to the St. Louis Academy of Sciences, the London Institute of Cold Storage and Refrigeration, and the Engineers', Salmagundi, and Transportation clubs in New York. He is survived by a daughter.

JOSEPH S. STEPANOV

Joseph. S. Stepanov, research engineer for the Worthington Pump & Machinery Corp., Harrison, N.J., who died suddenly on April 10, 1931, was born at Ismail, Russia, on April 15, 1881. He secured his early education in Odessa and was graduated from the Imperial Institute of Technology in 1908.

During the first five years after his graduation Mr. Stepanov was employed by the Petrograd Metal Works Co., Inc., in the design and testing of steam turbines, boilers, and auxiliaries for power stations, waterworks, and ships. During the same period he gave lectures to evening classes on thermodynamics, steam turbines, internal combustion engines, and boilers. These lectures later appeared in book form under the title of "Steam Boiler Operation."

From 1913 to 1919 Mr. Stepanov was chief designer, production engineer, and chief engineer of the mechanical branch of the Russian Baltic Engineering & Shipbuilding Co., Inc., Reval, Baltic Sea. Prior to the World War he spent some time at Berlin with the General Electric Company studying developments in the design and construction of steam turbines; and at Hamburg with the Vulcan Works investigating progress in boilers and auxiliaries and power plants as a whole. During the War he designed, constructed, and tested machinery for destroyers, light cruisers, dry docks, submarine mine layers, and other equipment for the Russian Navy. He also had charge of the cost estimating and planning bureaus of the Russian Baltic Company. He devised a nomogram of chief dimensions of propellers, to facilitate the calculations for ship propellers; an article describing this nomogram was published in the Russian "Engineering News-Record" in 1915 and later included in programs of Russian institutions for engineering training. Following the War he helped to solve the problems connected with changing the work of the company from shipbuilding to locomotive repairing. He was appointed manager and a member of the board of directors by the Esthonian government. In 1919 Mr. Stepanov was made manager of the Ilmarine Metal Works, Inc., at Reval. He handled locomotive repair work for Esthonia and the construction of small engines.

Mr. Stepanov came to the United States in 1924 and was engaged as draftsman, designer, and experimenter by the Akimoff propeller Company, Philadelphia, Pa. The following year he worked on the design of reinforced-concrete arch bridges for the Hollywood Dredging & Construction Co., Hollywood, Fla., and then was engaged as designer by the City Engineering Department of Coral Gables, Fla. From 1927 to 1929 he worked on designs for dry quenching plants for heat utilization of hot coke, hot cement clinker, etc., for the Dry Quenching Equipment Corporation, New York, N.Y., a subsidiary of the International Combustion Engineering Corporation. He had been with the Worthington Corporation since that time and had suggested several refinements in design, including the diffusion ring of continuous curvature used in their "W" pumps.

Mr. Stepanov became a member of the A.S.M.E. in 1927 and had presented papers on centrifugal pumps and condensers at meetings of the Society. He was also active in the work of the American Standards Association.

Mr. Stepanov is survived by his widow, Mrs. Alexandra (Doemke) Stepanov.

ALBERT J. STRATMAN

Albert J. Stratman was born at Newport, Ky., on July 1, 1882, the son of John F. W. and Carolina (Kuehnle) Stratman. He studied mechanical drawing at the Ohio Mechanics Institute, Cincinnati, for four years, and took a mechanical engineering course through the International Correspondence Schools. He also served an apprenticeship as tool and die maker with Fay & Egan of Cincinnati. From 1906 to 1910 he was employed at the Baker Iron Works, Newport, Ohio. He then went to Ontario, Calif., where he was connected for eleven years with the Hot Point Appliance Company, now the Edison Electric Appliance Company. He worked for four years in the engineering and experimental departments, and during the latter part of his connection with the company was in charge of the heat treating department and metallurgical research. From 1921 to 1923 he assisted in the design and development of the "Silent" Rich lawn mower for the Rich Steel Products Company, Los Angeles. During the next year he was connected with the Teetor Adding Machine Company, Pomona, Calif., and subsequently with the Pacific Gas Radiator Company, the Pribnow Saw Sharpening Machinery Company, and the Simonds Saw & Knife Works, all of Los Angeles. For about four months prior to his death on March 15, 1931, he was superintendent and mechanical engineer for the Ballou Manufacturing Company, Ontario, Calif.

Mr. Stratman carried on extensive experiments in fuel atomization for internal-combustion engines, relative to which he was granted a patent. He served as a corporal in the Infantry in the Spanish American War, and formed a Home Guard Company during the World War at Ontario. For six years he served on the Ontario Board of Education. He became an associate member of the A.S.M.E. in 1922 and also belonged to the Masonic fraternity. He is survived by his widow, Christine Stolle Stratman, whom he married in 1904, and by three sons.

SAMUEL WESLEY STRATTON

Dr. Samuel Wesley Stratton died suddenly at his Back Bay, Boston, home of coronary occlusion on October 18, 1931, at the age of seventy. He had just dedicated a tribute to Thomas A. Edison, whose death occurred a few hours before. His tribute to Edison reads in part: "It seldom has fallen to the lot of any one man to be of such service to humanity. The world mourns that great benefactor." The tribute might very well have been written for Dr. Stratton.

Dr. Stratton was born in Litchfield, Ill., on July 18, 1861, the son of Samuel and Mary B. (Webster) Stratton. He spent his early life on a farm, where he acquired a habit of hard work and accumulated knowledge of the common things of life which formed the basis for his evaluation of the fitness of things. He liked all living creatures and folks most of all, but the mechanical features of farm life seemed to fascinate him.

He earned his way through the University of Illinois by doing various kinds of mechanical work. His warmth for machines was somewhat akin to his unconscious warmth for persons who had befriended him. Throughout his life he was never without his private shop where he could work with his own hands. This was recreation and exercise. Moreover, the shop was to him a retreat, like unto the mountain top for others. He cast his lot with the machine age wholeheartedly and at the same time he recognized that men were even more important than the machine.

The University of Illinois granted him his degree of Bachelor of Science in mechanical engineering in 1884 and afterward he went through the usual steps from instructor to professor of physics and electrical engineering. In 1892 he left his alma mater to become associate professor of physics and later professor at the University of Chicago, where he remained until 1901, when he was forty years old. In his previous position his research work, such as time sufficed, tended in the direction of the applications of physics to engineering. At Chicago he was enraptured with pure research as conducted by Michelson. He cherished the inspiration that he received from this great man and ever afterward he kept within the easy touch of his hand interferometers, Nichols prisms, quartz flats, and the like. Thus it is clear that he had the vision of the importance of pure research, as well as of engineering. He ever regretted that he could not have the pleasure of doing more work in both these fields with his own hands and often felt that sometime this privilege might come.

At forty Dr. Stratton was not marked as a genius either in teaching or research, but his foundations were well laid and his accumulated understanding of human nature was great.

Through the instrumentality of his college chum, Mr. Frank A. Vanderlip, who was Assistant Secretary of the Treasury, a new opportunity was opened. Almost single-handed Dr. Stratton engineered through the enactment of March 3, 1901, which created the Bureau of Standards out of the office of weights and measures. Secretary Lyman J. Gage invited him to become the first director. His vision of the usefulness of such a bureau was so clear that during his entire administration he was not handicapped by the organic act setting forth the functions and scope of activity. In 1903 the Bureau was transferred to the Department of Commerce. From the beginning he sensed the need of basic work in science and engineering as well as of testing, for the development of commerce and industry. He was one of the pioneers in this country in creating interest in basic research as an aid to industry, but near the end of his life he smiled ironically when a board of directors seemed to think that research was a "cure all." He knew that research was only an important essential need but that it called for rare talent and patience, working undisturbed by impatient executives and committees.

Dr. Stratton, like most great men, was modest, even approaching shyness. In building up the Bureau of Standards on the foundations that he had laid, he depended upon friends and genuine material interests to carry the structure forward. The technical needs of the Government during war time brought forth the large industrial building. The needs of aviation called for wind tunnels. Many unselfish and far-seeing men such as Ambrose Swasey and the late John Brashear were continually his advisers. He assisted and encouraged many organizations that seem to foster technical advance. With Dr. Walcott, there was started the National Advisory Committee for Aeronautics. He was a leader in the National Screw Thread Commission and also in the International Conference on Weights and Measures. Many other such organizations knew the value of his membership and quiet counsel. He served in the Spanish-American War as lieutenant.

His success at the Bureau of Standards may be measured by the output of the institution plus the material assets, and the fixation of ideals which it will probably always maintain. The elements which entered into this were many. His evaluation of men and his warm hold on the staff counted greatly. The conception of the tie-in of

basic research with the industries as well as with Government departments more than doubled the size of the Bureau and also doubled its output and recognition. The members of the staff were encouraged to maintain connections with engineering societies. He had been a member of The American Society of Mechanical Engineers since 1913, and served on the Power Test Codes Committee on definitions and values for seven years. He met every demand of the Society in providing men and equipment. Likewise he cooperated with other organizations, never unmindful of the interests of the Bureau.

In 1923 all New England seemed to glory with Massachusetts Institute of Technology when Dr. Stratton was elected president. By correspondence with Prof. Samuel C. Prescott, who was chairman of the faculty (1929-1931), I have learned that the Massachusetts Institute of Technology entered on an era of new progress in national usefulness, characterized by the extension of research in engineering and industry, as well as in pure science, and the expansion of the work of the Institute into new fields of applied science. Courses of instruction were offered which are not only meeting the demands of today but which prevision those of tomorrow. During his administration, departments or divisions of aeronautical engineering, hydraulics, physical metallurgy, municipal and industrial research, public health engineering, and ship operation were established.

Dr. Stratton saw at once the significance of the important contacts with great industries which had been previously made and which have resulted in the cooperative courses in electrical and chemical engineering, and with his support these have been fostered and made highly effective. Research has been greatly stimulated, and new laboratories for special fields of engineering investigation have been secured through the generosity of alumni and friends. Special opportunities have been made available for students of exceptional ability by the development of the so-called "honors groups," while a broadened range of general studies has provided varied cultural opportunity for all students. Foreign lecturers have brought the results of the best teaching, research, or practice abroad. Advisory departmental committees made up of men eminent in professional or business life have aided by wise suggestion in the solution of curricular questions.

In thus enhancing the opportunities for scientific training, Dr. Stratton was insistent that the personal or human factor should not be overlooked. A new group of comfortable and convenient dormitories for undergraduates has been made available through alumni cooperation. A new infirmary, given as a memorial to a former student and equipped with ample facilities for physical examination and medical service, has made possible close supervision of the health of students and proper care and treatment in case of accident or illness.

The foundation of a chair of humanics has made it possible for students to secure through lectures and discussion groups, some real knowledge of those matters dealing with the human relations which are essential factors in business and industrial life, and in the personnel problems of administration. These significant and liberalizing tendencies in technical training, as well as the continued insistence on the teaching of fundamental principles and the cultivation of a true spirit of research, are outstanding features characterizing the trend of the development of Technology under Dr. Stratton's presidency.

With the change in administrative organization in 1930 by which he became Chairman of the Corporation and Dr. Compton assumed the duties of President, Dr. Stratton's interest in educational problems remained undiminished. With this division of the onerous duties incident to the administration of this great school, the splendid way in which these two men cooperated for the development of the Institute redounds greatly to the credit of each. There existed between them a close bond, not only of community of interests and of ideals, but of mutual considerateness and deep affection.

Dr. Stratton took a deep and friendly interest in the undergraduates and easily won their highest respect and admiration. His door was ever open to them and they appreciated his true friendliness. He very skilfully exerted his influence on those who were leaders in student life, as well as in their professional interests. He saw in the undergraduate professional societies a great opportunity for training in the presentation and discussion of scientific papers and reports, as well as preparation for membership in the national engineering and scientific societies to which they would later become eligible, and to stimulate their interest in this important respect he personally provided, in 1931, worthy prizes to be won in competition between men from the various societies. This established a precedent for what will probably became an annual event of significance—the Stratton Prize competition.

Another little known act illustrating his vision was the establishment of a joint committee of students and faculty members for the mutual discussion of problems of curricular character, and of other matters in which both students and staff had a common interest. This committee has been highly serviceable, and has great potential value in bringing the two great groups together through recognized representatives who can clearly present the problems to be considered. While little publicity has been given to this and similar agencies for securing solidarity of aims and ideals in staff and student body, the results already indicate the far-reaching possibilities which Dr. Stratton had visioned.

The University of Cambridge honored him with the Doctorate of Science in 1909, Yale University gave him the D.Sc. degree in 1918, and Harvard later honored him with the L.L.D. degree. He was an officer of the Legion of Honor. There were many other honors and connections which only emphasize the social aspects of the great and yet retiring man.

F. C. Brown[1]

LAFAYETTE PINKNEY STREETER

Lafayette Pinkney Streeter, who died at Long Beach, Calif., on August 7, 1931, was born in Brooklyn, N.Y., on December 25, 1874, the son of Lafayette V. and Emma (Pinkney) Streeter. He prepared for college at the Columbia Institute and attended the Stevens Institute of Technology for three years, receiving a certificate with the class of 1900. In the following year, he entered the Mechanical Engineering Department of the New York Air Brake Company, Watertown, N.Y., as an assistant. He was appointed first assistant mechanical engineer a year from that time and, not long afterward, engineer of tests. In 1905 Mr. Streeter resigned from the New York Air Brake Company to become assistant air brake inspector for the Southern Pacific Company in Los Angeles, Calif. Four years later, he was promoted to the position of air brake inspector. He remained with that company until 1912, when he went to Chicago as air brake engineer for the Illinois Central Railroad. He returned to Los Angeles in 1925 and made his home there and at Long Beach until his death.

Mr. Streeter became a junior member of the A.S.M.E. in 1902 and an associate in 1903. He is survived by his widow, Cora (Doussang) Streeter, whom he married in 1918.

VICTOR O. STROBEL

Victor O. Strobel, president of the Rue Manufacturing Company, Haverford, Pa., was born on January 6, 1859, at Cincinnati, Ohio, the son of Mr. and Mrs. John Strobel. He attended high school in that city and subsequently the Ohio Mechanics Institute. From 1875 to 1879 he secured practical experience in the machine shop of Lambert & Gordon at Ironton, Ohio. He then secured a position as designer and chief draftsman for Witherow & Gordon at Pittsburgh, designers and constructors of blast furnaces and steel plants and agents for Whitwell firebrick hot blast stoves. In 1884 he spent six months in Europe compiling data for the manufacture of iron and steel. He returned to the United States to become secretary and treasurer of the firm of Gordon & Strobel of Philadelphia, designers and builders of blast furnaces and steel works. His company constructed the blast furnaces of the Tennessee Coal, Iron & Railroad Co. at Ensley, Ala., and many others in the South and West. In 1886 the firm was enlarged, its name becoming Gordon, Strobel & Laureau, and the business extended to small Bessemer steel and open-hearth plants. About this time, Mr. Strobel also became treasurer of the Philadelphia Engineering Works, Ltd., a position which he held until 1898. In that year he was appointed general manager of the Crane Department of the Niles-Bement-Pond Company. He resigned this position in 1925 when he became president of the Rue Manufacturing Company.

Mr. Strobel became a member of the A.S.M.E. in 1892. His death occurred on October 1, 1931. Surviving him are his widow, Annie S. (Harbaugh) Strobel, and a daughter, Mrs. Springer Starr.

GEORGE FILLMORE SWAIN

George Fillmore Swain, who won distinction both as a teacher of civil engineering for nearly fifty years and as a consulting engineer, died at his summer home at Holderness, N.H., on July 1, 1931. In addition to his teaching and consulting work Professor Swain had served on many important commissions and written a number of books on civil engineering; he was recognized as one of the leading authorities in the United States on that subject.

Professor Swain was born on March 2, 1857, at San Francisco, Calif., the son of Robert Bunker and Clara Ann (Fillmore) Swain. After receiving an S.B. degree from the Massachusetts Institute of Technology in 1877, he went to Berlin, where he studied at the Polytechnic School for three years. Returning to this country, he entered the teaching field as instructor in civil engineering at the Massachu-

[1] Chairman, Scientific Exhibits Committee, American Association for the Advancement of Science, Washington, D.C.

setts Institute of Technology. He was made assistant professor in 1883 and associate professor two years later. In 1887 he was appointed Hayward professor of civil engineering and head of the department. During these early years at Cambridge he spent several summers at Lowell, Mass., in the Offices of Locks and Canals on the Merrimac and did some engineering work for Essex County at Lawrence, Mass. From 1880 to 1883 he was hydraulic expert for the tenth United States census, reporting on water power of the Atlantic slope.

In 1909 Professor Swain became Gordon McKay professor of civil engineering at Harvard University and he held this post for twenty years, until a shock in 1929 necessitated his retirement. Perhaps no other man has contributed so much to the development of correct methods of both the presentation and assimilation of engineering knowledge.

The Massachusetts Railroad Commission was the first of many bodies to avail itself of Professor Swain's knowledge of railway engineering. He was consulting engineer to the Commission from 1887 to 1914, during which time he had supervision of bridges in the state and made many important valuations. In 1893 and 1894 he was a member of the Boston Subway Commission and from then until 1918 a member of the Boston Transit Commission, serving as its chairman during the last five years of that period. The subway system of the city was begun in 1895, and Professor Swain was largely responsible for its development.

His appraisal work extended to the New York, New Haven & Hartford Railroad, on whose assets and liabilities he reported to the Massachusetts State Commission in 1910 and 1911, and he made like valuations of the New York Central Railroad and Chicago Elevated Railway in 1911 and 1912. In 1916 he was engaged by East Orange, N.J., to serve as an expert on the elimination of grade crossings in that city by the Delaware, Lackawanna & Western Railroad, and also by the Royal Commission on Railways and Transportation to make valuations of certain Canadian railways. The city of Rochester, N.Y., consulted him in 1921 and 1922 in connection with subway construction, and during the next two years he was a member of two engineering boards, representing New York, to report on New York and Brooklyn elevated railways. In 1927 he again served on a committee on the New York transportation problem. Many other states and municipalities, as well as private concerns, sought his help as a consulting engineer in solving problems relating to railway and highway crossings, subways, and bridge and dock constructions, and he was frequently in demand to give expert testimony in cases involving civil engineering.

Professor Swain also served on a number of commissions and boards dealing with matters other than railway engineering, including the Commission to Revise the Building Laws of Boston (1904), United States Conservation Commission (1909), Commission on Graduate Instruction at the United States Naval Academy, as chairman (1916), and Board of Arbitration on Reconstruction of the Galveston Causeway (1916). In 1918 he was a member of a delegation of American engineers invited to discuss rehabilitation problems at the French Engineering Congress. A Franco-American Engineering Committee was formed on which all the American delegates were asked to serve. In 1921 he was in charge of an investigation and report on the Toronto waterfront problems for the Canadian Pacific Railroad and Grand Trunk Railroad.

The first award of the Benjamin G. Lamme gold medal, established by the Society for the Promotion of Engineering Education "for accomplishment in technical teaching or actual advancement of the art of technical training," was made to Professor Swain in 1928. He received an honorary degree of LL.D. in 1906 from New York University and in 1918 from the University of California.

Professor Swain had a continuous record of important contributions to engineering literature since 1880. In addition to many papers for engineering periodicals and treatises for students he wrote a number of books, including "Conservation of Water by Storage" (1914), "How to Study" (1916), "The Young Man and Civil Engineering" (1924), "Strength of Materials" (1924), "Fundamental Properties of Materials" (1924), and "Stresses, Graphical Statics and Masonry" (1927).

Professor Swain was director of the American Society of Civil Engineers in 1901–1903, vice-president in 1908–1909, and president in 1913, and was made an honorary member of that society in 1929. He had also served as president of the Society for the Promotion of Engineering Education (1894–1895), and of the Boston Society of Civil Engineers (1896–1897). He became a member of the A.S.M.E. in 1883, was a fellow of the American Academy of Arts and Sciences, honorary member of the American Institute of Steel Construction, and a member of the National Academy of Sciences, New England Railroad Club, American Institute of Consulting Engineers, Engineering Institute of Canada, Institution of Civil Engineers (Great Britain), American Railway Engineering Association, American Electric Railway Association, American Society for Testing Materials, New England Water Works Association, and the American Forestry Association. His clubs included the Boston Art, Boston City, Harvard, University, Union, St. Botolph, and Commercial. He had served as a member of the Board of Judges of the Hall of Fame, New York, since it was first established.

Professor Swain was married three times. He is survived by his widow, Mary A. (Batchelder) Rand Swain, of Brookline, Mass., and by two daughters and a step-daughter.

WILSON ERWIN SYMONS

Wilson Erwin Symons, consulting engineer, well known in the railroad field, died of heart failure on May 1, 1931, at the Elks Club, New York, N.Y., where he hade made his home for the past fifteen years. He had been a member of the A.S.M.E. since 1899 and served on its Finance Committee in 1919. In the same year he was also treasurer of the American Welding Society. Other organizations to which he belonged were The Franklin Institute, the Western Society of Engineers, the American Railway Association (Mechanical Division), the American Railway Master Mechanics Association, and the Master Car Builders Association of America. He was also a member of the Society of Friends and the Odd Fellows, as well as of a number of clubs.

Mr. Symons was born at Farmland, Ind., on December 18, 1858. He attended the public schools of Wayne County, Ind., and learned the machinist's trade with the Nordyke & Marmon Co., which moved from Richmond to Indianapolis during his apprenticeship with it. He was employed in 1878 and 1879 as a machinist at the Atlas Engine Works, Indianapolis, and during the next two years at the Columbian Iron Works, Chicago. He was also connected for a time with the Chicago, Rock Island & Pacific Railway in a similar capacity.

For four years beginning in 1881, Mr. Symons was chief engineer in the Merchant Marine Service on the Great Lakes. During the next three years he was locomotive engineer for the Wisconsin Central Railway Company, the Atlantic & Pacific Railway, and the Atchison, Topeka & Santa Fé Railway Co. After being operating engineer for a year at a Chicago steam electric plant, he became division foreman for the Atchison, Topeka & Santa Fé Railway Co. at Chanute, Kan., later being promoted to the position of master mechanic at Raton, New Mexico. He went to Mexico as master mechanic of the Mexican Central Railroad at San Luis Potosi, in 1895. His work during the next two years as mechanical expert for the Galena Signal Oil Company took him to various parts of the United States, and to England and France. Upon returning to this country he was engaged for four years by the System Railways at Savannah, Ga., as superintendent of motive power and equipment. From there he went to Cleburne, Tex., to become a mechanical superintendent for the Gulf, Colorado & Santa Fé Railway Co. In 1904 and 1905 he was superintendent of machinery for the Kansas City Southern Railway Company, at Pittsburgh, Kan.

In 1906 Mr. Symons entered the consulting field as a member of the firm of Burt & Symons in Chicago. He retained this connection for ten years. He was also made president of the Pioneer Cast Steel Truck Company, of Chicago, in 1909, superintendent of motive power and machinery for the Chicago Great Western Railway, St. Paul, Minn., the following year, and superintendent of motive power for the San Antonio & Aransas Pass Railway Co., San Antonio, in 1913. His partner, Mr. Burt, was ex-president of the Union Pacific System. The firm handled problems in connection with the rehabilitation of railways, their valuation, rate cases, etc. In 1911 Mr. Symons delivered an address before The Franklin Institute on "The Practical Application of Scientific Management to Railway Operation." This was inspired by attacks on railway efficiency made by opponents to requests of carriers for an advance in rates, at hearings in 1910 and 1911.

From 1916 to 1919 Mr. Symons was vice-president of the Wilson Welder & Metals Co., Inc., at New York. Since then he had carried on a consulting practice in that city and had also served during part of the time as associate editor of the engineering publications of the Angus Sinclair Company, especially of *Railway and Locomotive Engineering*. During his connection with the System Railways at Savannah, Mr. Symons designed a locomotive that made a speed record of 120 m.p.h. on a special run made for the purpose of securing a Government mail contract. During the latter part of his life he spent a good deal of time in developing his patents on appliances for steam locomotives and other rolling stock of railroads, such as draft gears and locomotive exhaust nozzles. He was consulting engineer for the Congressional committee conducting an investigation regarding the adoption of the metric system, and had been consulted as an

expert in regard to the moving of the shops of the Boston and Maine Railroad from Concord, N.H.

WILLIAM HINMAN TABOR

William Hinman Tabor, time-study chairman for the Gilbert & Barker Manufacturing Co., West Springfield, Mass., died from septic poisoning on October 22, 1931.

Mr. Tabor was born at Elmira, N.Y., on April 26, 1892, the son of Leroy and Anna F. Tabor, and attended the public schools there. The family moved to Philadelphia in 1907, and Mr. Tabor was employed as a toolmaker by The Tabor Manufacturing Company of that city for several years. At the same time, he attended night classes in scientific management at Drexel Institute. The family returned to Elmira for the year 1910-1911, during which time he was under instruction in the toolroom of the Willis-Morrow Company there. He was again employed by The Tabor Manufacturing Company when the family relocated in Philadelphia and continued his work there until 1913, when he was appointed instructor in scientific management and in machine shop practice at The Pennsylvania State College. He resigned his position in 1916 to become associated with the Winchester Repeating Arms Company of New Haven, Conn. For three years he was connected with its tool division as industrial engineer, for one year was production superintendent of the Forged and Edged Tool Division, and for a year production engineer working on cost reduction by improving manufacturing methods, tool equipment, and factory layouts. During these years, he also conducted consulting work as a member of the firm of Diemer & Tabor. In 1922, he returned to Philadelphia to install production control and time-study standards for the Miller Lock Company. Upon the completion of this work in 1923, he became assistant to the general manager of the A. J. Deer Company, Hornell, N.Y.

His association with Gilbert & Barker dated from 1925. In addition to the regular time-study and job analysis work for which he was responsible, he assisted in the selection and layout of factory equipment and in improvements in tools and methods. He was also connected in 1925 with the time-study and rate division of the Willis-Morrow Company.

Mr. Tabor joined the A.S.M.E. as a Junior in 1914 and became an associate member in 1920. He is survived by his widow, Maud E. (Hinmand) Tabor, whom he married in 1910.

GUSTAVE ROBISHER TUSKA

Gustave Robisher Tuska, consulting engineer and lecturer of recognized authority on railroad engineering and maintenance, was born in New York, N.Y., on July 15, 1869. His parents were Adolph and Elsie (Robisher) Tuska.

Colonel Tuska received the degree of Bachelor of Science from the College of the City of New York in 1888 and a master's degree from the same institution three years later. In 1891 he also was graduated from the School of Mines of Columbia University as a civil engineer.

For his first position following graduation Colonel Tuska secured employment as assistant engineer in the drawing room of the Link-Belt Engineering Company of Philadelphia, Pa. Early in 1892 he returned to New York as bridge engineer for the Long Island Railroad Company. The following year he took up the dual role of instructor in bridge design and masonry construction at Columbia University, and consulting engineer. After three years at Columbia he decided to give all of his time to consulting work and did not return to lecturing at Columbia until 1907. His later lectures dealt with the subject of municipal engineering and waste disposal.

In consulting work Colonel Tuska served both public and private enterprises. Between 1893 and 1899 he had a variety of assignments, including the design and rebuilding of the Lonesome Valley Viaduct of the Knoxville, Cumberland Gap & Louisville R.R. and the Stone Brook Viaduct of the Central New York & Western R.R., the design of the dome of St. Matthews Church, Washington, D.C., a pier and pier shed for the New York Central Railroad at New York, an elevator plant for Kohler's Brewery in that city, and a plant for the Atlantic Fisheries Company at Cape Charles. During the last three years of this period he was chief engineer in charge of design, inspection, and maintenance of terminals, wharves, and tracks for the Panama Railroad and superintending engineer for the Panama Steamship Company. For several years he was also engineer to the Health Department of the City of New York.

Subsequent to 1900 Colonel Tuska was for many years president and chief engineer of the American Process Company, builders of plants handling waste products, and of the Atlantic Construction Company, constructors of plants for manufacturing oils and ammoniates. His consulting work included water power plants in New Hampshire, a gas plant in Syracuse, N.Y., piers in New York, and trolleys in Manchester, N.H., and Richmond, Va. Other projects took him to St. Louis, Mo., Chicago, Ill., Buffalo, N.Y., and Washington, D.C. He was elected president and director of the International Engineering Corporation in 1923. His wide experience caused him to be called upon for expert testimony and investigations of patents and his work also included the examination of boiler plants and electric lighting and central steam heating plants.

When the United States entered the World War he joined the Engineers Corps as a major. He was attached to Camps Lee and Humphreys in Virginia and later assigned to a special staff and worked with the director-general of military railways. In 1919 he was on the staff of the Chief of Engineers in Washington, D.C. He was made a Lieutenant-Colonel of Engineers, in the Officers Reserve Corps, U.S.A., in 1923.

Colonel Tuska also rendered service on several government commissions. During 1918 he was a member of the steel committee of the War Industries Board and after the close of the War was a member of the United States Mexican Commission. He was a delegate to the second and third Pan-American congresses.

Much of his experience was recorded by Colonel Tuska in papers for the engineering and scientific organizations of which he was a member. He was a fellow of the American Association for the Advancement of Science, and belonged to the American Academy of Political and Social Science, National Institute of Social Science, American Society for Municipal Improvements, American Public Health Association, Pan-American Society, Society of Naval Architects and Marine Engineers, American Society of Civil Engineers, and The American Society of Mechanical Engineers, which he joined in 1900. He was a life member of the Metropolitan Museum of Art and American Museum of Natural History (New York), a trustee of the Allied Patriotic Societies, director of the Technical Institute of New York, and Past Commander of the New York Commandery of the Military Order of Foreign Wars of the United States. He belonged to the Phi Beta Kappa, Theta Delta Chi, and Pi Gamma Mu fraternities and Kings Crown of Columbia University. His clubs included the Bankers, Lotos, Century, Baltusrol, and Engineers' in New York, Cosmos, Washington, D.C., and Union Interalliée in Paris.

Despondence because of poor health led Colonel Tuska to take his life by shooting on May 28, 1931. His widow, Isabel (Pappenheim) Tuska, whom he married in 1902, survives him.

CHARLES REGINALD VAN TRUMP

Charles Reginald Van Trump was born near Baltimore, Md., on April 14, 1867. He was graduated from Cornell University with an M.E. degree in 1889. Prior to, and during his college days, Mr. Van Trump secured practical experience in the drawing room and shops of the Trump Bros. Machine Co. Following his graduation, he entered the Instrument Shop at Cornell and later spent some time in the Testing Laboratory of the Edison Machine Works at Schenectady. At the beginning of 1890, he took charge of the Electrical Department of the central station at Wilmington and was subsequently promoted to the position of general superintendent there. In 1891 he accepted a position as engineer and manager of the Wilmington City Electric Company.

In association with his father, Mr. Van Trump conducted experimental work in electric lighting, during which he lost his eyesight. Through his efforts, the Delaware Commission for the Blind was organized and The Blind Shop founded in Wilmington. The remainder of his life, until his death in 1924, was devoted to teaching the blind people of Delaware how to become self-supporting.

Mr. Van Trump was a life member of the A.S.M.E., which he joined as a junior in 1893, and in which he attained full membership five years later.

AUGUSTAS THEODORE VICK

Augustas Theodore Vick, who was killed in an automobile accident near Houston, Texas, on November 7, 1931, had been a member of the A.S.M.E. since 1911. He was active in the work of the Houston Section, and a member of its Executive Committee.

Mr. Vick was born on April 11, 1877, at Marion, Ill., the son of Monroe Tyler and Elvyra R. (Roberts) Vick. He attended high school at Anna, Ill., and served an apprenticeship in the machine and repair shops of H. M. Detrich of that place. In 1903, he entered the employ of the Western Electric Company, in St. Louis, and was sent to Desoto, Mo., as superintendent of the Consumers Electric & Power Co. He was next connected with the Hoyt Metal Companies, as chief engineer and electrician of their plant at Granite City, Ill. From 1905 to 1907 he was stationed in the Test Department of the Hawthorne, Ill., plant of the Western Electric Company. In 1907

and 1908, he was chief engineer at the plant of the American Car & Foundry Co. at Madison, Ill.

For some years, beginning in 1908, Mr. Vick was identified with the Missouri, Kansas & Texas Railway Co. His headquarters were at Overland, Mo., and his duties included the design and specifications for a number of new plants for the company. He went to Houston in 1916, and shortly afterward became manager for the P. E. Newberg Electric Company of that city. The contracting firm, A. T. Vick Company, of which he was president, was formed in 1919.

Mr. Vick is survived by his widow, Mrs. Sue. B. (Marshall) Vick, whom he married in 1916.

GEORGE A. VOLZ

George A. Volz, demonstrator and survey man for Cincinnati Grinders, Inc., Cincinnati, Ohio, died on September 19, 1931. He is survived by a son, Milton R., and a daughter, Helen R. Volz.

He was born at Cincinnati on October 2, 1894, the son of Joseph H. and Anna (Busam) Volz. He received training as a machinist and tool and die maker in various plants in which he was employed from the age of fifteen. In 1918 he was engaged as a model maker at the Eagle Manufacturing Company of Cincinnati, developing working models from patent drawings and specifications. He was successively employed as toolmaker by the Van Blerck Motor Company, Monroe, Mich., and the Weis Fibre Container Corporation of that city, the Lincoln Tool Company, the Blakeley Manufacturing Company, and the Briggs Manufacturing Company, all of Detroit, as tool and die maker from 1919 to 1925. Ill health in 1926 kept him from steady employment. The following year he worked for a time for the American Can Company and since then had been employed by the Cincinnati Grinders, Inc.

Mr. Volz was an associate member of the A.S.M.E., which he joined in 1930.

LUCIUS MORTON WAINWRIGHT

Lucius Morton Wainwright, president of the Diamond Chain & Manufacturing Co., Indianapolis, Ind., who died on February 24, 1931, was born at Noblesville, Ind., on January 4, 1860. His parents were William A. and Hannah Guy (Pontius) Wainwright.

Mr. Wainwright was a salesman for bicycles for a number of years after he was graduated from the Noblesville High School. For two years he also was in charge of construction for the Pipe Line Department of the Ohio Oil Company, Lima, Ohio. In 1890 he organized the Central Cycle Manufacturing Company, Indianapolis, and until 1898 served as its president and designed and superintended the manufacture of Ben Hur bicycles. Two years later he took charge of the Diamond chain factory of the American Bicycle Company at Indianapolis and developed pitch chains for the transmission of power. In 1905 he organized the Diamond Chain & Manufacturing Co., which took over the Diamond factory, and as its president and general manager engaged in the development and manufacture of chains and sprockets for the transmission of power and the production of automatic machines for making chains.

Mr. Wainwright was twice married. His first wife, Victoria Gray, of Noblesville, died in 1893. He is survived by his second wife, formerly Edith Palmer, of New York, whom he married in 1900, and by one son, Guy A. Wainwright, vice-president of the Diamond Company.

Among the organizations to which Mr. Wainwright belonged were the A.S.M.E., which he joined in 1907, the Society of Automotive Engineers, and a number of clubs, including the New York Yacht and the Manhasset Bay Yacht clubs. For more than twenty years he was treasurer of the Motor & Accessory Manufacturer's Association. He was a director of the Fletcher American National Bank and of the Morris Plan Company. He died at his winter home in Miami, Fla., after a long illness.

HENRY HEILEMAN WAIT

Henry Heileman Wait, who died on November 16, 1931, was born in Chicago, Ill., on October 27, 1869, the son of Horatio Loomis and Chara Conant (Long) Wait. He was graduated from the Chicago Manual Training School in 1886 and received a B.S. degree in electrical engineering from the Massachusetts Institute of Technology five years later. In 1891 and 1892 he was employed by the Economic Electric Manufacturing Company, Brockton, Mass., making a special investigation of incandescent lamps. From then until 1905 he was associated with the Western Electric Company in Chicago, first as draftsman and subsequently as assistant electrical engineer in the dynamo department and as chief engineer of the power apparatus department. From 1906 and 1913 he was president of the Rateau Turbine Company in Chicago. During the next four years he engaged in design and manufacture of the Wait bulldog turbines and dynamos. Since 1917 he had been identified with the Hill Pump Company of Anderson, Ind., which was merged with the Midwest Engine Company in 1918, manufacturing the Wait bulldog turbines.

Mr. Wait was especially active in the evolution and improvement of dynamos, both of bipolar and multipolar types, including type "L" Western Electric multipolar direct-current generators. He was the patentee of a varipolar motor or dynamo and semi-automatic machine for winding small armatures, spools, etc., as well as improvements on arc lamps. He was also the inventor of the Wait turbogenerator.

Mr. Wait became a member of the A.S.M.E. in 1905. He also belonged to the American Institute of Electrical Engineers, the Institution of Electrical Engineers, and the Société Internationale des Electriciens, and was a lieutenant in the Illinois Naval Reserves from 1898 to 1905. He is survived by his widow, Edna (Kimball) Wait, whom he married in 1913, and by three sons.

GEORGE LLOYD WALL

George Lloyd Wall, retired lieutenant-colonel, U. S. Army, Washington, D.C., died on May 30, 1931. Colonel Wall was born at Hoboken N.J., on January 1, 1872, the son of Edward B. and Sarah B. Wall. He received an M.E. degree from the Stevens Institute of Technology in 1893. During the next two years, he served an apprenticeship as a machinist and worked in the drawing room of the Southern Railway Company at Knoxville, Tenn., and for the next year he was resident inspector of the Department of Tests of that road.

In 1897 and the first half of the following year Colonel Wall was employed by the Hazelton Boiler Company. He then became motive power inspector for the Pennsylvania Company, with which he remained for several years. For a time he was assistant foreman of the erecting shop and roundhouse of the company at Ft. Wayne, Ind., and later, general foreman of the Erie and Ashtabula Division, reporting direct to the superintendent of motive power. He was assistant engineer of motive power at Ft. Wayne in 1903 and subsequently director of tests for the P.R.R. system during the Louisiana Purchase Exposition.

For ten years, beginning in 1906, Colonel Wall was associated with the Lima Locomotive & Machine Co. and its successor, the Lima Locomotive Corporation, of Lima, Ohio, serving as its vice-president from 1913 to 1916. During the following year, he was chairman of the Subcommittee on Designs and Specifications for Standard Freight Car Equipment of the American Railway Association.

When the United States entered the World War in 1918, Mr. Wall was commissioned a major in the Ordnance Department and stationed at the Watertown Arsenal, Watertown, Mass. He was made a lieutenant-colonel the following year and served successively at Camp Eustis, Va., the Erie Proving Ground, LaCarne, Ohio, the Philippine Ordnance Department at Manila, and the Frankford Arsenal in Philadelphia, where he was Commanding Officer in 1928 and 1929. He had retired from service and established his home in Washington a short time before his death. His wife, Myra K. (Crane) Wall, predeceased him.

Colonel Wall became a junior member of the A.S.M.E. in 1897 and was promoted to full membership in 1904. He belonged to a number of clubs, including the Union League.

HAROLD MERRIOTT WALMSLEY

Harold Merriott Walmsley, a member of the A.S.M.E. since 1917, died on January 5, 1931, at the Christ Hospital in Cincinnati, Ohio, from embolism following an operation for appendicitis.

Mr. Walmsley was born in Chicago, Ill., on March 11, 1880, the son of William and Mary (Wrigley) Walmsley. He was graduated from Purdue University with a B.S. degree in electrical engineering in 1905. During his college vacations he worked as a draftsman for the Western Electric Company, Chicago, as construction foreman for the C.F. & I. Co., Pueblo, Colo., as engineer in the Maintenance of Way Department of the N.Y.C. & H. R.R., at Buffalo, and as construction engineer for the South City Chicago Railway.

Following his graduation Mr. Walmsley took an apprentice engineering course at the General Electric Company, Schenectady, N.Y. In 1909 and 1910 he was engaged by the Commonwealth Edison Company, Chicago, as construction engineer at two of its stations. After completing this work he became associated with Sargent & Lundy and the American Gas & Elec. Co. He had complete charge of the construction of a new boiler plant for the Canton (Ohio) Electric Company, a new 9,000 kw. station for the Wheeling (W. Va.) Electric Company, and a new 6,000 kw. station for the Ohio Light & Power Co., Newark, Ohio. In 1914 and 1915 he engaged in contracting and engineering as a member of the firm of the

Steam Electric Engineering Company, at Wheeling, and during the next two years was resident engineer for Sargent & Lundy in charge of a 120,000 kw. plant for the Union Gas & Elec. Co., Cincinnati, until 1918, when he went to Tulsa, Okla., as resident engineer for Sargent & Lundy, in charge of a new power plant for the Oklahoma Power & Light Co.

Mr. Walmsley returned to Cincinnati in 1919 to become chief engineer for the Post-Glover Electric Company, with which he remained until 1925. From then until 1929 he was district sales manager for the Champion Switch Company and Jeffrey-DeWitt Insulator Company, Cincinnati, and since that year had maintained his own office for consulting work and sales engineering in Cincinnati.

Mr. Walmsley's widow, Josephine H. Walmsley, survives him.

EDGAR CARRUTH WALTHALL

Edgar Carruth Walthall, who was killed in an aeroplane accident on August 21, 1931, was born in Atlanta, Ga., on January 19, 1903, the son of W. P. Walthall. He attended the Technical High School there and was graduated from the Georgia School of Technology with a B.S. degree in mechanical engineering in 1924. After a short time as draftsman for Warren Webster & Co., at Atlanta, he secured a position as sales engineer for the Boiler Equipment Service Company of that city. In 1929, he joined the sales force of the Combustion Engineering Corporation and was connected first with its Atlanta office and later was made manager of the New Orleans branch office.

Mr. Walthall became interested in aviation about two years before his death. In 1930, he enrolled for a training course at Kelly Field, San Antonio, Texas, from which he was graduated in June, 1931, and was commissioned a lieutenant in the Air Corps. He had taken part in recent aerial maneuvers along the Eastern Seaboard and, at the time of his death, was stationed at Fort Crockett, Galveston, Texas. Together with another member of the Army Air Corps, Lieutenant Walthall was making low-altitude observations when the plane in which they were flying struck a power line and dropped into the Brazos River, near Freeport, Texas.

Lieutenant Walthall became a junior member of the A.S.M.E. in 1927, and belonged to a number of social and technical clubs.

GEORGE AUSTIN WARD

George Austin Ward, design engineer for the Wilputte Coke Oven Corporation, New York, N.Y., died at the Mountainside Hospital, Montclair, N.J., on January 19, 1931, of complications following an operation. He had been in poor health since Christmas.

Surviving Mr. Ward are his widow, Mrs. Susie G. Ward; a son, Dr. Raymond S. Ward, of Montclair; and a daughter, Mrs. Edwin R. Reed, of Bloomfield, N.J.

Mr. Ward was born on June 7, 1864, at Worcester, Mass., the son of Austin W. and Sarah Ann Ward. After his graduation from the Worcester Polytechnic Institute in 1887, with a B.S. degree, he did drafting and installation work for the Worcester Elevator Company and the Morgan Construction Company until 1893. During the next six years he was at Springfield, Mass., as mechanical engineer and erecting superintendent for the Springfield Foundry Company, elevator and pump builders and founders. From 1899 until 1901 he was on the staff of the engineering department of the Electric Vehicle Company, Hartford, Conn. He had charge of the design and installation of equipment of a maintenance shop for the New York Transportation Company. During the next two years he was in the engineering department of the McAdams & Cartwright branch of the Otis Elevator Company, New York.

From 1903 to 1910 Mr. Ward engaged in consulting and design work, particularly in connection with patent litigation. After that he had five years' experience as assistant engineer and chief draftsman of the Pearson Engineering Corporation, handling power house design and equipment on Mexican, Brazilian, and Spanish properties. In 1915 and 1916 he had charge of the appraisal squad working on the electrical and mechanical equipment of the Hazelton (Pa.) branch of the Lehigh Valley Coal Company. He had been with the Wilputte Coke Oven Company, contractors for coke ovens with by-product equipment, since that time.

Mr. Ward became a member of the A.S.M.E. in 1919. He was one of the oldest members of the Montclair Glee Club, and belonged to the Republican Club and First Baptist Church there.

HENRY CHRISTIAN WEIKEL, JR.

Henry Christian Weikel, Jr., who died at Baltimore, Md., on July 17, 1931, was born in that city on March 5, 1905. His parents were Henry Christian and Ethel Louise (Stewart) Weikel. He studied four years at the Polytechnic Institute of Baltimore and three years at Johns Hopkins University. In 1925 he entered the employ of the United States Printing & Lithograph Co. at Baltimore, of which he was assistant superintendent at the time of his death. Mr. Weikel's duties with the company during the first two years related to both mechanical and electrical construction and maintenance work. In 1928 he was appointed production engineer and made improvements in design in order to increase production. He had applied for a patent for testing apparatus.

Mr. Weikel became a junior member of the A.S.M.E. in 1929. He is survived by his widow, Elizabeth (Babylon) Weikel.

ROBERT WELER

Robert Weler, of whose death in Paris the Society has been notified, was born at Bayonne, France, on March 15, 1890. He attended the Lyceum of Bordeaux and for several years traveled and studied in various countries. From 1914 to 1918 he was a member of the French Flying Corps. After the close of the war he established the Weler Motor Car Company at Levallois, Perret, France. After two years, during which he worked on the design and production of a cyclecar, he accepted a position as production engineer for the Citroën Motor Car Company in Paris. He was later made assistant to the chief engineer of that company. The Boyer-Weler Engineering Company of Paris and New York was organized in 1924.

Mr. Weler became a member of the A.S.M.E. in 1927.

EDWARD ALSTON WHALEY

Edward Alston Whaley, who died on November 6, 1931, was born at Suffolk, Va., on November 27, 1892. He was graduated from Princeton University in 1914 as a civil engineer. During his college years, he engaged in mill construction and work in lumber camps, and, for a year following his graduation, was superintendent of construction for Baker & Brinkley, general contractors of Norfolk. He then became machinery salesman for W. M. Whaley & Co. of that city, where he remained two years. In 1917, he entered the United States Navy as Lieutenant in charge of construction of new shops at the Norfolk Navy Yard. Later, he designed shops for the Bureau of Yards and Docks at Washington, and was sent overseas to construct naval air stations. After the close of the War, he established the Edward A. Whaley & Co. of Norfolk, and subsequently the Whaley Aircraft & Engineering Co. This plant specialized in the production of certain types of airplane instruments, machinery, etc., largely for the United States Army and Navy.

Mr. Whaley became an associate member of the A.S.M.E. in 1924. He is survived by his mother, Fletcher (Norfleet) Whaley. His father, William Mitchell Whaley, died in 1926.

JOHN RAMSEY WILDE

John Ramsey Wilde, chief engineer of power plants for The Detroit Edison Company, died of pneumonia on March 29, 1931, in Detroit. He is survived by his widow, Margaret (Maloney) Wilde, whom he married in 1889, and by three daughters and a son.

Mr. Wilde was born in New York on September 20, 1865. His early education was secured in Toledo and subsequently he served an apprenticeship as a machinist with the Dry Dock & Detroit Specialty & Machine Co. In 1886 he entered the employ of the Edison Illuminating Company as assistant erecting engineer. He remained with this company and its successor, The Detroit Edison Company, until his death, serving as chief engineer of its Willis Avenue Station from 1891 to 1907, and as chief assistant engineer of the Delray Power Plants during the next six years. He was made chief engineer of power plants in 1913.

Mr. Wilde became a member of the A.S.M.E. in 1915 and also belonged to the Detroit Engineering Society and to the Masonic fraternity.

TURNER A. WILKINSON

Turner A. Wilkinson, for some years connected with the Erie City Iron Works, Baltimore, Md., was born in that city on February 26, 1864. His parents were Thomas Strike and Frances (Bennett) Wilkinson. He attended the Baltimore City College, and in 1881 became assistant superintendent of the Rasin Fertilizer Company in Baltimore, in charge of steam plant and mill machinery. He remained with this company until 1893, when he took charge of the foundry and machine shop, building watertube boilers for the Campbell Zell Company of Baltimore. Ten years later he accepted a position with the Erie City Iron Works, directing erection work. Subsequently he served as manager of the sales office, and at the time of his death on April 6, 1931, he was district sales agent for the company.

Mr. Wilkinson became an associate of the A.S.M.E. in 1918. He is survived by his widow, Louise B. (Hellweg) Wilkinson, and by one daughter.

HARVEY D. WILLIAMS

Harvey D. Williams, who was awarded the Howard N. Potts Gold Medal by The Franklin Institute for his invention of the hydraulic variable speed transmission used by the U.S. Navy for operating gun turrets, airplane hoists, steering vessels, and for various applications in submarines, died on March 9, 1931. He had retired from active professional work, owing to failing health, and had been seriously ill for several months.

He was born in Shaftsbury, Vt., on February 24, 1864, the son of Lewis and Sarah J. (Sleeper) Williams, and received his early education in the public schools of Bennington, Vt., and his technical training at Stevens Institute of Technology, from which he was graduated in 1885 with the degree of M.E. He then entered the employ of the Yale & Towne Manufacturing Co. at Stamford, Conn.

In 1888, he joined the faculty of Sibley College, Cornell University, where he taught mechanical drawing and machine design until 1898, when he became draftsman with the Bureau of Steam Engineering, U.S. Navy Department, and in 1903 was appointed ordnance engineer in the Bureau of Ordnance, Washington, D.C. This latter position he held until his retirement from Government service in 1911.

In addition to his development of the hydraulic variable speed transmission, considered one of the most important artillery inventions of recent times, he did important research work for the Navy Department, on the rifling of guns, constructional details of torpedoes and of submarines, and in the burning of fuel oil. Leaving Washington, he became consulting engineer for the Waterbury Tool Company, Waterbury, Conn., now manufacturer of the hydraulic variable speed transmission in this country, Vickers' Limited of London being the foreign licensee. He also took out numerous patents on forms of gear teeth and on systems of gearing, contributing much to the art of efficient gear transmission and to methods of economical gear production, and devised important improvements in types of ratchet drills and in forms of universal joints.

He had been a member of the A.S.M.E. since 1912, and was also a fellow of the American Association for the Advancement of Science and an honorary member of the Sigma Xi, Tau Beta Pi, and Chi Psi fraternities. He is survived by his widow and a stepson, J. J. Crain.

JOSEPH AUGUSTINE WILSON

Joseph Augustine Wilson died on February 21, 1931, of influenza, at his home in Philadelphia. He is survived by his widow, Helen (Keib) Wilson, whom he married in 1921.

Mr. Wilson was born in New York, N.Y., on July 25, 1899, the son of Charles Paul and Rose (McFadden) Wilson. He spent seven months at the Philadelphia Power Plant Engineering School, and was graduated from Drexel Institute with a B.S. degree in mechanical engineering in 1925. His early experience was secured as draftsman and machinery repair man with J. D. Barcaly & Sons, blueprinting and engineering supply company of Philadelphia, from 1915 to 1917. During the next two years he was with the American Expeditionary Forces, after which he returned to the Barcaly company as foreman of the reproduction department for two years, before entering Drexel Institute.

Following his graduation in 1925, he entered the educational department of the Westinghouse Electric & Manufacturing Co. at East Pittsburgh. After completing the course he was made a traction salesman for the company. Since 1929 he had been in charge of the internal combustion engine sales section at the South Philadelphia Works.

Mr. Wilson served in the Infantry during the World War, was gassed twice and wounded in action. He received the Croix de Guerre for action in the Battle of the Marne. This was not his first military experience, however, as he served on the Mexican Border in 1916 when only sixteen years of age.

Mr. Wilson became a junior member of the A.S.M.E. in 1925.

HENRY G. WOLFE

Henry G. Wolfe, mechanical engineer for the Electric Bond & Share Co., New York, N.Y., died on July 25, 1931, at the White Plains (N.Y.) Hospital, of pneumonia.

Mr. Wolfe was born in Philadelphia, Pa., on August 21, 1872. He was graduated from the mechanic arts course at Drexel Institute in 1895 and from its electrical engineering course two years later. He was machinist for the D'Olier Engineering Company of Philadelphia for six months following his graduation, and draftsman for the Technic Electric Company of that city for the next six months. For a short time in 1898 he served as an oiler in the U. S. Navy. He then became assistant superintendent of the Harrisburg Pipe & Pipe Bending Co., where he remained until the first of June, 1904. Mr. Wolfe went to New York in that year as block engineer for the New York Telephone Company. In 1905 he entered the employ of the New York branch of the Crane Company and two years later became draftsman for the New York Edison Company. He was promoted to the position of assistant chief mechanical draftsman in 1915, engaged in research work at the Edison Laboratories in Orange, N.J., during 1918, and the following year returned to New York to work in the Engineering Department of the New York Edison Company. He resigned in 1920 to become mechanical engineer for the Chile Exploration Company, New York, and after four years with this company accepted a position as assistant mechanical engineer with Thomas E. Murray, Inc. For several years prior to his death, he had devoted himself largely to consulting engineering.

Mr. Wolfe became a member of the A.S.M.E. in 1914. He is survived by his widow, Blanche (Irwin) Wolfe, and by one son, Henry G. Wolf.

ARTHUR JULIUS WOOD

Arthur Julius Wood died on April 18, 1931, a few hours after being struck by a motorcycle in front of his home in State College, Pa. He is survived by his widow, Helen M. (Kerr) Wood, whom he married in 1904, and by one son, Reginald.

Professor Wood was born on September 3, 1874, at Newark, N.J., the son of DeVolson and Frances (Hartson) Wood. His father was a professor of thermodynamics at the University of Michigan and later of mechanical engineering at the Stevens Institute of Technology, and was well known as an engineer, writer, and teacher.

After being graduated from Stevens Institute in 1896 with the degree of mechanical engineer, Professor Wood was associate editor of the *Railroad Gazette* for four years. In 1900 he went to Worcester, where he was instructor in mechanical engineering at the Worcester Polytechnic Institute for two years. During the next two years he was professor of mechanical and electrical engineering at Delaware College, now the University of Delaware, Newark, Del.

Professor Wood became a member of the faculty of The Pennsylvania State College in 1904. For the first four years he was assistant professor of experimental engineering, then associate professor for two years. In 1910 he was made associate professor of railway mechanical engineering, and eight years later professor in that subject. He became head of the department of mechanical engineering in 1922. He formulated the curriculum so that fundamental scientific principles predominated, and built up a staff of instructors devoted to him and working smoothly together.

For three years, 1919–1922, Professor Wood was also in charge of the Thermal Plant of the Engineering Experiment Station, his work dealing with fundamental aspects of heat transfer and their application to various problems of every-day insulation which form such a large portion of many industrial processes and maintenance problems.

Professor Wood was the author of many articles on heat transmission, railway mechanical engineering, and refrigeration. He published one textbook, "Locomotive Operation and Train Control." For some years he was associate editor of *Railroad Mechanical Engineering*.

He was for many years a member of the American Society of Refrigerating Engineers, serving as vice-president, and as president during its twenty-fifth year, 1929. He brought a scientific viewpoint to the discussion of the industrial problems which came before this society which was of particular value because so many of its early members belonged to industrial life.

Professor Wood first joined the A.S.M.E. in 1898, and became a full member in 1907. He was the first chairman, in 1922, of the Central Pennsylvania Section, and had been active in heat-transfer committee work. He was also a member of the American Railway Association (Mechanical Division) and the Society for the Promotion of Engineering Education, a fellow of the American Association for the Advancement of Science, honorary member of Sigma Tau fraternity, and a member of the honorary engineering fraternities, Pi Tau Sigma and Sigma Xi, and of Phi Pi Phi. An M.S. degree was conferred upon him by Penn State in 1916.

WILLIAM PARKER WOOD

William Parker Wood, who died on June 11, 1931, was born at Pueblo, Colo., on March 16, 1905. He attended the Washington, Pa., High School, and was graduated from the University of Pennsylvania in 1926 with a B.S. degree in mechanical engineering. Following his graduation he entered the employ of the Suburban-Counties

Gas & Electric Co., Chester, Pa., as a cadet engineer. He became a junior member of the A.S.M.E. in 1928.

CARL B. WOODWORTH

Carl B. Woodworth, manager of the Railroad Division of the Vanadium Corporation of America, Chicago, Ill., died suddenly at his home on October 24, 1931. He was born at Fort Wayne, Ind., on December 16, 1883, the son of Charles B. and Laura (Case) Woodworth. He attended the public schools there and was graduated in mechanical engineering from Purdue University in 1907. From then until 1916 he served in the Mechanical Engineering Departments of the Wabash and the Baltimore & Ohio Railroads, passing through various grades from machinist to general foreman of the Mt. Clare shops of the Baltimore & Ohio. He then entered the employ of the American Arch Company with whom he remained until 1918, when he was commissioned Captain of Engineers, U.S. Army. He served fifteen months with the A.E.F. on railroad work in France.

Upon his return to this country Mr. Woodworth joined the Foreign Sales Department of the American Locomotive Company and spent six years in the Argentine and Brazil, engaged in sales and service work. During the last three years, he was assistant South American manager for the company. He also handled the business of the American Car and Foundry Company in Brazil. In 1926 he was identified with the Premier Staybolt Company as special technical representative throughout the United States, from which position he joined the Vanadium Corporation of America in May, 1927, as manager of its Western Division. In August, 1930, Mr. Woodworth was made manager of the Railroad Division of the Vanadium Corporation.

Mr. Woodworth was elected an associate member of the A.S.M.E. in 1917 and was promoted to full membership in 1929. He is survived by his widow, Margaret (Moore) Woodworth, whom he married in 1927, and by a daughter, Janette.

BEVERLY LYON WORDEN

Beverly Lyon Worden, president of the Cutler-Hammer Manufacturing Company of New York and Milwaukee, died on March 28, 1931, at his home in West Orange, N.J., after an illness of several months.

Mr. Worden was born in Chicago, Ill., on February 8, 1871, the son of Albert Lyon and Anne (Robson) Worden. He received his B.S. in civil engineering from the University of Wisconsin in 1893, and for several years following graduation was employed by the Wisconsin Bridge & Iron Corp., of North Milwaukee, Wis., spending some time in the shops and drafting room and later serving as chief draftsman and construction engineer. In December, 1901, he left this company to form the Worden-Allen Company, engineers and contractors for fireproof construction. He designed, constructed, and operated its structural steel plant in Milwaukee, and served as president of the company for twenty years.

In 1910 he founded the Lackawanna Bridge Company, and designed, built, and operated its plant at Buffalo, N.Y., continuing to give about half his time to the Worden-Allen Company. He remained president of the Lackawanna Bridge Company until it was taken over by the Bethlehem Steel Corporation in 1924. In 1917 he also assumed charge of the design of the Newark Bay (New Jersey) Shipyard, for the Submarine Boat Corporation, and was general manager of the shipyard during the World War. He introduced the fabricated principle into shipbuilding, and constructed a large number of cargo ships on that principle for the Emergency Fleet Corporation, the parts being manufactured in shops located at different points and shipped to Newark for assembly. In 1918 he bought the Ferguson Steel & Iron Corp., at Buffalo, a company then engaged in the fabrication of steel for barges and buildings and also operating a shipyard. Under his supervision six seagoing tugs for the United States Navy were completed during the World War, after which the shipyard was closed. The fabricating shop was kept in operation for about five years.

Early in 1923 Mr. Worden became identified with the Cutler-Hammer Manufacturing Company as its vice-president and director, succeeding A. W. Berresford as president the following year. He was also interested for a time in the Beaver Board Company, which he served as president and chairman of the board, and was a director of the Buffalo Sand & Gravel Corp.

Mr. Worden became a member of the A.S.M.E. in 1921. He also belonged to the American Institute of Mining and Metallurgical Engineers, the American Institute of Electrical Engineers, the Western Society of Engineers, the National Electrical Manufacturer's Association, the Masonic fraternity, and a number of clubs in Chicago, Milwaukee, Buffalo, and New York.

Besides his widow, the former Katherine Elizabeth Taylor of Milwaukee, whom he married in 1901, he is survived by two married daughters, Mrs. Willis S. Brace and Mrs. William S. Valentine, of East Orange, N.J.

BARZILLAI GARDNER WORTH

Barzillai Gardner Worth, vice-president and chief engineer in charge of power plant design and development of Walter Kidde & Company, Inc., New York, since 1916, died at the Fifth Avenue Hospital in that city on May 25, 1931, following an operation for appendicitis. Surviving him are his widow, Mabel D. (Palmer) Worth, whom he married in 1899, and one son, Barzillai G. Worth, Jr.

Mr. Worth was a fellow of the American Institute of Electrical Engineers and had been a member of the A.S.M.E. since 1917. He also belonged to the American Electrochemical Society, New York Electrical Society, and American Chemical Society.

Mr. Worth had made his home in West Orange, N.J., for some years. He was born at Tenafly, N.J., on June 5, 1880, the son of Archibald C. and Elizabeth H. (Anderson) Worth. He was graduated from the Stevens Institute of Technology in 1901 and immediately entered the employ of Walter Kidde, who has supplied the following statement regarding his work.

"Mr. Worth was essentially an engineer who thoroughly enjoyed his work. He looked upon every new problem as a grand adventure, as an opportunity to offer a solution that would be a step in advance of anything that had been done before. As a result, his whole engineering career was studded with unusual accomplishments in his chosen field of industrial and power plant building. While it cannot be said that he was an inventor in the sense that he evolved fundamental ideas in mechanics, it is true that he had a mind that constantly sought to do old things in new ways by applying what was already accepted in one branch of engineering to uneconomical methods or processes used in another branch.

"One of his first successes was the development of, as far as we can learn, the first push-button elevator. This elevator was built under his specifications by the Otis Elevator Company back in 1905 for John Nehl & Company in Jersey City. You can imagine the difficulty of building this complicated mechanism in those days when the electrical industry was so little prepared in the matter of electro-mechanical devices. This elevator has been in operation up to within the last two or three years.

"As early as 1907 Mr. Worth equipped the plant of the American Hard Rubber Company with alternating-current synchronous motors for the correction of low power factor. This was the first rubber plant and among the first industrial plants to be driven with synchronous motors. In addition to this, some of these motors were laid out as direct drivers for the rubber mills and calendars, something unheard of at that time. Shortly after this he also adapted from central-station practice the use of the push-button remote-control switchboard, another long step forward in industrial plant power applications.

"The spray pond for cooling condenser water built by him at the plant of the Hyatt Roller Bearing Company at Harrison, N.J., adjacent to the main line of the Pennsylvania Railroad Company in 1913, was said to be one of the first in the country. Certainly it was a great novelty to the passengers of the Pennsylvania Railroad, but to the Hyatt Roller Bearing Company it represented a saving of $25,000 per year.

"Mr. Worth's tremendous energies took him into the field of electrochemistry, where he designed a battery of what was then revolutionary electrochemical cells for the manufacture of chlorate of potash, which during our participation in the World War supplied a large part of this chemical for the manufacture of primers or cartridges.

"Something of his sheer versatility may be gleaned from the fact that in the early post-war period he developed a high-carbon electric-furnace tool steel which was far in advance of anything in the market at the time. This experience with the use of electric furnaces so fired his fertile imagination that he set about designing a new principle in the design of electric furnaces in which a creditable start was made before he died.

"All these things and many more were sandwiched between his main work of power-plant design and equipment installations for industrial plants. No matter what job he tackled he brushed aside the obvious solution until he was convinced that his was the most economical way that his lively imagination could devise. As an illustration of this he developed a scheme which revolutionized the method of liming skins in the leather industry when we built the new plant up-state New York. The ancient and accepted method in this stage of tanning leather was to soak the raw skins in lime vats set at floor level. At frequent intervals the skins in each vat were agitated by turning them about with long poles in the hands of skilled and high-priced labor. Mr. Worth set to work to plan a more economical scheme and devised

a system of mechanically operated wooden drums which, at the required intervals, revolved a certain number of times. This new method of agitation reduced the force from fifty men to two men in this one department. The saving in cost of labor in ten years time will go far toward paying for the cost of the entire new plant."

AMBROSE EVERETT YOHN

Ambrose Everett Yohn, general manager for the Huntingdon & Broad Top Mountain Railroad & Coal Co., died on October 7, 1931. He is survived by his widow, Mrs. Edna Rhodes Yohn, of Huntingdon, Pa., whom he married in 1911.

Mr. Yohn was born at Reading, Pa., on October 5, 1870, the son of Charles Rutter and Hannah Rebecca Yohn. He attended the Bethlehem Preparatory School and was graduated from Lehigh University with a mechanical engineering degree in 1897.

In 1918 and 1919, Mr. Yohn was general foreman for the Juniata Division of the Pennsylvania Railroad, and the following year general manager of the coal mines and plant of the Kenrock Coal Company. With the exception of those two years, he was connected, from the time of his graduation until his death, with the Huntingdon & Broad Top Mountain Railroad & Coal Co., being advanced from the position of assistant master mechanic to that of general manager.

Mr. Yohn became a member of the A.S.M.E. in 1897.

Depositories for A.S.M.E. Transactions in the United States

BOUND copies of the complete Transactions of The American Society of Mechanical Engineers will be found in the libraries in the United States and other countries which are listed on the following pages.

Alabama

Auburn — Engineering Library, Alabama Poly. Inst.
Birmingham — Public Library

Arkansas

Fayetteville — Engineering Library, Arkansas University

California

Berkeley — Library, University of California
Long Beach — Public Library
Los Angeles — Public Library
University of Southern California
Oakland — Oakland City Library
Teachers' Professional Library
Pasadena — Library, California Institute of Technology
Santa Clara — Library, University of Santa Clara
San Diego — Public Library
San Francisco — Public Library (Civic Center)
Engineers Club of San Francisco
Mechanics Institute
Stanford Univ. — Library, Stanford University

Colorado

Boulder — Library, University of Colorado
Denver — Public Library
Fort Collins — Colorado State Agricultural College

Connecticut

Bridgeport — Public Library
Hartford — Public Library
New Haven — Public Library and Yale University
Waterbury — Silas Bronson Library

Delaware

Newark — University of Delaware
Wilmington — Wilmington Free Institute

District of Columbia

Washington — Scientific Library, U. S. Patent Office
Library of Congress
Bureau of Standards Library
George Washington University
Catholic University

Florida

Gainesville — University of Florida
Jacksonville — Free Public Library
Miami — Public Library
Tampa — Public Library

Georgia

Atlanta — Carnegie Public Library
Georgia School of Technology
Savannah — Public Library

Idaho

Moscow — University of Idaho

Illinois

Chicago — John Crerar Library
Western Society of Engineers
Library, Armour Institute of Technology
Public Library of Chicago
Moline — Public Library
Urbana — University of Illinois

Indiana

Evansville — Public Library
Fort Wayne — Public Library
Indianapolis — Public Library and Indiana State Library
Notre Dame — Library, University of Notre Dame
Terre Haute — Rose Polytechnic Institute
West Lafayette — Library, Purdue University

Iowa

Ames — Iowa State College
Des Moines — Public Library
Iowa City — State University of Iowa

Kansas

Kansas City — Public Library, Huron Park
Lawrence — Library, University of Kansas
Manhattan — Kansas State Agricultural College
Wichita — Wichita City Library

Kentucky

Lexington — University of Kentucky
Louisville — Speed Scientific School
University of Louisville

Louisiana

Baton Rouge — Louisiana State University
New Orleans — Howard Memorial Library
Louisiana Engineering Society
Public Library

Maine

Orono — University of Maine

Maryland

Annapolis — United States Naval Academy
Baltimore — Johns Hopkins University
Engineers Club of Baltimore
Public Library

Massachusetts

Boston — Northeastern University
Boston Public Library
Cambridge — Harvard University (Engineering Library)
Massachusetts Institute of Technology
Fall River — Public Library
Lowell — Lowell Textile Institute
Lynn — Free Public Library
New Bedford — Free Public Library
Springfield — Springfield City Library
Tufts College — Tufts College
Worcester — Worcester Polytechnic Institute
Free Public Library

Michigan

Ann Arbor — University of Michigan
Detroit — Public Library
Cass Technical High School
Highland Park Public Library
University of Detroit
East Lansing — Michigan State College
Flint — Public Library
Grand Rapids — Public Library
Houghton — Michigan College of Mining & Technology
Jackson — Public Library

Minnesota

Duluth — Public Library
Minneapolis — University of Minnesota
Minneapolis Public Library (Engineering and Circulating Libraries)
St. Paul — James Jerome Hill Reference Library

Mississippi

Columbus — Mississippi Agri. & Mech. College

Missouri

Columbia — University of Missouri
Kansas City — Public Library
Rolla — Missouri School of Mines and Metallurgy
St. Louis — Engineers Club of St. Louis
Public Library
Washington University
Mercantile Library

Montana

Bozeman — Montana State College

Nebraska

Lincoln — University of Nebraska
Omaha — Public Library

Nevada

Reno	University of Nevada Library

New Hampshire

Durham	University of New Hampshire

New Jersey

Bayonne	Free Public Library
Camden	Free Public Library
Elizabeth	Free Public Library
Hoboken	Stevens Institute of Technology
Jersey City	Free Public Library
Newark	Newark College of Engineering
	Free Public Library
New Brunswick	Rutgers University
Paterson	Free Public Library
Princeton	Princeton University
Trenton	Free Public Library

New York

Albany	New York State Library
Brooklyn	Polytechnic Institute
	Pratt Institute
	Brooklyn Public Library
Buffalo	The Grosvenor Library
	Engineering Society of Buffalo
	Buffalo Public Library
Ithaca	Cornell University
Jamaica, L. I.	Queens Borough Public Library
New York	Engineering Societies Library
	Public Library
	College of the City of New York
	Cooper Union
	Columbia University
	N. Y. University Library
Potsdam	Clarkson College of Technology
Rochester	Rochester Engineering Society
Schenectady	Union College
Syracuse	Syracuse University
	Public Library
Troy	Rensselaer Polytechnic Institute
Utica	Public Library
Yonkers	Public Library

North Carolina

Chapel Hill	University of North Carolina (Engineering Library)
Raleigh	North Carolina State College

North Dakota

Fargo	North Dakota State College
Grand Forks	University of North Dakota

Ohio

Ada	Ohio Northern University
Akron	Public Library
	University of Akron
Canton	Public Library
Cincinnati	University of Cincinnati
	Public Library
	Engineers Club of Cincinnati
Cleveland	Public Library
	Case School of Applied Science
	Cleveland Engineering Society
Columbus	State of Ohio Library
	Public Library
	Ohio State University
Dayton	Engineers' Club of Dayton
Toledo	Public Library
	University of Toledo
Youngstown	Public Library

Oklahoma

Norman	Oklahoma University
Oklahoma City	Public Library
Stillwater	Oklahoma Agricultural and Mechanical College
Tulsa	Public Library

Oregon

Corvallis	Oregon State Agricultural College
Portland	Portland Library Association

Pennsylvania

Allentown	Free Library
Bethlehem	Lehigh University
Easton	Public Library
	Lafayette College
Erie	Public Library
Lewisburg	Bucknell University
Philadelphia	Engineers Club
	Drexel Institute
	University of Pennsylvania
	Franklin Institute
Pittsburgh	University of Pittsburgh
	Engineers' Society of Western Pennsylvania
	Carnegie Institute of Technology
	Carnegie Library (Schenley Park)
	Carnegie Free Library of Allegheny
Reading	Public Library
Scranton	Public Library
State College	Pennsylvania State College
Swarthmore	Swarthmore College
Villanova	Villanova College
Wilkes-Barre	Public Library

Rhode Island

Kingston	Rhode Island State College
Providence	Brown University
	Providence Engineering Society
	Public Library

South Carolina

Clemson College	Library, Clemson College

Tennessee

Kingsport	Public Library
Knoxville	University of Tennessee
Memphis	Goodwin Institute
Nashville	Vanderbilt University

Texas

Austin	University of Texas
College Station	Agricultural & Mechanical College of Texas
Dallas	Public Library
El Paso	Public Library
Fort Worth	Carnegie Public Library
Houston	Rice Institute
	Public Library
Lubbock	Texas Technological College (School of Engineering)
San Antonio	Carnegie Library

Utah

Salt Lake City	University of Utah
	Public Library

Vermont

Burlington	University of Vermont

Virginia

Blacksburg	Virginia Polytechnic Institute
Charlottesville	University of Virginia
Norfolk	Carnegie Public Library
Richmond	Virginia State Library

Washington

Pullman	State College of Washington
Seattle	Public Library
	Engineers Club
	University of Washington
Spokane	Public Library
Tacoma	Public Library

West Virginia

Morgantown	West Virginia University

Wisconsin

Madison	Library, University of Wisconsin
Milwaukee	Public Library
	Board of Industrial Education, Vocational School Library
	Marquette University

Wyoming

Laramie	Wyoming University

Depositories for A.S.M.E. Transactions Outside the United States

Argentine

Buenos Aires	Biblioteca de la Sociedad Cientifica

Australia

Adelaide	Public Library of Adelaide
Melbourne	Public Library of Victoria
Perth	University of Western Australia Library
Sydney	Public Library, N. S. W., Sydney

Brazil

Rio de Janeiro	Bibliotheca da Escola Polytechnica
	Bibliotheca Nacional
Sao Paulo	Bibliotheca da Escola Polytechnica

Canada

Montreal	McGill University
	Engineering Institute of Canada
Toronto	University of Toronto, Library

Chile

Santiago	Universidad de Chile, Facultad de Ciencias Fisicas y Matematicas (Engrg. School)

Czechoslovakia

Prague	Masarykova Akademie Prace
	Association of Czechoslovak Engineers

Danzig Free City

	Bibliothek der Technischen Hochschule

Denmark

Copenhagen	The Royal Technical College

England

Birmingham	Birmingham Public Libraries
Bristol	University of Bristol
Cambridge	University of Cambridge
Leeds	University of Leeds
Liverpool	Public Library of Liverpool
London	City & Guild Engineering College
	Institution of Mechanical Engineers
	Institution of Civil Engineers
	Institution of Electrical Engineers
	The Royal Aeronautical Society
Manchester	Manchester Public Libraries (Reference Library)
Oxford	University of Oxford
Newcastle-upon-Tyne	The North East Coast Institution of Engineers and Shipbuilders

Wales

Cardiff	Cardiff Public Library

France

Lyons	University of Lyons
Paris	Écoles Nationales des Arts et Metiers
	École Supérieure de Mécanique et d'Aeronautique
	École Centrale des Arts et Manufactures de Paris
	Société des Ingénieurs Civils de France

Germany

Berlin	Bücherei des Vereines deutscher Ingenieure
	Bibliothek der Technischen Hochschule
Breslau	Bibliothek der Technischen Hochschule
Cologne	Universitäts- und Stadtbibliothek
Dresden	Bibliothek der Technischen Hochschule
Düsseldorf	Bücherei des Vereins deutscher Eisenhüttenleute
Frankfort	Technische Zentralbibliothek
Hamburg	Bibliothek der Technischen Staatslehranstalten
Hanover	Bibliothek der Technischen Hochschule

Germany (Continued)

Karlsruhe	Bibliothek der Technischen Hochschule
Leipsic	Stadtbibliothek
Munich	Bibliothek der Technischen Hochschule
	Bibliothek des Deutschen Museums
Stuttgart	Bibliothek der Technischen Hochschule

Hawaii

Honolulu	University of Hawaii Library

Holland

Amsterdam	Koninklyke Akademie von Wetenschappen
The Hague	Koninklijk Instituut van Ingenieurs
Rotterdam	National Technisch-Schlep-Caartkundig Museum

India

Bangalore	Mysore Engineers Association
Calcutta	Bengal Engineering College
Poona	Poona College of Engineering
Rangoon	University of Rangoon

Ireland

Belfast	Queen's University of Belfast

Italy

Milan	Biblioteca della R. Scuola d'Ingegneria
	Comitato Autonomo per l'Esame della Invenzioni
Naples	Biblioteca della R. Scuola d'Ingegneria
Rome	Biblioteca della R. Scuola d'Ingegneria
	Consiglio Nazionale delle Ricerche presso il Ministero della Educazione Nazionale
Turin	Biblioteca della R. Scuola d'Ingegneria

Japan

Kobe	Kobe Technical College
Tokyo	Imperial University Library
	The Society of Mechanical Engineers
Yokohama	Library of Yokohama

Mexico

Mexico City	Asociacion de Ingenieros y Arquitectos de Mexico
	Library of the Escuela de Ingenieros Mecanicos y Electricistas

Norway

Oslo	Universitetsbibliothek den Polytekniska Forening

Poland

Warsaw	Bibljoteka Publicazna

Porto Rico

Mayaguez	University of Porto Rico

Portugal

Lisbon	Institute Superior Technico

Roumania

Bucharest	Scoala Polytechnica din Bucharest

Scotland

Glasgow	Royal Technical College
	Mitchell Library

South Africa

Johannesburg	South African Institute of Engineers

Sweden

Stockholm	Kungl. Tekniska Hogskolan
	Svenska Teknologföreninger
Gothenburg	Chalmers Tekniska Institut

Turkey

Constantinople	Robert College

Index photographed at the
beginning for the convenience
of the microfilm user.